Inter-Noise

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INTER-NOISE 72

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FOREWORD

We are all aware that noise levels in our cities, in industry, and our homes are steadily increasing. There are two main reasons for this. The numbers of machines which we use are increasing and the power of individual machines is increasing. We find that unless we specifically design a machine to be quiet, usually the sound power it produces is somewhat proportional to the mechanical power. Thus the continuing increase in the demand and use of energy: electrical power, oil and gas suggests that unless we make determined efforts at noise reduction, noise levels will continue to rise. Fortunately noise reduction on many machines can be obtained by the application of existing scientific principles and knowledge. The correct use of reactive and absorptive mufflers, enclosures, vibration isolation, absorption materials and vibration damping materials can often result in considerable reduction in noise levels. For some cases, in industry, such knowledge has been applied to noise problems and noise level reductions in the order of 10 or 20 dB have resulted. In the last fifteen years similar knowledge has been applied to aircraft noise. Noise levels from new passenger aircraft have steadily decreased by 20 PNdB in the last fifteen years and further reductions should be possible. All that is lacking at the present time is our determination to apply such technology widely to the many existing noise problems. For this reason funds are also lacking. However the situation is changing.

The public has recently become increasingly aware of noise. Scarcely a day seems to pass without a mention of 'noise pollution' in the press or on radio or television. Partly due to public pressure, legislation has recently been enacted at the federal, state and local levels. Most of us are familiar with the amendment to the Walsh Healy Act (1969) and the Occupational Safety and Health Act (1970) which restricted noise levels to 90 dB(A) for an eight hour work day in U. S. industry. We may also be aware that in 1969, the Federal Aviation Agency set noise certification standards for new aircraft (FAR-36). Presently before congress is legislation which if passed will become the Noise Pollution Control Act of 1972. This Act would not only require the EPA administrator to set noise emission standards for new products--construction equipment, transportation vehicles, motors, engines, and electrical equipment, but would also authorize many million dollars for development of a national noise control program.

It seems that INTER-NOISE 72 has come at an opportune time in view of the current high interest in and need for noise control. INTER-NOISE 72 has several objectives. One is to provide an opportunity for those interested in advancing noise control engineering to gather together for discussion consultation and the exchange of technical information. Another objective is to provide information and tutorial lectures for those individuals in industry and government who are just now entering the noise field. A two day series of tutorial lectures on the basics of noise control has been arranged as a service. The tutorial papers have been printed in a separate booklet. Although individual noise abatement projects are of course necessary, a concentrated national attack on noise by all concerned is very desirable. For this reason, the first day will include panel discussions on the topic: "What constitutes a practical national program for the control of environmental noise."

This volume of proceedings includes the papers to be presented in the technical sessions on the evening of the first day and the next two days. The papers are divided into eleven different session areas. However the divisions in spme cases are somewhat arbitrary and it is suggested that readers may wish to consult papers in several different session areas. Thanks are offered to the authors who wrote these papers despite their busy schedules and to the session chairmen and organizers who lent their talents and special knowledge to make the planning of this conference possible. Thanks are also offered to the many magazines, journals and organizations which generously helped to publicize this meeting.

The papers in this volume have mainly been printed as submitted by the authors although a few editorial changes have been made as time permitted. In addition a name index, a subject index and an author index were added and also a bibliography on noise control prepared from the references given by the authors. Due to late arrival of some papers, the references from a few papers could not be included in the bibliography.

It is hoped that INTER-NOISE 72 and this volume of proceedings will help further noise abatement efforts in the USA and throughout the world. A welcome is extended to all those attending INTER-NOISE 72, particularly those who have traveled here from abroad.

Malcolm J. Crocker
General Chairman INTER-NOISE 72
and Editor of the Proceedings
September 4, 1972

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HEARING IMPAIRMENT FROM NOISE

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HEARING IMPAIRMENT - OCCUPATIONAL DEAFNESS

As there have been many presentations on the effects of excessive noise exposure on the hearing mechanism, no attempt will be made to go into detail here, but for your aid in judging the need and justification for noise control measures, let me briefly review the highlights of the disease process.

Ears exposed to loud sounds for even short periods of time may show a significant change in their ability to hear soft sounds presented soon after the exposure. If the exposure is short, most of this temporary effect will disappear and no permanent change is apparent. For longer more intense exposures, a permanent change may accompany the temporary effect after only a few years of daily exposure. The permanent change in hearing is noticed first at frequencies above 2000 Hz and, if the exposure is continued, will gradually spread to all frequencies associated with the understanding of speech. In a severe case, the damage to the inner ear may mean that almost no nerve signals corresponding to sounds above 500 Hz will be received by the brain, resulting in an almost total inability to understand spoken words. We can stimulate this effect for your ears by electrically filtering speech and recording it on magnetic tape. What has been described is sensory-neural damage and represents a permanent change that cannot be reversed or even significantly nullified by hearing aids. A second type of hearing impairment results from mechanical damage or blockage in the outer and/or middle ear system and results in a more nearly uniform threshold change at all frequencies. This type of change is not the result of noise exposure, and is often repairable or can be ameliorated by external amplification of the sounds. Both types of threshold shift or hearing loss can be demonstrated through the use of tape recordings.

Although the effect of speech filtering is nearly the same for all of you with normal hearing, the losses you might receive from noise exposure would vary widely. Some of you will acquire impairment at a faster rate than others, while some will not suffer damage until extremely high-intensity exposures are present. As Botsford (1) has indicated, the percent of impaired ears found in a noise exposed population varies from about 10 percent of the population at 90 dBA continuous 8-hr daily exposure to 24 percent at 115 dBA.

There are some data relating impulse sound characteristics to hearing impairment. Most of the information, like that of Coles, et al (2) is for gunfire exposure rather than industrial impulses associated with drop hammers, etc. Multiple impacts in excess of 140 dB peak have been associated with hearing impairment in both exposures but the hazard is also thought to be a function of pulse duration and repetition rate. It is interesting to note that early hearing loss claims were associated with industries having predominantly impulse noise exposures yet today we have relatively little data upon which to base a criterion.

Note that the exposure for steady noise is evaluated in terms of dBA, indicating that the hazard of low frequency sounds is not as great as the middle and upper frequencies. Because the permanent change in hearing acuity associated with noise exposure is gradual in its onset, the person affected may be quite unaware of a change until an appreciable loss has occurred. There are other causes for sensory neural ear damage besides industrial noise, and one finds in the general population an appreciable portion of impaired ears resulting from such things as childhood disease, over-use of certain antibiotics, weapon firing, and even rock music. Removal from the exposure will halt the noise-induced threshold shift from further development.

CLAIMS FOR LOSS OF HEARING

In 1948, a precedent was set in Workmen's Compensation (WC) activities. Prior to this time the existing concept of compensation was loss of earnings and this was modified to yield our present rulings of compensation for loss of function regardless of loss of earnings. The now famous case of Slawinski versus J. H. Williams and Co. in 1948 (3) was the precedent setting decision in New York State, and there followed a number of test cases in other states. The rulings in other jurisdictions in the main reaffirmed the loss of function concept.

At the present time, we find that nearly 1/3 of our states have routine procedures in claims handling for loss of hearing resulting from noise exposure, while in many others, such noise-induced loss is considered to be covered under the occupational disease law. Of these states there are over 13 with specific legislation for noise-induced impairment. Although the interpretation of WC laws with respect to coverage may be changed during litigation, it is probably safe to say that, in most states, claims for WC arising out of valid noise-induced hearing loss will be settled in all but a few jurisdictions.

Even in those states with specific legislation on hearing loss arising out of and during the course of employment, we find considerable variation in the determination of impairment associated with a measured change in pure tone perception by the claimant. The most commonly accepted method of computation is that of the AMA Subcommittee on Noise of the Committee on Conservation of Hearing (4) which is based on the pure tone hearing thresholds (ANSI 1969) at 500, 1000, and 2000 Hz. Averaging the audiogram readings at these three frequencies and subtracting 25 dB, the resulting difference is multiplied by 1.5 to establish monaural impairment. For binaural loss, the percentage of the better ear is multiplied by 5 and added to that of the poorer ear and the sum divided by 6. Thus, the loss of function in one ear is considered only 1/6 of the total. In some jurisdictions, loss at other than the above frequencies may be included in the computation and different factors may be used for weighing monaural loss in calculating binaural loss.

Due to rapid changes in cost of compensation, it is dangerous to attempt any statement on amount of compensation afforded by 100 percent impairment, but the average of the maximum compensation is close to $8000 for 100 percent binaural impairment or approximately $100 per dB. Many claims filed to date have been for less than $1000 total including medical costs.

One might ask how many claims there have been since 1948, but the answer is not readily available since no inclusive registry exists. Our experience would indicate that the number is in the tens of thousands, rather than hundreds of thousands. The direct out-of-pocket cost of hearing loss is thus probably well below 200 million dollars to date, but this can hardly be considered inconsequential.

What exists in the way of potential hearing loss claims? We can only speculate on the basis of rather spotty surveys of noise exposure in industry in conjunction with published data relating exposure to noise-induced hearing loss. As indicated by Botsford (1), the risk of noise induced impairment is approximately 10 percent for a working population exposed to 90 dBA for 8 hr per day or equivalently at higher levels for shorter daily exposures. It would appear that Cox's survey of industrial exposures (5), although old and including only plants of 50 or more employees, is the only available measure of prevalence of noise in industry. His data show that approximately 4.5 percent of the industrial population is exposed to 90 dBA or equivalent. This means that approximately 0.5 percent of the working population may be expected to have incurred noise induced hearing loss in approximately 10 years' duration. Of course, this expresses an estimation based on assumptions with regard to age distribution, temporal patterns, and other factors. It is more difficult to estimate the potential claims liability at some level of exposure as many other factors, such as legal interpretations, waiting periods, and the distribution of hearing loss in the affected group enter into such a determination.

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ENGINEERING CONTROL - EUROPE

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Slowly but surely, management are coming round to the realization that damage to the hearing of their employees can cost them money. Many of them are responding simply by embarking on hearing conservation programmes centred round ear protection, but the more enlightened employers are turning to noise control by means of acoustical engineering. The United Kingdom Department of Employment estimate that there are approximately one million employees working in equivalent continuous noise levels (L_{eq}) of more than 85 dBA, and of these, over 600,000 suffer in excess of 90 dBA.

On top of this, neighbourhood noise is becoming an acute problem as standards of living increase and populations of mixed residential and industrial areas become highly sensitive to industrial noise, particularly at night.

The hearing damage problems are found in two basic types of situation: those of traditionally noisy industries such as foundry work, boilermaking, spinning and weaving, and relatively modern problems which result from the speeding up of processes in the lighter types of industry, for instance can making, bottling, and some processes in the electronics and plastics industry. Noise problems of the traditional type tend to be somewhat intractable unless the industries concerned are prepared to change their ways considerably. In fields such as the rivetting of boilers and fettling of castings, some improvement can of course be readily achieved by methods of damping the workpiece. Noise in the textile industries is the subject of very specialized research, and significant progress is being made.

However, it is not in the traditionally noisy industries that the impetus for noise control is greatest; the workers in them have over generations come to accept noise and its effects as occupational hazards, and certainly their unions take little action. It is in the lighter industries drawing labour from areas not steeped in industrial history where noise is the most widespread problem. One such example was a factory manufacturing chocolate bars. The process involved depositing molten chocolate into aluminium or nylon moulds. The moulds themselves were positioned in metal carriers which were in turn fastened on either side to a large scale 'bicycle chain'. Several hundred moulds were transported in this way along a conveyor mounted on springs; by means of rotating eccentric weights, the entire conveyor was vibrated in the vertical mode in order to ensure even spreading of the chocolate and expulsion of air bubbles. Octave band sound pressure levels measured at an operator's position with a plant in normal production peaked at 104 dB at 4 kHz.

It was found that the noise was being generated in several ways. Firstly, there was a clearance of about 0·080" between the moulds and the retaining surfaces of the carriers which resulted in rattling between the two. This resulted in forced vibration of the moulds which would not respond greatly to damping methods. The solution lay in redesigning the method of clamping the moulds in the carriers. Secondly, the amplitude of vibration of the conveyors was so great that the helical springs on which it was supported were jumping about so that they were excited into giving off a high frequency ringing noise. Simply jacketing the springs in rubber tubing provided the answer here.

Lastly, the bed plate of the conveyor was fitted with a constrained damping system consisting of a cork layer bonded to the plate and backed with another sheet of thin steel and nylon faced foam strips were fitted to the rails along which the chains travel. As a result of these modifications, the 4 kHz peak was reduced from 112 dBA overall to

92 dBA at two feet from the conveyor.

A somewhat different approach was necessary in the case of the new headquarters for Vickers European Group - Sperry Rand Ltd in Surrey, England. The premises include an hydraulic test cell in which it had to be possible to run up to twelve hydraulic test rigs, some as great as 450 h.p. on a 24 hour basis, without disturbing residents in houses less than 200 feet away. This problem called for a basic building structure giving a sound transmission loss approaching 30 dB better than a conventional structure, to cope with internal sound power levels over 125 dB and meet a 35 dBA criterion at the houses.

Fortunately, there already existed a suitably sized building on the site which had 18" brick walls, but this alone was not enough, and of course the most critical aspect of the problem was not so much the walls as the doors, windows and service access apertures and ducts. A sophisticated ceiling was necessary to complement the 8" concrete roof slab, which was a carefully installed suspended ceiling, on rubber supports, of staggered-jointed double layered plasterboard, flexibly sealed at the edges. The cell was lined with 3" thick rock wool retained with perforated metal; the doors were of conventional steel construction and the double glazed observation windows had the benefit of the large interspace. An interesting aspect was the method by which the need to permit easy installation and removal of pipes running to and from the machines was overcome. Large diameter pipes were set into the walls and pipes were fed through. The surrounding space was then filled with rock wool and sealed with mastic at the ends. Because of the length of the large pipes, approximately 2 feet, they behaved like attenuators and small gaps between the small pipes were not detrimental.

The attenuation of noise travelling through the ventilation ducting was also of major importance, particularly in view of the necessity to avoid upsetting the benefit gained by the suspended ceiling. A system of multiple primary silencers, and large secondary units at the fans was designed, both to prevent internal test cell noise causing disturbance and to attenuate the fan noise itself.

One of the most critical decisions which had to be taken at an early stage was whether to provide a floating floor in the cell to cope with all possible sources of vibration, or whether to design individual anti-vibration mounting systems for the machines. The floating floor was an attractive idea because it promised to be relatively 'fool-proof', but the difficulty of achieving a low enough natural frequency by relatively straight-forward means such as fibre quilts meant that a reinforced slab resting on localized pads with the attendant constructional problems would have been necessary. It was therefore primarily for reasons of economy that individual mounting systems were used instead. However, there was the advantage that internal noise levels were reduced by isolating the whole area of the floor from the exciting forces of the machines, and preventing it from radiating machine noise internally.

Generally speaking, it can be said that the European approach to engineering noise control tends to have one major failing. There is as yet an excessive tendency to apply palliative treatment in the form of somewhat traditional sound insulating/absorbent enclosures, screens and room linings. While this method is often successful, a consultant's view ought to be that an acoustical enclosure is a last resort. There is very considerable scope for reduction of noise at source of the type described in the chocolate factory example which is often less costly than the cumbersome enclosures which had previously been attempted in this instance.

Of course the more sophisticated method of approach ought to be carried out by the manufacturer of the machine, and as yet most manufacturers are backward in this respect, but in a large number of cases, particularly with process plant, source treatment is easily and practicably possible.

The main difficulty to be overcome is simply one of widespread ignorance, at least as far as the manufacturers of machinery are concerned. As yet there are not strong enough market forces to coax them into building up their own acoustical research and development facilities, nor do the designers make full use of the expertise available from consultancy groups and universities.

As far as the solution to in-plant noise problems is concerned, much of the demand is supplied by the manufacturers of acoustical materials and devices, and they of course have a vested interest in the use of large enclosures and widespread sound absorbent treatment. The expertise required for efficient source reduction of noise is as yet only available from a small number of organizations, the existence of which large sections of industry are unaware.

We have not reached the stage, for instance where more than a small minority of industrialists when planning a new factory would have the foresight to segregate the noisier operations and construct appropriate buildings in which to house them, or to seek expert advice of any kind on both potential hearing damage risk problems within the building and external problems of nuisance.

As a consultant, one frequently meets cases where plant has been installed in a typical factory which causes acute noise problems and by its very nature must cause the same problems in almost every instance where the manufacturers instal their machines. Yet the makers are at a total loss to overcome the problem when confronted with it by the customer.

As a final example of the potential for acoustical design being applied to prevent generation of noise in the first place, a recent case was one of noise in a bottling plant. Some of the operators in the plant were suffering continuous noise levels of 111 dBA, and many others levels over 100 dBA.

The highest levels were thought to be due to impacts of metal caps in a hopper. The hopper fed the caps down a chute to the machine for filling and sealing the bottles. In fact the noise was almost entirely due to the method by which the caps were forced down the chute by two compressed air jets blowing at high velocity out of small diameter nozzles. Secondly, the bottles themselves on the filler machine were raised into position by pneumatic cylinders, which periodically released jets of air with considerable noise as the bottles were lowered again. Redesign of the nozzles in some cases, and piping the others into a common silenced duct was sufficient to solve the problem. Of more interest was the way in which noise from bottle-to-bottle impact was found to be greatly influenced by the way in which the spacing of bottles was regulated at various points along the conveyor. In one place the bottles were aligned before entering a capsuling machine by means of a star wheel with such broad teeth that it was necessary for the bottles to be forced apart to let the teeth in. The forcing apart of the bottles sent a wave of impacts back down the line, and simply substituting a star wheel with narrower teeth enabled the alignment to be carried out without any increase in noise.

The answer to the problem as a whole must surely be education. Research and development is of course needed on a large scale, but until manufacturers are made more aware of the problems and the scope for improvement, little progress will be made.

MEDICAL ASPECTS OF HEARING CONSERVATION

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A successful hearing conservation program in industry demands management support, employee cooperation and understanding and meticulous attention to the many medical aspects of such programs. Skillful handling of employee problems is essential if protection is to be achieved by all employees.

Commitment of all levels of management is the first step in implementing the program. Of help in obtaining this is a presentation to top management featuring an otologist or medical director who can discuss the problem of hearing loss, in particular, noise induced hearing loss. Management should understand the necessity for a commitment to abate noise where feasible. When short-term abatement is not practical, a hearing conservation program should be developed. Management should thoroughly understand the necessity of a high quality, medically supervised program for success.

Similar educational efforts should be directed to plant management, supervisors, and production employees. A scheduled meeting of approximately one half hour to discuss the program with employees prior to initiation will more than pay for costs involved in terms of better understanding and cooperation. In this meeting, discussions by the plant manager, plant nurse, showing of educational films, and providing descriptive literature all are very helpful.

Once sound measurements and evaluations of abatement possibilities are made, noisy areas in which ear protection is to be required should be designated. There is definite merit in designation of required protection by area rather than by job. For example, an employee who must come into a noisy area of 90 dBA only for one hour per day, would be required to wear ear protection when in that area, even though technically he would not exceed exposure limits set by OSHA. Likewise, visitors and plant management are provided ear protection when entering into the noisy areas. To have a mixture of employees, some wearing ear protection and some not, in a noisy area makes your program extremely difficult to administer and can destroy employee confidence in it. Hearing protection should be mandatorily worn in a noisy area.

Though OSHA regulations must be compiled with, employees should understand that the real reason for the hearing conservation program is protection of their hearing - that the program is a positive benefit for them. A sincere desire upon the part of management and professional personnel to protect employees' hearing will help to insure success of the program. In this respect, any indication to employees that the primary purpose of the program is to comply with the law, in my opinion, would have a negative effect. Also, providing hearing conservation programs in boarderline areas which may be slightly below the 90dBA gives the benefit of doubt to employees, and will be helpful in the event OSHA noise levels are lowered.

Careful evaluation of employees by history and examination prior to fitting of ear protection will save many complications. Nurses and other medical personnel can be trained in obtaining adequate otologic history, and also in technique of screening examinations of ears for infection, perforation, and other conditions which may make wearing of ear protection difficult. Being aware that an employee may have a potential problem with ear protection at the time of fitting lets you properly counsel with the employee and attempt to provide the best means of protection. In many cases, employees are advised of a pre-existing infection and are referred to otologists for correction prior to fitting of protection. Corrective surgery for perforated eardrums has been provided in some cases as a result of this preliminary screening.

For most industrial noise exposures, muffs or properly fitted plugs will provide adequate protection. In my experience, most employees prefer plugs, feeling that they are more comfortable and less cumbersome. We should allow employees the opportunity to select their protector and to exchange it for another type after a trail period, if not satisfactory.

Some shrinkage and hardening of plugs upon wearing has been a problem. Fitting with a size that is slightly tight can avoid a plug that is too small after several weeks of wearing. Universal size plugs are well accepted by some employees.

The condition of plugs should be monitored periodically by those responsible for the hearing conservation program to insure continued pliability, fit, and employee cooperation.

There are a number of reasons why employees have difficulty wearing ear protection. In my experience, poor cooperation is one of the least of these and even this is minimized with proper educational efforts. Simple discomfort with wearing of protectors is the most common reason. Some of this often occurs during the adaption period and can be handled by reassurance and allowing the employee to gradually become accustomed to wearing protection by wearing them for relatively short periods, gradually increasing to an eight hour period over a couple of weeks. Improper fitting can cause discomfort, and not infrequently, it is necessary to fit two different sizes of ear plugs due to the ear canal being larger on one side than on the other. Any dirt or abrasive material on the plugs will cause irritation. Preexisting infection, external otitis or otitis media can be aggravated by wearing of plugs or muffs. A common condition is itching of the external auditory canal with wearing of ear protector devices. There frequently is an underlying mild scaly dermatitis caused by seborrheic dermatitis, hair sprays or scratching. These conditions can often be readily corrected, in addition, placement of a small amount of a corticosteroid ointment on the plug will relieve this.

Temporomandibular joint conditions are a frequent cause of localized tenderness at the cite where the plug creates pressure. More difficult to deal with is the emotionally unstable employee. He frequently has considerable problems wearing ear protection and needs strong reassurance and occasionally treatment of his underlying disorder. Employees with any condition preexposing to dizziness or vertigo often have difficulty wearing protection. These include, hypertension, meniere's disease, labyrinthitis. Employees with eustachian tube dysfunction sometimes related to temporomandibular joint disease, sinus conditions, and other problems predisposing to ear aches may be uncomfortable in wearing ear protectors.

It has been our experience that seldom can we not achieve successful wearing of ear protection by prompt attention to the underlying problems. An occasional uncooperative employee needs firm counseling by management. Those with medical problems need competent handling in the medical department by medical personnel.

Audiometric examinations provide monitoring as to the effectiveness of the program. In addition, they provide an employee benefit; namely, evaluation of hearing status which will often result in initiation of corrective procedures. Finally, they provide medicolegal protection to the employer. We feel that no hearing conservation program is complete without audiometric examinations. Audiometry should be done yearly on those employees exposed to noise over 85 decibels. It has been our policy to include non-exposed employees in the hearing conservation program in plants which have the program.

Audiometry must be performed only by technicians trained at an approved industrial audiometry training course. Monitoring, evaluation, and interpretation of audiograms should be under the direction of an otologist or physician experienced in industrial hearing conservation programs.

Location of audiometry testing facilities should be carefully evaluated for background noise. Except in occasional circumstances where an unusually quiet area is available, audiometric booths are necessary.

Air conduction audiometry is used for screening purposes. Where possible, we believe it most beneficial to obtain bone conduction audiometry on those employees showing 25 or more DB (ISO) loss at the frequencies 500 to 2000. It is our desire to determine whether there might be a corrective conductive loss which is never due to noise. This reduces referrals to those employees who can benefit. This greatly enhances the image of the program. It also provides the company with potential savings in workmen's compensation since these cases are readily established as non-occupational. Special equipment and training of personnel is necessary for bone conduction audiometry.

Audiograms should be done after an absence from noise exposure for at least 16 hours. We believe the initial audiogram should be done prior to the beginning of the shift in order to establish a baseline. Subsequent audiograms can be done anytime during the shift provided the employees are wearing ear protection during work. Any decrease from the baseline audiogram indicates the possibility of ineffective protection. Such audiograms should be repeated again prior to the beginning of the shift, and the employee counseled with about his wearing of protection. In our experience, audiometric monitoring reveals the value of hearing conservation programs. Limited data at this time indicate that over 90 percent of employees are showing no change or improved hearing upon retesting one year later, and less than 10 percent any further loss of hearing. A considerable proportion of this latter group has either non-noise induced hearing loss or are known not to be cooperating fully with the program.

Depending upon how it is presented to employees, the sincerity and skill of involved professional personnel, and the backing of management, a hearing conservation program will be a tremendous benefit, well received by employees or looked upon only as something tolerated because of legal necessity. It is very gratifying to be involved with a good hearing conservation program. A number of employees with correctable hearing loss and other ear conditions will be discovered and helped. Often those who have advanced sensori-neural hearing loss due to noise will improve slightly, but noticeably, due to elimination of temporary threshold shift. Most important, we know that with a successful hearing conservation program, young employees joining the industry with relatively normal hearing will keep it that way.

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EXPERIENCE OF LOCAL AND STATE GOVERNMENTS IN CONTROL OF ENVIRONMENTAL NOISE

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From the colonial period until the middle of the twentieth century, the federal government, and indeed the general populace, was largely unconcerned about noise as a pollutant of the environment. Indeed, noise was generally taken to be evidence of power, of economic activity, of welcome production of material goods and services. Those who complained of noise were often dismissed as romantics, reactionaries, and cranks.

As cities burgeoned, those who could afford to do so tended to move outward to the quiet suburbs, while those who remained were those with the least power to demand amenities such as a clean and quiet environment. The very technologies which once made such escape possible -- the automobile, the airplane, the substitution of machines for labor (in homes and offices as well as in the factory) -- have now caught up with us. Environmental noise assails us not only in the cities, but in the suburbs, in rural areas, and even in the wilderness. Growing dissatisfaction with this situation has brought about very strong pressure to control and abate environmental noise through the political and legal system, at all levels of government.

This paper will briefly outline the regulatory structure for noise control at state and local levels, with emphasis on the general effectiveness of noise control legislation.

States are just beginning within the last two years or so to enact comprehensive noise control legislation. Some states have established Environmental Departments, which deal with noise along with other pollutants. The administrators of such departments may usually set noise standards for any source of environmental noise. Most of these standards are still being developed. In the past, state laws defining noise as a nuisance have seldom been enforced, and almost never against major sources of noise such as factories, transportation equipment and vehicles. Now, however, states are becoming more sophisticated in the writing of anti-noise laws and some are substituting quantified, decibel limits for the traditional subjective standards such as "unnecessary," "unreasonable," and "excessive."

Until recently, the federal government had consistently taken the position that noise was a local concern to be regulated by states and localities under their police power. Yet there are areas, especially where interstate commerce is concerned, where the federal government has steadily resisted efforts of states or localities to control noise on the grounds that such regulation threatens to modify the activity itself. The federal government has effectively preempted the field of regulation of air transportation, including noise control. Yet the Griggs decision in 1962 placed the locus of liability for aircraft noise on the airport operator rather than on either the federal government or the scheduled air carriers. Thus there was in the past no pressing incentive for either the government or the airlines to take drastic steps to reduce noise.

In spite of the fact that federal preemption in the aircraft noise area is almost complete, some states -- driven by widespread protests against noise around airports -- have moved to attempt to set overall noise limits for airports. California took the lead in 1969, empowering the State Department of Aeronautics to set standards both for overall airport noise and for single events (i.e., takeoffs and landings). The law will allow large airports 15 years to shrink their noise contours to an acceptable level as defined under statutory standards of "economic and technological feasibility" and "noise acceptable to a reasonable person living near the airport." Some airport officials allege that unless the fleet is substantially converted to quieter engines

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beginning of the shift, and the employee counseled with about his wearing of protection. In our experience, audiometric monitoring reveals the value of hearing conservation programs. Limited data at this time indicate that over 90 percent of employees are showing no change or improved hearing upon retesting one year later, and less than 10 percent any further loss of hearing. A considerable proportion of this latter group has either non-noise induced hearing loss or are known not to be cooperating fully with the program.

Depending upon how it is presented to employees, the sincerity and skill of involved professional personnel, and the backing of management, a hearing conservation program will be a tremendous benefit, well received by employees or looked upon only as something tolerated because of legal necessity. It is very gratifying to be involved with a good hearing conservation program. A number of employees with correctable hearing loss and other ear conditions will be discovered and helped. Often those who have advanced sensori-neural hearing loss due to noise will improve slightly, but noticeably, due to elimination of temporary threshold shift. Most important, we know that with a successful hearing conservation program, young employees joining the industry with relatively normal hearing will keep it that way.

NOISE LEGISLATION AND ORDINANCES

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EXPERIENCE OF LOCAL AND STATE GOVERNMENTS IN CONTROL OF ENVIRONMENTAL NOISE

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From the colonial period until the middle of the twentieth century, the federal government, and indeed the general populace, was largely unconcerned about noise as a pollutant of the environment. Indeed, noise was generally taken to be evidence of power, of economic activity, of welcome production of material goods and services. Those who complained of noise were often dismissed as romantics, reactionaries, and cranks.

As cities burgeoned, those who could afford to do so tended to move outward to the quiet suburbs, while those who remained were those with the least power to demand amenities such as a clean and quiet environment. The very technologies which once made such escape possible -- the automobile, the airplane, the substitution of machines for labor (in homes and offices as well as in the factory) -- have now caught up with us. Environmental noise assails us not only in the cities, but in the suburbs, in rural areas, and even in the wilderness. Growing dissatisfaction with this situation has brought about very strong pressure to control and abate environmental noise through the political and legal system, at all levels of government.

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In California the single-event limit was deliberately set so high as to be effective only in controlling operating procedures of existing aircraft rather than as a push for technological improvement. Enforcement of the single-event limit is up to the county in which the airport is located. In some states, however, airport noise most detrimentally impacts counties adjacent to but not including the airport. This point should be noted by states considering similar legislation.

Twenty-five states own and operate airports and can exercise some control over them as proprietors. The bi-state Port of New York Authority, for example, established maximum single-event noise levels. This limit is effective in terms of compliance -- the overall compliance rate is 99.5 percent, with 80 percent of takeoffs well below the limit of 105 dBa. Violation rates are high, however, for transoceanic jets. In terms of actual noise reduction the limits are very ineffective -- compliance is high because the limit is high. No punitive enforcement action has been carried out against any airline. There is also said to be systematic cheating, in which pilots cut power momentarily as they pass the monitoring equipment.

More importantly, airport operators, including Port of New York, have no authority over landing procedures, which are controlled by FAA. Landings involve long glide paths and thus subject larger numbers of people to noise than do takeoffs. In New York about 80 percent of the complaints are caused by landings.

Restricting or prohibiting night flights is effective in reducing complaints but may seriously disrupt national and international flight patterns because of time differences. Moreover, as airports become more congested, safety considerations drive toward more, rather than fewer, night flights.

The fiscal condition of most state and local governments and the shortage of housing in large metropolitan areas limit the potential effectiveness of land purchase or special zoning around existing airports. In the case of new airports, however, land-use controls such as industrial buffer zones offer great promise.

Highways are also a part of interstate commerce, but here the courts have traditionally recognized a strong state and local interest in regulation. States, however, have not taken a strong role in control of highway or vehicle noise. Most states prohibit defective or modified mufflers, but few or none systematically enforce this prohibition, in spite of substantial evidence that rigorous enforcement would significantly reduce vehicle noise.

In vehicle noise control, also, California has taken the lead with a comprehensive noise law setting decibel limits for all classes of vehicles, but enforcement is low, with six two-man teams operating over 162,300 miles of highways. In one 12-month period 600,000 vehicles were monitored; 0.5 percent were charged with violations. In the context of a widely recognized highway noise problem, this is clear evidence that the statutory standards are too high to be effective.

A similar law in New York has an even lower level of enforcement. No special enforcement team is provided and reportedly only six violations were charged during the first two years. The reasons for the low level of effectiveness of states' operating vehicle noise laws are:

1) <u>Inadequate standards</u>: setting the acceptable noise level at a point which would cause 7 or 8 percent of vehicles now on the road to be in violation would reduce noise substantially; compliance of the remaining 93 percent argues that it is a technically feasible standard.

2) Technical difficulties in monitoring noise sources and identifying the individual offender. In California, for example, there must be 100 feet of free space around both the monitoring device and the vehicle, and noise must be monitored at 50 feet from the center line, in order to prove a violation. This makes it impossible to monitor trucks and automobiles on city streets and crowded highways. Some states have strict muffler laws, but the only inspection is in state-licensed commercial garages, where there is no sound monitoring equipment.
3) Disregard of noise sources other than engines and exhaust systems. Much vehicle noise comes from tires and running gear, but in California, in spite of statutory provisions specifying control of total vehicle noise, state police do not cite offenders whose noise stems from other than engines and exhaust systems, on the grounds that the individual has little choice in purchase of tires.
4) Assignment of enforcement responsibility to regular police officers (other than in California). Police universally give highest priority to criminal apprehension and to safety; they also rapidly lose proficiency with measuring equipment when they rarely use it, thus becoming even more reluctant to use it.
5) The low statistical probability of any one vehicle being monitored and the low penalties involved do not encourage compliance; this may be the single most important factor in a low level of effectiveness.

Some states now have noise laws dealing with leisure vehicles such as motorboats and snowmobiles. Standards are generally not strict enough to push technological changes, and the enforcement of such laws is often left to game wardens and conservation officials. It is generally conceded to be at best spotty.

The major burden in control of most kinds of environmental noise falls on local governments. They frequently express a need for technical assistance from states and federal government in the form of guidelines, standards, model ordinances, and financial aid. However, local jurisdictions are also jealous of local prerogatives and generally wish to preserve the option of adopting stricter standards than they expect the larger jurisdictions would be likely to choose. Communities near very large airports perceive themselves to be in a state of seige from noise intrusions which they are powerless to control.

Attempts by local governments to prohibit or restrict airplane noise have almost without exception been struck down. On the other hand, as proprietors and owners of airports local governments often find themselves the defendants in suits for damages from airport noise. There are at present about 1000 such suits pending, and courts appear to be showing a tendency to broaden their interpretation of noise as a nuisance. This poses a danger to the financial situation of many jurisdictions. Other, neighboring local governments are, of course, often the plaintiff in such cases.

A large number of cities have anti-noise laws which apply to vehicle noise, but most of these simply require "adequate" mufflers, and/or restrict "unnecessary" hornblowing, or set subjective limits to noise, such as forbidding excessive or unreasonable noise. Where the limits are discretionary, such laws are sometimes struck down when challenged. Few cities can provide data on enforcement of these laws since there is generally no index to routine citations and no compilation of city court cases. It appears, however, that the rate of enforcement varies widely but is generally low. Where a high level of enforcement and effectiveness is claimed, e.g., in Memphis and Boulder, city officials attribute this success to educational campaigns which sensitize the public to vehicle noise and to encouraging police to give noise enforcement high priority.

A few cities have enacted vehicle noise laws with decibel limits. They report severe limitations on their effectiveness:

1) Cities have difficulties in formulating standards which are technologically feasible, yet strict enough to be effective. There are limited funds and a scarcity of technically trained personnel available to city officials, who frequently express the need for federal or state guidelines and assistance in setting standards and developing enforcement procedures.

2) There are again technical difficulties in separating and identifying individual noise sources on city streets with high ambient noise levels -- the same difficulty which prevents enforcement of state noise laws in cities.
3) There is a lack of personnel and equipment for systematic monitoring, and as noted, police have other competing priorities for their time.

Some local governments are experimenting with noise standards for new vehicles, but they face a particular difficulty in that a large fraction of vehicles using city streets are purchased elsewhere. Within most metropolitan areas there are many separate jurisdictions and many such areas straddle county and even state lines. New vehicle noise standards are probably ineffective without operating vehicle standards to ensure maintenance.

In Hawaii, and probably also in California and New York, local governments are preempted by the state from control of vehicle noise even though state laws are poorly enforced.

Mass transit facilities are a major source of noise in some large cities. These facilities usually represent a large capital investment in aging and deteriorating stock and equipment, and regardless of local noise laws, the cost of acoustical treatment precludes effective action to retrofit existing stock.

Most local noise laws are general nuisance laws which include prohibition on unnecessary noise from any source, including commercial and industrial sources and domestic or residential sources at the property line. Police attempt to enforce such laws on the basis of complaints and usually depend more on persuasion and warning than on official action. Most such laws specify "unnecessary" or "unreasonable" noise and thus rely on the discretion of the police officer; some are struck down on this basis. There is a model ordinance prohibiting unnecessary noise, prepared by the National Institute of Municipal Law Officers, which has been fairly widely adopted, but it also uses subjective criteria.

Only a few cities (perhaps on the order of a dozen) have general noise laws with decibel limits. These are subject to some of the same problems discussed above for vehicle noise laws, especially the lack of monitoring equipment and enforcement personnel. Inclusion of quantified noise standards in zoning laws is relatively recent. Formulating technically reasonable standards which are effective is again a serious problem. However, quantified standards are very useful for planning and zoning commissions who must screen applicants for industrial locations. Only a very few cities have added quantified noise standards to their building codes (to limit allowable transmission of noise into the building, between dwelling units in the building, and from building equipment such as plumbing and elevators). New York City now has such criteria applying to new buildings occupied this year.

Experience with control of construction noise is largely restricted to curfew laws, which are often relaxed on a plea of convenience, especially where daytime traffic interferes with construction. This is one of the greatest gaps in local noise control.

There is a very small but promising trend for cities to adopt comprehensive noise ordinances with specified decibel limits and to set up Offices of Noise Abatement to superintend enforcement. This arrangement has very pronounced advantages:

1) A directorate whose primary responsibility and priority is noise abatement;
2) Investigators specifically responsible for responding to complaints;
3) A staff with the initiative to seek out large-scale and persistent offenders and trained in the use of monitoring equipment;
4) A focal point for mounting a public education campaign.

The cost of such a program is not high relative to other kinds of pollution control but is nevertheless another strain on city budgets. Inglewood, Calif., pop. 90,000, spends about $60,000 annually on such a program. New York City and Chicago plan to use about 40 to 50 investigators each.

The National Environmental Policy Act now requires local governments as sponsors of federally aided projects such as airports and highways to prepare environmental impact statements which include an assessment of noise impacts of planned projects. While these treatments were, at least at first, usually cursory and judgmental, recently EPA in reviewing draft statements has increasingly pushed for more thorough and quantified projections of noise impacts. In addition to their preventive effect on new noise impacts, this action is likely to have significant educational effects in sensitizing local decisionmakers to the importance of noise.

To sum up:

- States are exhibiting increased technical proficiency in evolving noise control programs but are generally preempted from dealing effectively with aircraft noise, and are ineffective in controlling vehicle noise because of technical, legal, and financial constraints.
- City ordinances are in general still vague and technically deficient in controlling noise from all sources. Under increasing citizen pressure to abate noise, some cities are becoming more sophisticated in developing and using workable quantified standards, but are hampered by the difficulty in formulating feasible criteria, by budgetary constraints, especially in the context of more urgent urban problems, by technical difficulties in monitoring, and by administrative and personnel weaknesses in enforcement.
- Local governments are also faced with the problem that two of the most important sources of noise which afflict their residences, schools, churches, hospitals, outdoor facilities, and work environments are effectively beyond their control -- namely, aircraft and vehicle noise.
- Courts are becoming increasingly involved in the controversies over noise, but in general private suits for money damages have not accomplished a great deal of noise suppression. If this situation changes, local governments themselves may suffer serious financial losses.
- The most promising outlook for the future is for control at the source as the federal government sets noise standards for a wide range of noise-producing equipment; and for more rational and effective control at the operating level by local governments through the use of comprehensive noise ordinances with effective quantified limits, enforced by Municipal Offices of Noise Abatement.

REGULATION AND ENFORCEMENT OF URBAN NOISE

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The paper will present the implementation program of the Chicago noise ordinance. It will delve into the department organization reviewing engineering and enforcement activities in detail. It will include the specific needs of the engineering group to provide backup for the urban noise enforcement program. Statistics will be presented relating to the engineering activities which include field measurements and special research projects completed within the City.

The enforcement activities for both vehicle and stationary sources will be discussed and a review of the first year's activity will be presented. Included will be the types of complaints and numbers of citations, and the court record achieved by the department.

An assessment will be made of the actual ordinance and its ability to fill the needs of urban noise control. The advantages and disadvantages of the ordinance will be discussed and the proposed modification to the ordinance will be presented.

DEVELOPING A STATE-WIDE NOISE CONTROL PROGRAM--THE ILLINOIS EXPERIENCE

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In the State of Illinois, the Environmental Protection Act of 1970 set up a three-pronged attack against environmental damage including noise. The Act created the Illinois Pollution Control Board, the Illinois Environmental Protection Agency and the Illinois Institute for Environmental Quality. Each of these groups are separate from one another, but are designed to work in concert. The Illinois Pollution Control Board is a quasi-judicial body whose purpose is to consider enforcement cases as well as variance petitions and permit appeals. Further, they are charged with issuing orders and setting penalties after open public hearings where a written court transcript is developed using the rules of evidence, sworn testimony and the right of cross-examination. Also the Illinois Pollution Control Board is the body that adopts environmental quality standards and regulations after conducting open public hearings governed by the same parameters mentioned above.

Another important arm of environmental control in Illinois is the Institute for Environmental Quality. This body is the research arm in the partnership to reduce the burdens on our environment. The Institute is designed to cooperate in the development of interdisciplinary approaches to environmental problems as well as selecting environmental projects for study. Also, the Institute recommends long-range goals for technical, administrative and legislative changes and aids in the development and proposal of standards and regulations governing environmental problems as well as providing expert testimony in support of regulations. The Institute carries on the important activity of implementing the studies and programs necessary to support environmental activities and serves as a data bank for environmental information.

The third activity in the partnership is the Illinois Environmental Protection Agency. This Agency proposes, advises on, and responds to formal inquiries for standards controlling environmental pollutants. It also carries on activities leading to recommended action on variance requests from established standards. Importantly, the Agency conducts surveillance activities to detect violations of the law, standards or regulations and provides evidence adequate for successful prosecution of violations. Also, an important part of the Agency's activity is the issuance of permits for construction and operation of certain types of facilities capable of causing environmental problems. Many other activities are contained in the broad responsibilities of the Environmental Protection Agency, including such things as certifying facilities for tax purposes, administering anti-pollution bond acts for construction purposes and preparation of environmental quality management plans.

Also working with the Agency and the Board is the Illinois Attorney General's Office. When any violation of an environmental regulation is reported or discovered through normal surveillance, all data and evidence is assembled and compiled into a legal form by the attorneys of the Environmental Protection Agency located in each of the control divisions. After preparation, these cases are then referred to the Illinois Attorney General for filing before the Illinois Pollution Control Board for a decision. As discussed earlier, these decisions are based on firm proven judicial process, using rules of evidence, cross-examination, sworn testimony and the right of appeal. This program is designed to protect the rights of all parties as well as protecting the environment.

Contained in the Illinois Environmental Protection Agency is the Division of Noise Pollution Control. This independent Division contains a Standards Section designed to work in concert with the Institute for Environmental Quality and may also draw upon external sources for necessary information in designing logical regulations for the control of noise pollution. This Section is continually interested in the improvement of existing regulations as well as the development of new regulations as needed. The Surveillance Section contains the largest number of personnel and it is designed to collect the surveillance evidence adequate for prosecution of violations of the Board's rules and regulations. This Section has offices in various regions throughout the State to be more responsive to citizens' complaints. The Enforcement Services Section contains the Division's attorney whose function is to assure that enforcement cases are properly prepared and to advise on other legal matters. A Permit Section and a Variance Section are also authorized but will not come into full operation until the advent of certain regulations.

When the Illinois Environmental Protection Act became effective, it created Title VI entitled Noise. This was the State of Illinois' first excursion into a comprehensive program of controlling noise, and the introduction to Title VI states: "The General Assembly finds that excessive noise endangers physical and emotional health and well-being, interferes with legitimate business and recreational activities, increases construction costs, depresses property values, offends the senses, creates public nuisances, and in other respects reduces the quality of our environment." This, then, was the beginning of a state-wide noise control program for Illinois. In viewing the problem as it affects the entire State, it was decided to divide attention into four broad categories of control for which separate regulations would be developed. One of the categories decided upon was for the control of stationary sources, including all sources of noise sources within a fixed boundary or property line. Not only would this include such obvious sources as cooling towers and punch presses, but the sources operating within the confines of a marshalling yard such as trucks and railroad equipment as well. A second category for which separate regulations are necessary is the problem of airport noise. The noise from airports is an obvious and indisputable source of nuisance. A third grouping considered for control is the problem of ground transportation as experienced in the operation of trucks, buses, automobiles, motorcycles and off-road recreational vehicles. The last of the four categories concerns the problem of construction noise, and as yet little has been done in this area insofar as developing rules.

For the most part, with the exception of construction noise, the attention given to these broad categories has been going on simultaneously. However, because of the constraints of time and the requirement of public hearings before the Illinois Pollution Control Board, it is necessary to treat each of the categories separately. Naturally, thought was given to arraying the various categories in their order of importance. This becomes somewhat difficult and obviously some subjective thought must be included. If one were to look at the categories from the standpoint of the emotional impact as well as the available literature, one would probably develop a list beginning with airport noise followed by ground transportation noise, then stationary sources and finally, construction noise. However, using the criteria of pervasiveness as well as the record of citizens' complaints in Illinois, one finds a different picture with stationary sources heading the list and ground transportation in second order. The complaint percentages found in the files of the Illinois Environmental Protection Agency are roughly as follows: Stationary sources - 57%, ground transportation - 33%, airport related noise - 9% and construction noise - 1%. Few jurisdictions are as yet keeping statistical records on the types of noise complaints received, but the above percentages seem to be supported by the experience of several other jurisdictions I consulted.

The decision was made to present rules governing the control of stationary sources to the Pollution Control Board for public hearings and approval first. The initial hearings were held during the month of June in various locations throughout the State. Hearings before the Illinois Pollution Control Board relative to proposed new regulations are not window dressing. The Pollution Control Board, unlike many legislative law-making bodies, must by Statute consider many things in making its Orders and final determinations. The Board must consider the character and degree of the possible injury or interference with the protection of health, general welfare and physical property of the people; the social and economic value of the pollution source; the suitability or unsuitability of the pollution source to the area in which it is located, including the question of priority of location in the area involved; and the technical practicability and economic reasonableness of reducing or eliminating the emissions before the pollution source.

In preparing support for the stationary source regulations which were proposed by the Environmental Protection Agency and the Illinois Institute for Environmental Quality, it was necessary to consider all of the parameters I just mentioned and prepare testimony that could be substantiated in an adversary type hearing where both sides have the right of cross-examination. To support the proposed regulations which would have a far reaching effect on the entire State, it was necessary to draw upon sources external to the Environmental Protection Agency. Fortunately, in Illinois, we have as one arm of our environmental family the Illinois Institute for Environmental Quality. The Institute organized a Task Force group to study the problem of noise and selected for this group members of the University of Illinois' faculty on law, science and engineering. We also drew upon the expertise of the firm of Bolt, Beranek and Newman and recently Kamperman and Associates.

The first hearings into the proposed regulations on stationary sources were held in various locations throughout the State to prepare a stenographic court record for the Board's deliberation. As the hearings progressed through the month of June, it became obvious that further hearings would be necessary to more fully develop the support offered by the Agency and Task Force and to allow those persons opposing certain sections of the proposal time in which to further develop their presentation. This unfolding pattern is a familiar one in the development of regulations for the Illinois Environmental Protection Agency. Air and water pollution regulations have been similarly conducted. After the preparation of the stenographic record of the hearings and assessment of the many exhibits and written comment, a rewriting of the proposal was undertaken to improve those areas that examination proved were inadequate or unwieldy.

As was naturally expected, the areas receiving the most attention from both the supporters and detractors of the proposed stationary source regulations were the areas of the technical feasibility of meeting the proposed noise limits and the economic burden of doing so. In support of the limits, the Task Force was able to draw upon numerous studies and actual data collected by the Illinois Environmental Protection Agency as well as numerous other governmental and private bodies. Supporting the economic reasonableness of the proposed regulations posed many more problems in that published information was sparse, indeed. Little information could be generated from the NTID series of publications from the Federal Environmental Protection Agency in the area of economics. It was necessary, therefore, to obtain the type of information needed from individual companies and associations located in Illinois. Some information was also generated from corporations not located in the State. During the initial hearings, persons giving testimony before the Board were queried as to the availability of economic related information which their company or association might have available. Generally speaking, a lot of this type of data was volunteered to the Board. In some instances, however, it appeared that management decisions were made to withhold some of this economic data. The Agency then developed an interrogatory consisting of nearly 40 questions aimed at generating the specific type of financial information needed by the Board in order that they could have a complete record on which to make a decision. Using the Board's power of subpoena, these interrogatories were sent to a representative cross section of noise generating activity in the State. The subsequent information received generated valuable data concerning the economic burden of complying with a stationary source noise regulation.

In summary, then, the comprehensive program of noise control that is planned in the State of Illinois is just beginning to get a running start. Valuable experience was gained during the stationary source hearings and practically all major Illinois industry and other major operators of noise sources, based on their prior experience with the Illinois Environmental Protection Agency and the Illinois Pollution Control Board, began to take the problem of noise control seriously. The interplay between the operators of noise sources and the controlling agencies proved to be a valuable experience on both sides. The hearings further pointed out that stationary sources of noise are a very real and important source of noise and annoyance to the public in general. I submit that while transportation related sources of noise impact heavily upon our environment, we have been obsessed with air and ground traffic as a source of noise. This inordinate interest may have been generated by the available funding for transportation related noise studies almost to the exclusion of stationary sources; but the void is filling by the demand of the citizenry to act in this long neglected area. We look forward, in Illinois, to the day when a comprehensive program can be brought to bear on all of the various forms of noise which now trouble and burden our environment.

TRAFFIC NOISE LEGISLATION IN EUROPE

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Over the past ten years quite a few European countries set national regulations on the maximum permissible noise levels from motor vehicles. A detailed description of these measures would take far too long for inclusion in this short presentation. Those who may be interested might like to refer to the OECD publication : "Urban Traffic Noise - Strategy for an Improved Environment" which appeared in 1970.

It is more important, I think, that I should describe the international regulations adopted in Europe for motor vehicles.

In 1968 already, the United Nations Economic Commission for Europe drew up international norms for noise emissions but these were no more than recommendations and countries were free to adopt them or not, as they wished.

In 1970 the Common Market approved noise standards for new cars and commercial vehicles. These are of considerable importance since they became compulsory in the six Common Market countries with effect from 1972. The method of measuring noise that was adopted is that set out in the ISO Recommendation No. 362. The permissible levels are 83 dBA at 7.5 metres for private cars, 90 dBA for trucks and buses of less than 200 DIN HP and 92 dBA for trucks and buses with more powerful engines.

If these measurements were taken at 50 feet as in the United States, the figures would have to be reduced by 5 to 6 dBA.

PROPOSALS FOR LOWERING EXISTING NOISE NORMS

Although various European countries have in mind the gradual lowering of maximum permissible noise levels, no coherent plan has yet emerged, like that for air pollution caused by motor vehicles in the United States where a very detailed, stage-by-stage programme has been approved. One country however - the United Kingdom - has officially announced its intention to lower its maximum permissible noise levels in 1975/1976. The standards proposed are 80 dBA for cars (which is 3 dBA below the current Common Market standard), 86 dBA for commercial vehicles of less than 200 HP (4 dBA below the Common Market standard) and 89 dBA for more powerful commercial vehicles (3 dBA less than in the Common Market). When the United Kingdom officially enters the Common Market in 1973, it will be interesting to follow the discussion which will take place between the U.K. and the present members of the Common Market with regard to the progressive lowering of noise levels. Legally, however, the United Kingdom will initially be obliged to comply with the rules already adopted by the Common Market; in other words, it will have to allow the import of vehicles from other Common Market countries even though their standards may not be regarded by the United Kingdom as sufficiently strict.

However, we are not yet in a position to forecast what is going to happen in the next few years. No one can yet say whether the Common Market will follow the United Kingdom's lead or whether the United Kingdom will have to come into line with the Common Market. One thing is clear, and that is the avowed intention of the United Kingdom to reduce the level of noise from motor vehicles in the future, at least in its own territory. This is confirmed by the fact that the British Government is currently financing a "Quiet Truck" research project for the development of a diesel truck emitting not more than 80 dBA as measured by the ISO method. This is 12 dBA lower than the figure for existing trucks and the "Quiet Truck" would thus be quieter than a private car.

PROPOSALS FOR CHANGING THE NOISE MEASUREMENT PROCEDURE

On the question of how noise is measured, it is important to note that several European countries - and the United States too, I believe - have serious reservations with regard to the ISO method. This is important because the way in which noise is measured has a considerable influence on the standards which result. If the measurement technique is wrong it may give a false "sound picture" in practice.

I would therefore like to deal in some detail with the problems now being discussed under this heading in Europe since they are also of interest to the United States.

The first question that has to be answered is this: what are the objectives of a standardized procedure for measuring vehicle noise? In theory there should be three. I say "in theory" because two of them are contradictory. The first objective is to measure the maximum noise that a vehicle is capable of emitting during maximum acceleration, irrespective of the circumstances. The reason for this is that the nuisance caused to the public is mainly associated with noise peaks, particularly during night-time. The second objective is to measure the maximum noise of a vehicle in normal city traffic conditions. The vehicle's normal conditions of use are simulated and the ISO method as it stands satisfies this aim to some extent since it consists in measuring the maximum noise corresponding to normal city driving. The noise is measured during full-throttle acceleration in an intermediate gear, starting from an engine speed equivalent to three-quarters of that at which maximum power is developed; the vehicle's road speed is, however, limited to fifty kilometres an hour.

This second objective, unlike the first, presents a major disadvantage in that it fails to show up the nuisance caused by the noisiest vehicles (sports cars and motor cycles) and by aggressive driving.

The third objective is to provide the police and vehicle testing authorities with a simple and rapid method of checking, either on the road or during periodical inspections, that vehicles comply with the standards laid down.

As I have said, the present ISO procedure achieves only the second objective; in other words it enables normal city driving to be simulated. But what do we mean by "normal driving"? If say 95 per cent of drivers are considered to drive normally, what about those sports car drivers or motor-cyclists who indulge in fast acceleration, habitually exceed the speed limit, and generally drive with the deliberate intention of creating noise?

To arrive at a better notion of the nuisance caused, it would seem necessary to measure the maximum noise emitted during maximum acceleration and in low gear. But this would penalise not only sports cars and motor-cycles but also sedans with big engines, which would be unfair since such cars, and especially those with automatic transmission as is usual in the United States, are almost always much quieter in city traffic than cars with smaller engines.

Those who advocate measuring maximum noise are therefore clearly at variance with those who prefer to measure normal noise.

The discussions now going on in Europe on this subject seem to suggest a move towards a mid-way solution. The normal city driving speed would still be simulated, that is to say, 50 kilometres an hour, but the noise would be measured with the vehicle in second gear. This would clearly eliminate those vehicles which are really the most noisy but it might well also penalise vehicles which are not noisy, namely cars with automatic transmission and big engines.

The only answer to this problem would be to work out a noise emission cycle on the lines of the pollutant emission cycle. This approach is already under study in France. Research is in hand with a view to measuring the noise emitted by different types of vehicle, with different drivers, on various trips in built-up areas, so that a kind of standard urban route might be defined for reference purposes in establishing permitted noise levels.

Unfortunately, other than by merely adding together all the noises emitted - and the only practical method is to add together the sound energies (L_{eq}) and this has the serious disadvantage of attenuating the dispersion and fluctuations of noise peaks which are precisely the worst nuisance particularly when sleep is concerned - it is difficult to see what weighting should be given to each of the noises emitted during a standard trip. The aim must therefore be to arrive at a simplified procedure similar to that now used for aircraft certification. For example, one might take into account only the noise from the vehicle when stationary, during maximum acceleration and at cruising speed; the readings in each case would, of course, not have to exceed the permitted levels, which would be entered on the vehicle's registration papers for reference at the time of periodical tests or checks on the road. The maximum permitted noise from a stationary vehicle would, in particular, be very useful for rapid police checks. So far the only country that has introduced maximum permitted levels of noise from stationary vehicles is Switzerland. The Swiss believe this is the only rapid method for checking whether a vehicle in service is not making more noise than it was when it left the factory. Countries are thus faced with a twofold problem: they must lay down noise limits, and ensure compliance with these standards. Noise measurement procedures should therefore take into account:

1. Usual traffic conditions - although these vary greatly depending on the place and the time of day.

2. The nuisance caused to the public, especially at night; and

3. The need to carry out rapid checks on the road.

The present ISO methods of measurement do not meet these requirements in all respects and the work now in hand should therefore be aimed at satisfying all these three conditions simultaneously. International co-operation between all the countries concerned is the only way in which a speedy solution will be found; otherwise the vehicle manufacturers may well be slower to reduce their noise levels or even become discouraged altogether.

To conclude this outline of the legislation in Europe, I would like to mention the new approaches now under consideration, especially in France, with a view to achieving a rapid reduction of vehicle noise.

REGULATIONS OR TAXATION?

At the moment the only measure applied by governments to limit vehicle noise is to lay down specific standards which must not be exceeded. This is the regulatory approach as opposed to an incentive to produce less noise; it is a ban on the generation of more noise. But regulations have major drawbacks; in particular, they are based on the worst of present-day technology. By this I mean that the standards are generally laid down in the light of the technical characteristics of the vehicles now on the road, and they are therefore drawn up in such a way that nine-tenths, and sometimes even 100% of vehicles currently in production comply with them. The noise standards now in force are an incentive to perpetuate the status quo, in that they authorise present-day noise levels; they provide no incentive to reduce noise. It is patently obvious that regulations hold back innovations, since vehicle manufacturers see no reason to improve upon the standards. The answer to this, you will say, is to set target norms, that is, to tell vehicle manufacturers in advance that they will have to meet more strict standards by stages. But in this case the authorities have only two courses open to them: either they agree on these targets with the vehicle manufacturers - in which case they may possibly have to negotiate a compromise, or else they can take a gamble and set the targets without consulting the manufacturers. In this case there is the danger that the targets might not be reached and involve expenditure out of all proportion with the results achieved.

The best solution would be to back up the existing regulations with an economic incentive designed to encourage the automotive industry to reduce the noise from the vehicles it produces to lower and lower levels. The proposal is in no way an alternative. It is not suggested that the regulations should be replaced by a fiscal system since this would allow wealthy road-users to create noise, which is obviously unacceptable. No, the idea should be regulations plus taxation.

Standards would be laid down to prohibit the noisiest vehicles and the purpose of the tax system would be to bring noise down to the desirable level, below that set by the standards. For sake of simplicity, the tax would be levied at the time of sale of new vehicles, either by changing the existing tax base applying to vehicles so that the tax payable would depend on the noise produced, or by introducing a new "pollution" tax in addition to the existing tax - with all the political problems that might well arise with the introduction of such new indirect taxation. Clearly, many problems remain to be solved before a practical answer can be found. How, for instance, would the tax base be determined? How could tax revenue be forecast? How could one ensure that this revenue does not decrease rapidly, to the dismay of our finance ministers?

But the two major problems are these:

1. The novelty of the system and, it follows, the opposition of all kinds that will be encountered before a simple scheme could be put into effect.

2. The immense difficulty of deciding the relative proportions of tax to be allocated to noise and to air pollution, road safety and maybe other aspects.

Although all of these problems are substantial ones, it is not unreasonable to hope that, in the long term, economic incentives might be introduced to reduce motor vehicle noise. Applied in conjunction with minimum standards, they would help to attain the two essential objectives of noise abatement, the first being to achieve a gradual reduction of the levels of noise emitted and the second to reduce, at the same time, the dispersion in these noise levels since economic incentives would encourage the sale of the quietest vehicles and induce manufacturers of noisy vehicles to make a special effort.

To sum up, our aim must be to reduce the mean level of noise in man's environment and also to eliminate noise peaks which are the main cause of nuisance to the public.

NOISE LEGISLATION IN BRITAIN

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The law on noise in Britain is of three basic types. There has, for twelve years, been a Noise Abatement Act; there are statutory regulations relating to vehicle noise emission; there is a substantial body of case history and precedent concerning noise as a form of common law nuisance. In addition, there are a number of other statutes which contain provisions capable of interpretation with respect to noise, but they have never been invoked in this way, and there are regulations concerning sound insulation in buildings.

It is now generally realized, by both the public and the central government, that the law on noise is unacceptably weak. The Noise Abatement Act is particularly unsatisfactory, not so much in its basic nature, but in the provisions for its enforcement. Fundamentally what it does is to make noise a statutory nuisance, so that it can be dealt with in much the same way as other nuisances under the Public Health Act, 1936. It is operated by the local authority, and if a member of the public makes a complaint, an officer of the local authority, usually a Public Health Inspector, will go along to investigate. If he is satisfied of the existence of a nuisance, the authority will issue an Abatement Notice, requiring the person responsible for the nuisance to abate it and to carry out whatever work is necessary to do so. If the person on whom the Notice is Served does not comply with it the authority must apply to a Magistrates' Court for a Nuisance Order. If the Nuisance Order is confirmed and not complied with the recipient will be fined £50 and £5 per day while the nuisance continues.

This of course is one of the weaknesses. The fines are much too low, and many corporations and certainly many building contractors engaged on a site for a limited period can easily afford to ignore a Nuisance Notice. Another major weakness is the opportunity the Act offers for an offender to use delaying tactics. If substantial works are necessary to abate the nuisance, the initial notice may specify a protracted period in which they are to be carried out, perhaps three months. In that case the matter may well not come before the magistrate until up to five months after the initial complaint, and even if there is no appeal to the Crown Court, abatement is unlikely to be achieved in under eight months. If there is an appeal, it might well be not far short of a year.

Another weakness in the Act, which may in fact be unavoidable, is that it provides for the defence that the defendant has used the 'best practicable means to abate the noise'. One can of course argue that a law which required a person to take anything other than practicable means would be wholly unreasonable, but the Act concerned goes on to say that in determining whether the best practicable means have been used, regard shall be had to cost and to local conditions and circumstances. If the defendant succeeds in proving that he has used the best practicable means, which he often does, then there is no Statutory Nuisance and the court will refuse to make an abatement order.

It is for these reasons that at the moment the only really effective way of going to law over noise nuisance is to make an application in the High Court for an injunction, being an order of court restraining the committing or continuing of some wrongful act or omission. Failure to comply with it may lead to imprisonment; no less. The important point is that the defence of the best practicable means is not available in common law.

The United Kingdom Noise Advisory Council set up a Working Group on the Noise Abatement Act which published last year a report recommending ways of strengthening the Noise Abatement Act. The Secretary of State for the Environment, who chairs the Noise

Advisory Council, has declared it his intention to bring a new Noise Abatement Bill before Parliament, and as well as tightening up the appeal procedures and substantially raising the fines, the new Bill will contain a new concept of Noise Abatement Zones. An entirely new legal procedure is proposed with which to operate them.

The philosophy behind the new proposals is that of getting away from the nuisance approach to noise. They set about providing a framework for attacking the general noise level in an area by requiring measures to be taken without having to prove nuisance. Noise Abatement Zones are designed firstly to halt the creeping rise in ambient noise levels from industrial and commercial premises, and then to begin to secure a progressive lowering of the existing levels. A 'Target emission level' will be set which will constitute a standard for the background noise level in the Zone.

It is in the enforcement of the level that the radical new powers come in. Few industrialists will charitably design for what they see as an absurdly low criterion, far below what they have hitherto thought necessary. A new legislative framework will be required.

The new procedure is designed firstly to secure the replacement or modification of existing major structures or capital equipment which is contributing to the general noise level, over a relatively protracted period, and secondly to control the noise emission of new installations and constructions. In conjunction, it is hoped that the immunity at present enjoyed by the nationalized industries will be removed, subject to a proviso that nothing should conflict with a duty imposed upon them by law if they are using the best practicable means to reduce noise emission.

With the aid of a new-style Noise Abatement Notice or Order, it will be possible for the local authority after compiling a blacklist, or 'loudlist' of local industrial premises, to serve upon them a notice requiring them to carry out major modifications over a specified, but economically reasonable, time in order to meet the target level, or upon the planned reconstruction of a plant to incorporate certain major noise control measures to meet the target.

There remains one unsolved problem. If a target emission level is set as much as ten decibels below the prevailing ambient noise level, how do you tell whether a Noise Abatement Notice has been complied with? The answer is that you cannot, at least not until the effect of the NAZ scheme has made itself apparent and the ambient drops. Is not the whole scheme then open to widespread abuse?

When a new installation is planned, or a modification designed, the industrialist has to rely on the expertise of acoustical engineers to predict what noise levels will result at the boundary of the factory, and to design appropriate noise control measures to achieve the specified result. The measures required of the factory owner will either have been laid down or approved by the local authority, and any obvious attempt to try to get away with less than adequate measures will be readily apparent. Should, however, the noise emission from an installation prove not to meet the target, but nevertheless be low enough to be incapable of measurement, then the non-compliance with the abatement notice will not become apparent for several years. Should this happen, when the ambient noise level does come down to a level that exposes it, the normal remedies for non-compliance with the order will still apply.

Clearly great emphasis will be placed on the ability of engineers to know what noise levels will result from their installations. One of the present difficulties in this respect is a lack of information regarding the sound power output of machines. Where similar machines are in fairly widespread use, this is not too much of a problem, but in many cases no comparative data can be obtained. Consequently it is important that legislation should require the Sound Power Level of specified classes of machinery to be declared.

The problem with construction noise is very different, of course, and long term measures to control it are useless. With this in mind, it is proposed that local authorities should lay down the maximum levels of sound at the perimeter of the site which are to be permitted, at the time of tendering.

It is probable, however, that the most severe noise problem in any industrialized country is not in fact industry at all, but road traffic. Certainly in the United Kingdom, nearly 20 million people have L_{10} levels outside their homes of more than 65 dBA caused by traffic. The Secretary of State for the Environment has said that no longer as an act of conscious public policy will people be subjected to an L_{10} of more than 70 dBA without compensation or remedial action.

Motorway noise, though, is more of a planning matter than a legislative one. It is the noise of traffic in streets and residential roads which is most susceptible to control by statutes. Here again, Britain has a very weak law. Although it was the first country to introduce specific levels of noise from vehicles in the form of regulations, the regulations are only really effective in respect of new vehicles.

The vehicle regulations are weak because the circumstances in which the noise of a suspected offending vehicle can be measured are so strictly laid down as to make them impractical in many cases. A space of 50 metres radius of which the central area of 10 metres radius consists of concrete, asphalt or similar hard material free of any soft covering such as snow, long grass, loose soil or ashes is required. There have to be no substantial obstructions within 25 m of the vehicle and '...... the sum of the angles subtended at the position of the test vehicle by surrounding buildings within 50 metre radius shall not exceed 90°....' Small wonder that the police are reluctant to enforce the regulations.

Once again, a Working Group of the Noise Advisory Council have looked at the question of the enforcement of vehicle noise regulations, and two ideas emerge. The first is that it appears to be feasible to use a static noise test for diesel powered vehicles, either as part of its annual roadworthiness check or in road side spot checks. Much less emphasis need be laid on the surrounding environment than in the existing regulations. In addition, it ought to be possible to pick out passing vehicles at the roadside with the use of a small check meter so that it can be sent to a testing station for an accurate noise test. This requires the development of a suitably accurate, small and cheap measuring device for the police to make widespread use of it.

In conclusion, it can be said that the law on noise in Britain is in a state of flux; most of the statutory legislation is inadequate and soon to be strengthened. The most notable gap in the spectrum is the total lack of any laws in the U.K. relating to hearing damage. We now have a code of practice on the subject, and it will be followed at some date by a similar law. For the present this is a field in which common law is the only way in which cases reach the courts.

Nevertheless, merely because the existing laws are weak, one cannot say that noise control is rarely achieved in Britain. Over 90% of the noise nuisance cases that come to the notice of local authorities are dealt with informally, and there are still tens of thousands of prosecutions of drivers under the old law simply because they had ineffective silencers.

THE U.S. ENVIRONMENTAL PROTECTION AGENCY'S ROLE IN A NATIONWIDE NOISE ABATEMENT PROGRAM

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An overview will be given of the current and acticipated future roles of the Environmental Protection Agency, in cooperation with other Federal agencies and with State and local governments, to control and abate environmental noise. As background, a brief review will first be given of (1) the current extent and severity of the "noise problem," based on studies performed as input to EPA's "Report to the President and Congress on Noise;" (2) the existing major Federal regulatory or administrative programs in noise abatement; and (3) the current status of State and local government noise programs as assessed by EPA's questionnaire survey. Finally, the EPA's present role will be described, including activities derived from the National Environmental Policy Act of 1969 and from Title IV of the Clean Air Act Amendments of 1970. The EPA's probable future role (based on legislation pending at this date) will be outlined and placed in context with a suggested nationwide noise abatement program. The desirability of continued and increased noise related activities (complementing EPA's role) in other Federal agencies and at State and local levels will be emphasized.

APPROACH TO THE PROBLEM OF NOISE CONTROL IN THE SOVIET UNION

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Continuous improvement of work conditions in all branches of the national economy has always been one of the main goals of the Soviet State and trade unions. In the twenties-thirties,six research institutes dealing with the problems of safety in different industries were set up in our country. At first the problems of safety engineering, ventilation, accidents caused by electricity, and protective clothing were of prime concern, but already in the prewar years one more hazard became evident, that of intensive industrial noise.

The work on noise abatement was initiated by the Leningrad Labour Protection Institute whose representatives are the authors of the present communication. In 1939-40 draft sanitary norms for noise reduction were prepared and a simple meter was designed (G. L. Navyazhsky). The war of 1941-45 interrupted this work, and we were able to resume it only in the early fifties.

Numerous examinations of the hearing of thousands of workers with the length of their service and physical characteristics of noise taken into account made it possible to establish norms and regulations on noise reduction in industry (I. I. Slavin), the first ever elaborated in the world practice. Those norms were approved by the USSR Ministry of Public Health in 1956 (N° 205-56). The specific feature of those norms was the fact that the admissible level of sound pressure was determined taking into consideration its frequency spectrum. Thus, for the noise with the peak frequency, for instance, 60 Hz the level of 100 dB was taken as admissible, and for the wide-band noise it was taken equal to 75 dB. In spite of some drawbacks, those norms were the first step in solving the problem of noise control in the country; thereafter, sanitary inspection bodies insisted on including engineering schemes aimed at the reduction of noise in industrial areas into the project reports of new industrial enterprises prior to their approval.

The next important step was a special decree called "Measures for Limiting Noise in Industry" passed by the USSR Council of Ministers in 1960 which committed the Ministries responsible for the production and operation of equipment to promote work on reducing its noise. Noise control laboratories were set up in every branch of industry. According to the Decree, the project reports of all new industrial enterprises had to include a section providing for noise control measures. Some organizations were instructed to start the development of acoustic measuring devices.

It became obvious that to radically solve the problem of industrial noise, equipment with noiseless or reduced noise characteristics had to be designed and produced. It was decided to begin with compulsary determination of the sound power frequency spectra of machine-produced noise by the manufacturing plants. A corresponding standard "Machines. Noise characteristics and methods of their measurement" was approved in 1966 to be put into effect starting from January 1, 1968.

The increasing scope of noise-control activities necessitated co-ordinating scientific and engineering efforts in this field on the national scale. In October 1970, by the Decree of the All-Union Central Trade Union Council and the State Committee on science and technique at the USSR Council of Ministers a Co-ordination Program for handling scientific and engineering problems in 1971-1975 was approved and a prospective assignment for the research and design organizations was formulated as follows: "0.69.232. Develop and introduce effective methods and means of noise and vibrations control in various branches of national economy" with particular emphasis on three trends to be followed in research:

- reducing noise in production areas by construction and acoustic methods;
- methods of establishing technical norms for machine-generated noise;
- fundamental methods of noise elimination in its source for the principal types of technological equipment and tools.

The last trend is the most important one, as the use of the results of this work will enable the designers of new technological equipment and processes to start manufacturing machines with reduced noise characteristics. Our Institute is in charge of the work in this direction.

CORRELATIONS BETWEEN DIFFERENT COMMUNITY NOISE MEASURES

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Introduction

Within the last several years there has been increased interest in the measurement of community noise, with new methods proposed for estimating community noise levels and for interpreting community noise measurements in terms of impact on people and communities. In distinct contrast to the direction of study in early post-war years, interest has shifted from detailed frequency spectral information to study of the statistical variation of noise levels with time. Thus, many recent studies rely primarily upon A-level measurements (perhaps supplemented with limited spectral analyses) but devote greater attention to the statistical properties of the noise signal. These statistical analyses are based upon study of noise samples taken over time periods ranging from a few minutes to a number of hours. Proceeding from rather simple descriptions of the distribution of noise levels in terms of various percentiles, several more complex single number measures, such as the traffic noise index (TNI) and the noise pollution level (NPL), have been proposed and are being utilized. Most of these newer measures are derivable from knowledge of the cumulative distribution of noise levels observed over a period of time.

This current interest in measuring the distribution of levels occurring within a given time period does deserve one word of caution. Analyses confined to noise level distributions necessarily omit any consideration of the time sequence in which the noise levels occur. This loss of time pattern information may be of little consequence in comparing noise samples dominated by similar noise sources. However, the time sequence of noise levels may be very important in determining human response to noise. And, similarities in statistical distributions of levels do not necessarily assure similarities in time patterns or sources of noise. Thus, in developing useful noise scales for correlation with individual or community response, time pattern information may well have to be considered, together with noise level distribution information.

The major purpose of this paper is to explore some of the relationships and correlations existing between common statistical measures extracted from a number of noise samples collected over the last several years. The noise samples studied were all of approximate 10 minute duration and were taken during day, evening and night periods in urban and suburban areas in five cities (Boston, Detroit, Los Angeles, Portland, Oregon and Honolulu). Seven-hundred sixty-three samples were recorded at 41 different locations. In recording, the microphone was placed five feet above ground, 35 feet from the nearest roadway whenever possible and at least 10 feet from any large objects or surfaces. The noise recordings were all taken during dry weather at temperatures ranging from approximately 30° to 90°F.

In analysis, the recorded noise signals were played back through the A-weighting network of a sound level meter into a statistical distribution analyzer. The statistical distribution information was then used to calculate several percentiles including: L1, the noise level exceeded one percent of the time; L10, the level exceeded 10 percent of the time; L50, the

CORRECTION!

THE PREVIOUS DOCUMENT(S) MAY HAVE BEEN FILMED INCORRECTLY . . .

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COMMUNITY NOISE

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In analysis, the recorded noise signals were played back through the A-weighting network of a sound level meter into a statistical distribution analyzer. The statistical distribution information was then used to calculate several percentiles including: L1, the noise level exceeded one percent of the time; L10, the level exceeded 10 percent of the time; L50, the level exceeded 50 percent of the time; and L90, the level exceeded 90 percent of the time. The noise pollution level, NPL, and the L-equivalent level, Le, were also calculated.* (Le is the continuous A-level that is equivalent, in terms of noise energy content, to the fluctuating noise level observed over the sample period.)

The noise samples were collected during a motor vehicle noise investigation[1] and during several highway traffic noise study programs.[2,3] The measurement locations encompassed a relatively broad variety of urban and suburban situations with respect to motor vehicle traffic. The localities may be categorized grossly with respect to traffic exposure as follows:

Near light traffic -- typically 8 vehicles or less per minute**
Near heavy traffic -- more than 8 vehicles per minute**
Near limited access highways (freeways).

Twenty-two locations were exposed to light traffic, 16 to heavy traffic, and three to freeway noise. Thus, the number of freeway localities sampled is much less than the number of heavy or light traffic situations.

All of the measurement locations were in residential areas or near buildings where noise might be of concern -- schools, hospitals, and two jails. One significant limitation is that few of the locations were exposed to major noise sources other than surface traffic. For example (and perhaps one of the more important omissions), none of the measurement locations were near major aircraft takeoff or landing paths. Therefore, few of the samples include frequent or high level aircraft noise intrusions. Similarly, few of the samples reflect any large influence from industrial noise sources.

Range of Observed Noise Levels

Before examining the correlation between different noise level statistics or measures, it is useful to first look at the range of noise levels encompassed by the noise samples. Some indications of the range of noise levels are given in Figures 1 and 2. Figure 1 shows the range of the L10 and L50 noise levels for 80 percent of the noise samples in each of the three traffic categories. Ranges of data are shown separately for day (7 am to 7 pm) and for night (10 pm to 7 am). One will note the wider range of noise samples in the heavy and light traffic categories. Including all categories, the noise levels (L10 or L50) span a range of 30 dB or more.

Figure 2 shows the cumulative distribution of day and night Le noise measurements for the three categories. In Figure 2, the range between day

*For a sample consisting of a sequence of N noise levels measured at small, uniform time intervals,

$$Le = 10 \log \frac{1}{N} \sum_{1}^{N} 10^{L_i}$$

and

$$NPL = Le + 2.56s$$

where

$$s = \left[\frac{1}{N-1} \sum_{1}^{N} \left(L_i - \frac{\sum_{1}^{N} L_i}{N}\right)^2\right]^{1/2}$$

**During peak daytime traffic flow.

and night distribution curves are shaded. The cumulative distributions are plotted on probability paper. Thus, if the samples followed a normal gaussian distribution, the cumulative distribution curve would form a straight line. The slopes of the curve also indicate the relative width of the noise level distributions. Comparison of the slopes of the distribution curves shows that the range of levels observed at night is typically broader than the range observed in daytime. It is also evident that the spread between day and night levels is greater for the light and heavy traffic samples than for the freeway samples.

Despite some overlapping of noise levels among categories, the median noise values for each of the three categories show quite distinct differences. This is illustrated in Figure 3 which shows the median noise level values for day, evening and night periods for the three traffic categories. There is a distinct trend toward higher noise levels with increasing traffic, accompanied by smaller differences between day and evening levels for heavy traffic and freeway situations. For example, the Le values for daytime increase by approximately 8 dB from light to heavy traffic and show another 5 dB increase for the freeway situation. There is an approximate 9 dB difference in Le levels between day and night for the light and heavy traffic positions, but only a 4 dB difference for the freeway situation.

One can note that, except for the daytime heavy traffic situation, NPL values are very close to the L1 levels. Similarly, the Le levels fall 2 to 3 dB lower than the L10 levels.

Correlations Between Noise Measures

It is not always feasible to make detailed statistical analyses of a noise sample. And sometimes, one must estimate noise levels from rather fragmentary information. Thus, there is interest in knowing the typical correlations that may exist between different noise measures.

To provide some of the correlation information, regression lines and correlation coefficients were computed for the correlation of several noise measures (NPL, L1, L10, and L90) with L50 and Le data. Regression lines were calculated for data in each of the three categories, for day, evening and night periods separately and combined. Figure 4 shows the correlation of various noise measures vs L50 for the heavy traffic and the freeway traffic data. The regression lines for the light traffic situation were quite similar to those for the heavy traffic and, hence, are not shown.

The regression lines for the heavy traffic data are quite different from those fitted to the freeway data. In particular, one may note the contraction between L1 and L90 lines with increasing level for the freeway data. For heavy traffic situations, the L1 and L90 lines are nearly parallel.

Correlation coefficients for the various regression lines are noted in Figure 4. There is particularly high correlation (correlation coefficients in excess of 0.9) for L10, Le and L90 vs L50, and relatively poor correlation of NPL and L1 levels with L50 values. Correlations of the noise measures with Le values show differences compared to the correlations with L50. There is very high correlation of L1, L10, L50 and NPL values with Le for both light and heavy traffic situations. For the freeway samples, Le shows high correlation with L10, L50 and L90, but not with NPL.

Detailed examination of regression lines for day and evening time periods for the different traffic categories show little differences for the limited freeway data but shows occasional significant differences for the heavy traffic and light traffic data. To illustrate the type and degree of differences, Figure 5 shows separate regression lines fitted to day and night data for the heavy traffic locations. There are small but distinct differences between the correlations of NPL and L1 values with L50 between day and evening situations, for example. The divergence between day and night regression curves tends to increase with level. Thus, at low or moderate noise exposure levels, below about an L50 value of 50 dB, differences between day and night measurements may usually be neglected. For higher noise levels, one may wish to take into account the difference between day and night regression lines.

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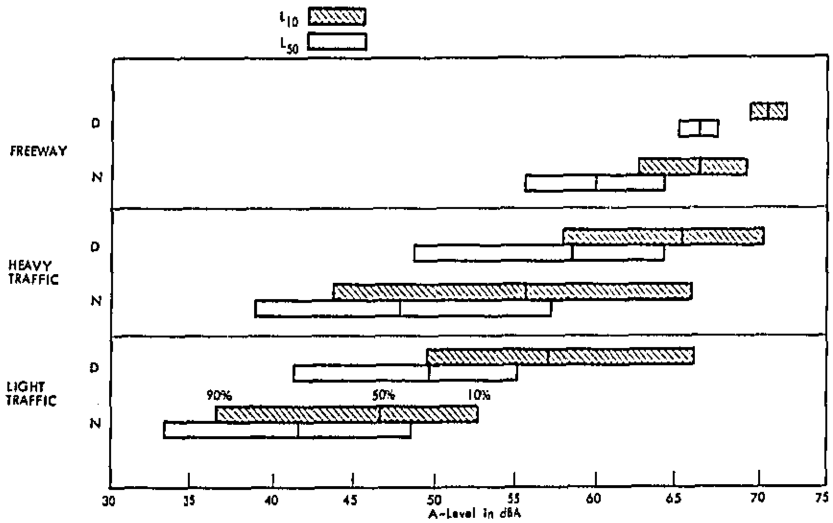

FIGURE 1. RANGE OF L_{10} AND L_{50} A-LEVEL MEASUREMENTS

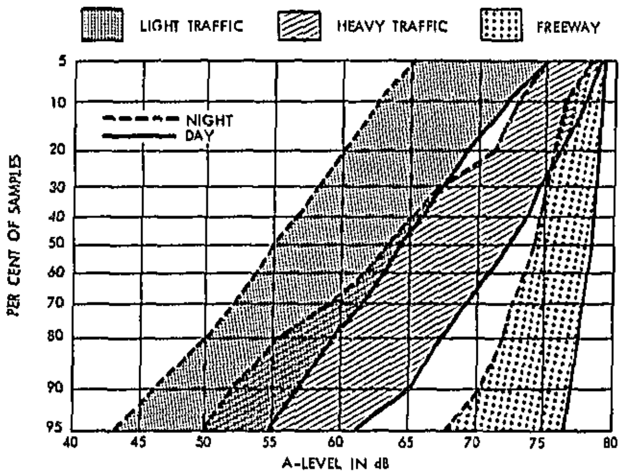

FIGURE 2. CUMULATIVE DISTRIBUTION OF L_e VALUES AMONG SAMPLES

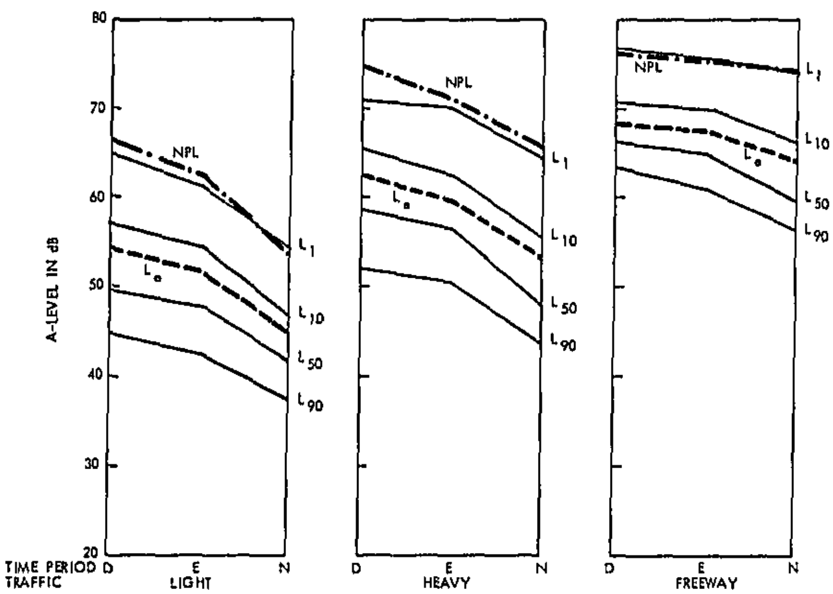

FIGURE 3. MEDIAN NOISE LEVELS FOR DIFFERENT TRAFFIC EXPOSURE

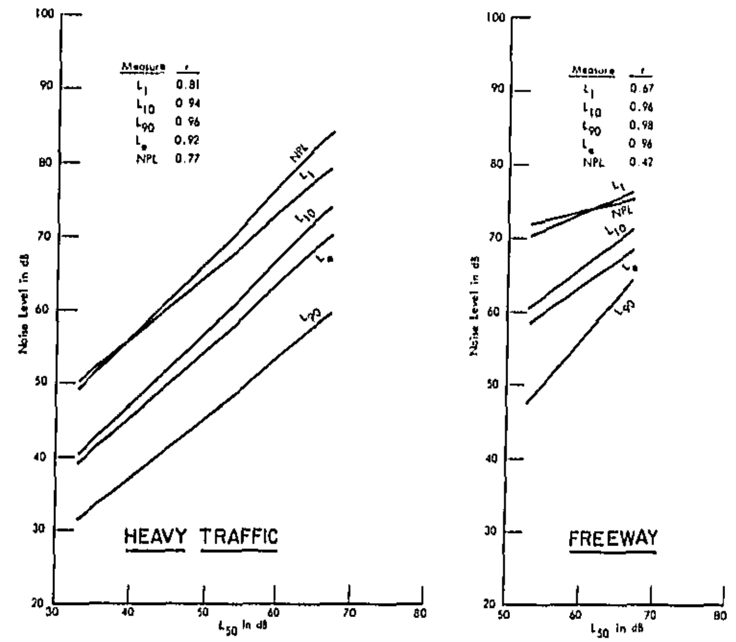

FIGURE 4. CORRELATION BETWEEN A-LEVEL NOISE MEASURES AT HEAVY TRAFFIC AND FREEWAY LOCATIONS

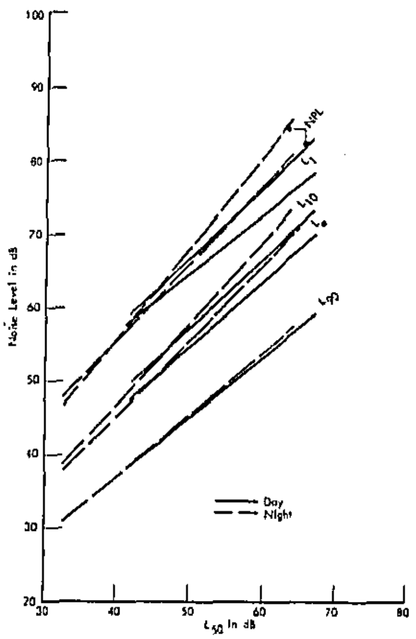

FIGURE 5 CORRELATION BETWEEN A-LEVEL NOISE MEASURES FOR DAY AND NIGHT MEASUREMENTS AT HEAVY TRAFFIC LOCATIONS

GUIDELINES FOR THE PREPARATION OF A MODEL NOISE ORDINANCE

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BACKGROUND

Beginning in 1967 the Sectional Committee on Bioacoustics of the American National Standards Institute established a Working Group on Measurement and Evaluation of Community Noise. This new Committee, referred to as S3-W-50, was established to help formulate United States comments on documents submitted by ISO (International Organization for Standardization), to consider ISO recommendations for adoption as American Standards and to draft whatever American Standards were necessary for dealing with problems of outdoor community noise.[1] Mr. James H. Botsford, Senior Noise Control Engineer, Bethlehem Steel Corporation, was originally appointed Chairman.

By design, members were selected from a variety of professional backgrounds (i.e. public health, sociology, industrial hygiene, psychology, city planning, physics and mechanical engineering),representing different institutional perspectives (i.e. government,citizen, university, big business and consulting). Beside the members appointed by the Chairman additional individuals have been invited to attend particular meetings representing certain city, state and federal agencies.

This Committee after defining its basic goal, to achieve maximum human privacy from intrusion by noise, established three projects to help attain this goal. These projects, which are at various stages of completion are

1. drafting a test-site measurement procedure of noise emitted by engine-powered equipment, both at the operator's position and at a distance of 50 feet

2. drafting a community noise measurement document containing a sampling procedure to make possible a reliable and repeatable assessment of community noise conditions

3. drafting guidelines for the preparation of a model noise ordinance whose provisions are to protect inhabitants from any noise interfering with human activity

NEED

The need for a document of this type is becoming increasingly recognized for a variety of reasons. First is the need for environmental protection, since noise constitutes a problem impinging upon the health and well-being of the population. However, in the U. S. there are fewer than 100 municipal noise ordinances representing a 1970 total population of just 40 million.[2] Secondly, the utility of these ordinances is questionable. They vary considerably, with less than a dozen containing acoustical criteria or standards. The remaining ones are generally unenforceable. Lastly, the provisions of many noise ordinances often are not current, and/or they contain requirements that are pre-empted by state or federal jurisdictions. The strategy, therefore, was to prepare a document that would address itself to these needs, among others.

GENERAL PROVISIONS

Several basic assumptions were made by the Committee after considerable deliberation. It was decided that this guideline should be an information document, not a standard. This procedure could expedite its preparation and ultimate distribution. Also, the provisions contained therein should be general in nature so as to have the widest possible application to the governmental bodies (i.e. city, county and state). No attempt was made to establish one specific set of acoustical criteria, since these will vary according to the ambient level conditions of each area. It was further agreed by the Committee to offer a guideline that contained enough essential ingredients to enable a governing body to understand the extent of the general community noise problem, acoustical instrumentation, acoustical measurement and criteria, administration requirements, enforcement and implementation. A guideline any smaller in scope would be professionally incomplete, compromising the Committee's goal.

SPECIFIC PROVISIONS

The specific provisions of this ordinance guideline are given in outline form with brief explanations since the committee document is not in final form at the present time. Underway are a series of community noise investigations designed to strengthen the acoustical instrumentation and measurement procedures outlined in this guideline. As the current Chairman of this working group, I anticipate that this document will be completed by early 1973.

A. Noise Impact
   1. Hazardous effects
   2. Nuisance effects

A survey of general community noise investigations pertaining to nuisance and hazardous effects will be provided as background information to assist governmental bodies in understanding the scope of the problem as well as providing authoritative support to a proposed ordinance or ammendment. Environmental education is an essential ingredient in the effective passage of any proposed noise regulation.

B. Acoustical Instrumentation
   1. Intermittent sampling
   2. Continuous sampling
   3. Instrument standards
   4. Calibration
   5. Maintenance

The type and use of acoustical instrumentation depends upon the sampling desired. A thorough discussion will be contained including existing instrument standards (ANSI, ISO, IEC) and calibration and maintenance requirements.

C. Acoustical Measurement
   1. Sampling procedures
   2. Mobile sources
   3. Stationary sources
   4. Data reduction

A variety of sampling procedures are available, but their statistical accuracy varies considerably. A table of sampling methods and their accuracy as a descriptor of community noise is planned. Techniques for mobile and stationary source analysis will be included along with methods for reducing community noise data.

D. Acoustical Criteria
   1. Federal and state standards/pre-emption
      (a) mobile sources
      (b) stationary sources
   2. Zoning ordinance
   3. Subdivision regulation
   4. Site design

Based primarily on existing information acoustical criteria promulgated at all governmental levels will be summarized, including pre-emptive areas. Elements of zoning, subdivision regulations as well as site design procedures are reviewed with recommendations provided.

E. Enforcement
   1. Penalties
   2. Administrative procedures
   3. Legal certification
   4. Review process

A summary of enforcement techniques is to be included based primarily on the experiences of current environmental noise programs. Effective enforcement techniques are essential for the successful implementation of any ordinance.

F. Variances/Permits
   1. Permit requirements
   2. Fees
   3. Variance procedure

The need for variances, primarily for emergency activities, is recognized along with a permit system for certain intermittent activities. Again drawing from the experiences of current programs and Committee member expertise, a series of guidelines is suggested.

CONCLUSION/SUMMARY

This Committee in an effort to achieve its stated goal is preparing an information document referred to as guideline for the preparation of a model noise ordinance. There is a major need for the professional acoustical community to provide support in this neglected area. The work of this Committee has generated considerable interest within both the public and private sectors of society. It is our intention to collaborate with the Office of Noise Abatement and Control of the Environmental Protection Agency, along with other Federal noise activities in the preparation of a meaningful guideline.

[1] James H. Botsford, "Proposed American Standard For Community Noise", Presented at the 138th Meeting of the American Academy for the Advancement of Science, Philadelphia, Pennsylvania, December 29, 1971.

[2] Clifford R. Bragdon, "Community Noise Management: A Social Evaluation", Presented at the 138th Meeting of the American Academy for the Advancement of Science, Philadelphia, Pennsylvania, December 29, 1971.

A COMMUNITY NOISE PROBLEM/RESOLUTION

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INTRODUCTION

In the fall of 1969, Environmental Noise Consultants, Inc., was asked to make a noise survey in a small community in Pickens, South Carolina. An initial investigation showed a community layout and source of noise as illustrated in Fig. 1. Plant noise resulted from construction of coal cars utilizing the renovated facilities of an old railroad depot. The construction process involved hot riveting under an open shed, metal grinding and welding, periodic use of a diesel train, and constant operation of an air compressor.

The dwellings along Cedar Rock Drive were of frame construction, rather old but well kept and in good state of repair. Only one or two of the dwellings were air conditioned. The occupants were typically middle aged to older married couples with several elderly women living alone. A group of the residents in the immediate vicinity of the plant joined together in hiring an attorney to seek relief from the noise through legal action. An initial noise survey was conducted for use in making an appeal to the courts.

INITIAL NOISE SURVEY

The initial survey consisted of determining the ambient noise level in the community and the levels produced inside and outside of the dwellings when the plant was in operation. The outside ambient noise level was found to be about 40 dB(A). Measurements were made inside with windows and doors closed and with them open. In each case the outside measurement location was on the sidewalk directly in front of each house. The data obtained is summarized in Table 1. The attenuation provided by closing windows and doors ranges from 10 to 19 dB(A). The outside data cannot be compared directly to the inside data due to measurement location. The highest

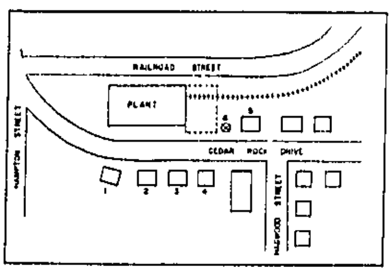

FIG. 1 COMMUNITY LAYOUT

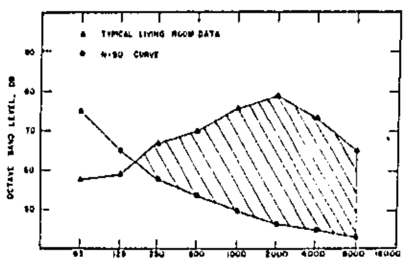

FIG 2 COMPARISON OF TYPICAL INTERIOR NOISE SPECTRUM WITH N-50 NOISE RATING CURVE

inside levels were obtained at locations (3) and (5), Fig. 1, which were in closest proximity to the open shed riveting area. The inside dB(A) levels are considerably above the levels normally considered to be reasonable for interiors of houses to allow for normal conversational speech, the use of the telephone, the enjoyment of radio or television, and restful sleep.

TABLE 1. AVERAGE NOISE LEVELS

Relative Measurement Location	Inside				Outside	
	Doors and Windows Closed		Doors and Windows Opened			
	dB(A)	dB	dB(A)	dB	dB(A)	dB
1	52	64	66	71	78	81
2	54	63	70	73	88	89
3	60	-	71	72	88	88
4	-	-	-	-	83	84
5	60	68	79	80	76	79

It is noted that for dwelling number 5, the inside levels are slightly greater than the sidewalk levels. The inside measurements were made in a sun room that faced the plant and had a lot of window space. Also, the sidewalk location was further removed from the noise source.

A typical inside spectrum is compared to the N-50 curve in Fig. 2. This figure dramatically illustrates the severity of the noise problem. It is noted also that the sidewalk levels for dwellings (2) and (3) are sufficiently high to produce noise induced hearing loss if exposure were sufficiently long and regular.

The following conclusions were obtained from the initial noise survey:

(1) Normal conversational speech was not possible inside of the dwellings and in the yards during normal plant operation;

(2) Based on several techniques for predicting community response to noise, the measured noise environment would normally cause a community to initiate legal action;

(3) Based on commonly accepted damage risk criteria, the inside noise levels would probably not result in permanent noise induced hearing loss.

A report prepared following the initial survey led to a favorable ruling for the plaintiffs handed down by the Honorable Frank Eppes, Resident Judge, Thirteenth Judicial Circuit, Greenville, South Carolina. The ruling requested that ENC, Inc., "make a study of the manufacturing facility of Golden Tye and report back to this court, with recommendations as to what measures, materials, devices, operational changes or location might be made, used, devised or employed to end or mitigate the noise which emanates from such facility toward the residence of the Plaintiff ...".

DETAILED NOISE SURVEY AND ANALYSIS

Almost one year elapsed between the times of the initial noise survey and the court ordered detailed study and analysis. The follow-up detail study involved:

(1) A general survey in the manufacturing facility and identification of the machinery and operations that represented the major sources of noise;

(2) A series of measurements to determine the contribution of particular noise sources to measured levels at specific neighborhood locations and an analysis of the abatement potential of each source;

(3) A neighborhood noise survey coupled with a community noise attitude survey for correlation of subjective reaction to noise with objective noise data in order to define what might constitute noise exposure acceptability.

An examination of the construction facility showed that the noise resulted primarily from (1) operation of diesel engine powered air compressors, (2) pneumatically powered metal grinders, (3) forced air gas furnaces for heating rivets, (4) riveting guns and structural vibration induced by the riveting operations. Fig. 3 illustrates the spectrums obtained in the near-field for each source.

To identify the relative contribution of each source to community noise, measurements were made at several locations under the following conditions:

(1) All machinery off and operations halted to obtain the ambient noise levels;

(2) Air compressor operating and all other machinery off;

(3) Air compressor operating and ordinary metal grinding in process with no other machinery operating or operations in progress;

(4) Air compressor and rivet furnaces operating only;

(5) Air compressor, furnaces, and riveting guns in operation simultaneously.

Fig. 4 shows the results of these tests for measurement location 6 which was in the yard of dwelling 5 towards the plant, Fig. 1. With no observable street traffic, the background noise was about 41 dB(A). The level increased to 56 dB(A) when the air compressor was started. The increase in noise level, 65 dB(A) and 67 dB(A) respectively. The noise level increased to 87 dB(A) when riveting was in progress. Plant noise thus contributed an increase on the order of 46 dB(A) in the side-yard of dwelling 5.

COMMUNITY NOISE AND ATTITUDE SURVEY

Noise measurements were made at dwellings along Cedar Rock, Hagwood, Railroad and Hampton Streets simultaneously with the conduct of an attitude survey. When residents were found to be at home, they were told that an attitude survey was being conducted and were asked to rate the noise around their homes as being in one of four categories: quiet, acceptable, noisy, or excessively noisy. The noise measurements were made in the front yard of the homes, typically midway between the sidewalk and the steps.

The housing layout in the vicinity of the plant is shown in Fig. 1. The reduced data from the attitude survey is based on a total of seventeen responses in the neighborhood. The average data of the subjective response versus noise level, dB(A), is shown in Fig. 5. Noise levels in the range from 50 to 55 dB(A) were considered typically to be quiet to acceptable while noise levels from 65 to 70 dB(A) were considered noisy to excessively noisy. One data point did not follow this trend - giving an acceptable rating for a noise level of 76 dB(A). This data point was based on two responses on Railroad Street which were business establishments.

NOISE CONTROL ANALYSIS

In order to establish the amount of attenuation that would be required, it was necessary to specify an acceptable inside noise level. A noise rating number N50 was selected based on (1) the attitude survey, (2) presence of noise continuously during the working day, (3) makeup of community near the industry, (4) presence of noise only during working hours, and (5) prior community exposure to noise. Fig. 2 shows the attenuation required based on N50.

Three alternatives were considered to achieve the necessary noise reduction:

(1) Reduce the noise of each contributing source to within acceptable limits;

(2) Move the manufacturing facility out of the community to an area that would not be affected by the noise emission;

(3) Contain the noise through construction of a complete enclosure around the fabrication area.

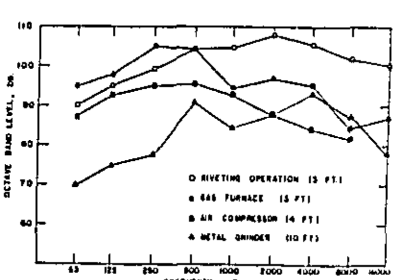

FIG. 3 NOISE SOURCE SPECTRUMS

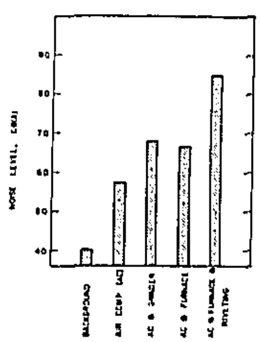

FIG. 4 RELATIVE CONTRIBUTIONS OF NOISE SOURCES

Panel radiation due to riveting precluded a serious consideration of (1). Riveting tests conducted using an enclosed part of the plant facilities coupled with computations indicated that a total enclosure could provide sufficient noise attenuation. Guidelines for construction of an enclosure structure around the fabrication area were subsequently prepared. Further consideration of (2) or (3) was left to company officials and the courts.

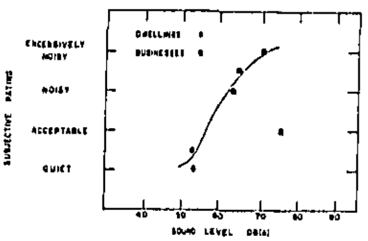

FIG. 5 COMMUNITY ATTITUDE SURVEY

RESOLUTION

Based on the detailed study and analysis, Judge Frank Eppes, Thirteenth Judicial Circuit, Greenville, South Carolina, prepared a court order to give plaintiffs the relief sought. The order was filed July 7, 1971, almost two years from the time the case was initiated. The company was ordered to either completely enclose the fabrication area where the riveting operation was carried out or move these operations from the premises within 18 months.

The court further ordered that the plaintiffs be paid sums ranging from $1350 to $2800 for past, present, and future monetary damages and for reimbursements of expenses and attorneys fees. There were indications that this settlement was a first in South Carolina in regard to a suit involving an alleged practice of environmental noise pollution on the part of a company.

CONTROL OF CONSTRUCTION NOISE

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The New York City Transit Authority has begun construction of the first sections of 40 miles of new subways to expand the 250 existing route miles of the New York City Transit System. This expansion program, costing well over 2 billion dollars, will take 10 years to complete. It will consist of a new subway line under Second Avenue in Manhattan, new lines in the boroughs of Queens, the Bronx, and Brooklyn, a tunnel under the East River connecting Manhattan and Queens, a tunnel under the Harlem River connecting Manhattan and the Bronx, and sections of subway beneath 63rd Street and Central Park connecting the existing subways along Sixth and Seventh Avenues in Manhattan, to the new East River tunnel.

Because this program is so large and will take several years to complete, a great deal of consideration has been given to reducing the adverse effects that this kind of heavy construction will have on the adjacent communities. Noise has proven to be an intrusive aspect of our construction activities, and therefore, we have conducted research to determine the most effective means to control construction noise.

One method, incorporated in noise control ordinances in several cities in this country, is to establish allowable sound levels for each item of construction equipment measured at a specified distance, usually 50 feet, except in the proposed New York City Noise Control ordinance, which is at 1 meter, and allow the contractor complete freedom as to placement and use of equipment. This requires contractors to either purchase new muffled equipment, when available, and/or modify existing equipment to reduce noise levels. This method requires no enforcement other than requiring a contractor to submit reports of the sound level of various pieces of machinery measured at specific distances.

It is our judgment that this method is entirely unsatisfactory, as it does not insure that the noise will be within an acceptable level at certain critical locations. Further, it would require contractors to modify existing machinery or purchase new machinery even if the construction activity was so far removed from any observer, or beneath a temporary 12 inch thick timber roadway, that the noise would not be objectionable or intrusive.

The second method investigated, modeled after noise control ordinances common to zoning requirements in several cities, including New York City, was to establish allowable sound levels to be measured at the perimeter of the construction site or at certain specified critical locations, such as at the nearest residential building. This would allow contractors to utilize existing machinery if it was far enough from a critical measuring location and the noise level at this location was acceptable. It would also allow a contractor more freedom to change construction methods or machinery to suit changing field conditions, something quite common in New York City.

The specification that is incorporated in all present and future New York City Transit Authority construction contracts takes the form of the second method discussed, a performance specification regulating the noise of the entire construction site. This specification sets the allowable sound levels on weekdays, between 7:00 a.m. and 11:00 p.m., at 75 dBA measured at the residential building nearest the construction site, and 60 dBA at all other times. It also establishes a table of maximum sound levels, to be measured at the location nearest the construction activity that is normally accessible to the general public, equal to the levels and duration of exposure as set forth in the Occupational Safety and Health Act of 1970 for protection of hearing. An upper limit of 85 dBA is specified for commercial locations, measured at the commercial building nearest the construction site.

This specification appeared for the first time in the contract for the construction of the two sections of subway which extend beneath Central Park in Manhattan, a 34 million dollar contract. Construction began in July 1971. The contractor's activities were monitored by the noise control engineers of my staff. It became evident that although the contractor's intentions were good, his understanding of acoustics and noise control engineering was far from adequate.

The initial phases of construction, which were close to residential buildings, consisted of excavating, removing rock and pavement, and the installation of soldier piles and a temporary timber roadway over the excavation area. This initial phase of construction is usually the noisiest. In order to meet the specification requirement of 75 dBA at the nearest residential building, the contractor purchased whatever muffled equipment, and installed mufflers having high insertion loss on all internal combustion engine driven machines. The only two commercially available muffled machines, incidentally, are air compressors, now made by several manufacturers, and paving breakers.

The contractor was very surprised to learn that even though he used the new muffled machines and modified his existing machinery, he failed to meet the required sound levels on many occasions. On those occasions, he was ordered to stop work until the noise level could be reduced to the allowable level. After several occasions of being ordered to stop work, the contractor realized that we were not going to permit him to work unless he met the requirements of the specification.

Several manufacturers of acoustical materials were contacted by the contractor, some of whom provided acoustical engineering advice with their product. The contractor also retained an acoustical consultant to advise him on the techniques and materials available that would bring the noise of his construction activities down.

The first machine that received major alterations was an air operated pile driver. The contractor constructed a shroud that enclosed the pile hammer. The shroud was made of 1/4 inch thick steel lined on the inside with sound absorbing open cell polyurethane foam. A "Flexopoly" pad was placed in the anvil of the hammer to prevent metal to metal contact, and the steel "H" piles were treated with a damping compound. Acoustic panels made of two sheets of 3/4 inch plywood with a two inch air space between them, were attached to the leads of the pile driving rig to act as acoustic barriers. Although the pile driving was slightly more difficult, the noise reduction of 20 dBA was obtained during driving of the last 1/3 of the pile, a reduction from 103 dBA to 83 dBA at 25 feet. Analysis of the noise of the pile driving operation made by the contractor's consultant, indicated that the air exhaust noise was the principal noise source. Based on this finding, the shroud was removed and a muffler was installed on the exhaust of the hammer, which worked equally well. This hammer operates at 60 blows per minute. The signature of the noise contained two distinct components; the noise of the hammer striking the pile and the noise of the air exhausting, occurring 1/2 second apart. By reducing the exhaust noise from 103 dBA to 85 dBA by means of the muffler, the character of the noise was dramatically changed, and the major components of the noise now occurred 1 second apart at 95 dBA.

Large wagon drills are extremely noisy, generating noise levels in the range of 105 dBA at 23 feet. It was necessary to use these drills very close to residential buildings. The contractor realized that he would be prohibited from using these drills if they were not quieted, and he therefore contacted two manufacturers and gave them his requirements. Each attempted to reduce the noise in a different manner. The Gardner Denver design consisted of a heavy molded fiberglass jacket around the body of the drill to reduce radiation from the drill, and a muffler on the exhaust. Sleeves were also placed over the drill steel to reduce radiation. The Ingersoll-Rand design consisted of a steel enclosure around the drill end of the machine. The steel enclosure had a door closed by pneumatic cylinders against a gasket, to allow changing the drill steel. Both worked quite well, and the Ingersoll-Rand design afforded a 20 dBA reduction measured at 23 feet, from 103 dBA to 83 dBA.

The modified Gardner Denver machine achieved a noise reduction of approximately 10 dBA, and was used by the contractor with a plywood housing around the drill end to obtain an additional 10 dBA reduction.

All internal combustion engine driven machines, including backhoes, welding machines, and generators, were fitted with new more effective exhaust mufflers, which reduced the noise by approximately 8 dBA at 23 feet.

Use of gasoline driven chain saws, which generate noise levels as high as 113 dBA at 3 feet, is prohibited. In their place, electric driven saws, which generate 75 to 80 dBA at 3 feet, are used. The gasoline engine driven generators which power the electric saws were enclosed in a sound insulating housing.

In many instances where it was not feasible to muffle the construction machine, or muffled machines were not commercially available, the contractor installed sound insulating housings around the tool and the worker. The housings consisted of plywood sheets with lead vinyl cloth attached to improve the transmission loss. Where reflected noise would be a problem, absorptive open cell polyurethane foam was installed on the noise side of the acoustic panel.

In general, by utilizing the commercially available muffled machinery, by modifying other machines by installing mufflers and housings, and by careful placement of the noisier pieces of machinery, the contractor was able to comply with the specification, at a cost of approximately 1% of the contract price.

We have found that construction noise can be brought under control at reasonable cost if it is required by the contracting party. The New York City Transit Authority is firmly committed to controlling construction noise, and urges other agencies, both governmental and private, to follow our example.

MEASUREMENT OF CONSTRUCTION SITE NOISE LEVELS

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INTRODUCTION

The purpose of this paper is to present a status report describing the efforts of the ad hoc members of the SAE Agricultural and Construction Machinery Sound Level sub-committee, to arrive at a usable method of measurement and description of construction site noise. The objective is to provide the construction industry and municipal enforcement agencies with a method usable by the unskilled which is simple, as accurate as possible, and requires only the simplest of instrumentation.

A procedure for measurement of construction site noise levels was quickly proposed. It consists of observation of a Class II sound level meter set on "slow" response - A-weighting network for the first five (5) seconds of each minute of a representative 30 minute period. The "central tendency" of the meter indicator during the five second period would be tabulated. Alternatively, it was proposed that the "maximum value" of the meter indicator observed during a 10 ± 2 second period at the start of each minute would be tabulated.

The arithmetic average $\bar{L}_a$ would be computed in accordance with equation (1)

$$\bar{L}_a = \left[\sum_{i=1}^{n} L_i\right] / n \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where L_i are those tabulated values within 6 dB(A) of the maximum tabulated value and n is the number of values used.

EVALUATION

The method was tested and proved to be easily used by the non-acoustician after preliminary instructions. The questions which had to be answered were:

1. How good is the sampling technique?
2. How representative of construction site noise are the values of L_a, thus obtained?

To accomplish the evaluation, members of the ad hoc committee were requested to forward tape recordings of construction site noise together with values of L_i obtained using the above procedures. Four recordings of individual construction sites were received and analyzed. These sites are noted as Cedar Knolls, Northern Trust, Seneca, and Hillquist. It is this analysis and comparison of averaging methods which are being reported.

These tape recorded data were analyzed using a computer-controlled real-time analyzer. Noise levels (A-weighted) for four second integration periods (the General Radio real-time analyzer doesn't have a five second integration), were stored on paper tape. The paper tape data were averaged as follows:

a) Arithmetic Average
Same as equation (1) using all L_i values.

b) Special Average
Equation (1) using only L_i within 6 dB(A) of maximum

c) "Energy" Average

$$\bar{L}_a = 10 \operatorname{Log}_{10} \left(\left[\sum_{i=1}^{n} \operatorname{Antilog} (L_i/10) \right] /n \right)$$

where L_i are all analyzed values.

The data tabulated from the "central tendency" and "maximum" sound level meter observations were treated in a manner as above. If the tabulations were not available or the calibration on the tape recording was suspect, a technician tabulated the noise level (A-weighted) values from the tape recorded data using the committee procedures.

Time histories of the A-weighted noise levels for two sites, the Cedar Knolls and the Northern Trust, obtained from the four second integrations, were plotted versus time in the following manner:

a) The levels stored on punch tape were energy averaged for one minute and the result plotted.
b) Only the first of the stored levels for each minute were plotted.

Superimposed on each plot are the results of the sound level meter "central tendency" tabulations. Figures 1 and 2 present these data for the Cedar Knolls site and Figures 3 and 4 present the Northern Trust data.

TABLE I
Construction Site Noise Level Summary in dB(A) re: 0.0002 microbar

	Cedar Knolls	Northern Trust	Seneca	Hillquist
Arithmetic Avg.				
Tape	64.9	86.4	84.0	76.8
Central Tendency Obs.	64.1	88.6	84.5	75.0
Maximum Obs.	66.1	88.9	86.2	77.5
Energy Avg.				
Tape	65.6	89.9	84.1	77.2
Central Tendency Obs.	64.2	92.2	84.5	75.8
Maximum Obs.	66.6	92.4	86.4	78.3
Special Avg.				
Tape	67.1	92.4	84.0	80.4
Central Tendency Obs.	64.1	95.0	84.5	77.4
Maximum Obs.	66.7	95.2	86.2	79.0
Statistics				
L_{90}	61	83	83	72
L_{50}	64	92	84	75
L_{10}	68	97	85	80

As seen in these plots, large differences do not occur often, and in the case of Northern Trust, occur during the transition period between noisy and less noisy periods. Since each averaging technique is a smoothing process, it appears that the sampling technique is satisfactory.

The A-weighted noise level histograms, Figures 5 - 8 , for each construction site contain the distribution of noise level for the four second integrations. The Northern Trust site is interesting and different from the others as the operation there consisted of fairly quiet periods, and periods when steel pile-driving was being accomplished.

Table I contains a summary of the results of these averaging methods, plus the statistical L_{90}, L_{50}, L_{10} noise level values. L_{90}, L_{50}, L_{10} being the noise level exceeded 90, 50, and 10 percent of the time, respectively, during the measurement period.

Note that, the arithmatically averaged values of "central tendency" agree with the L_{50} noise level except for the Northern Trust site. The special averaged maximum observations agree reasonably well with the L_{10} values.

A uniformly noisy construction site whose noise approaches a random distribution allows us to use either of the above measurement and averaging techniques; the construction site whose noise level distribution is very non-random may cause problems.

As was indicated earlier, this paper is presented as a status report of the committee's activities. Committee members would welcome suggestions for improvement, and in particular, the forwarding of single track tape recordings (30 minute) of construction site noise for similar analysis.

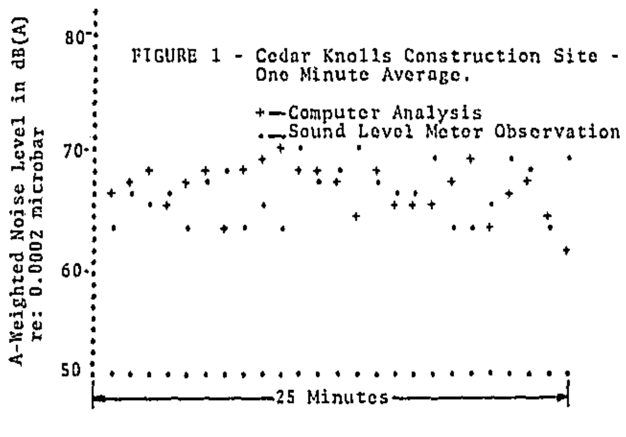

FIGURE 1 - Cedar Knolls Construction Site - One Minute Average.

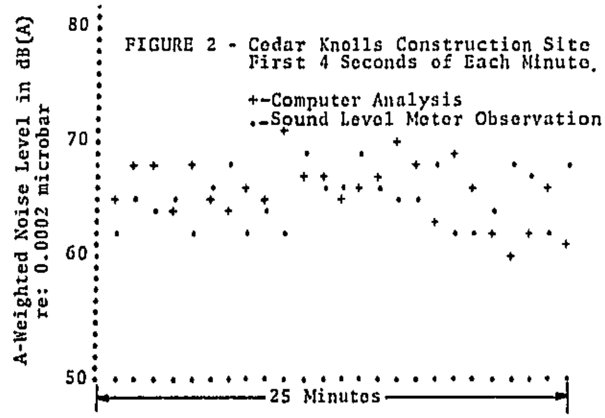

FIGURE 2 - Cedar Knolls Construction Site First 4 Seconds of Each Minute.

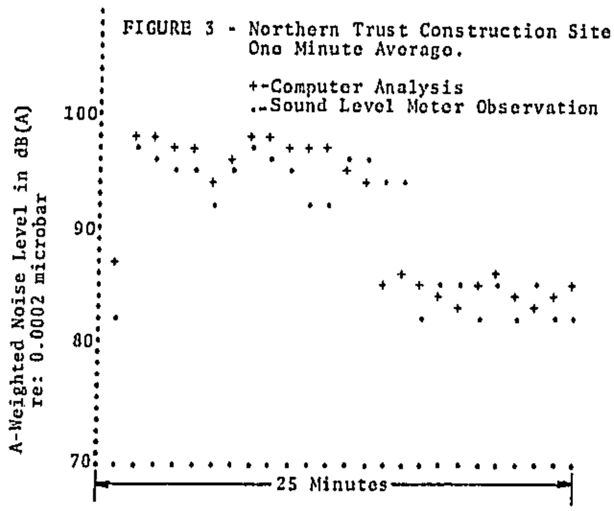

FIGURE 3 - Northern Trust Construction Site One Minute Average.

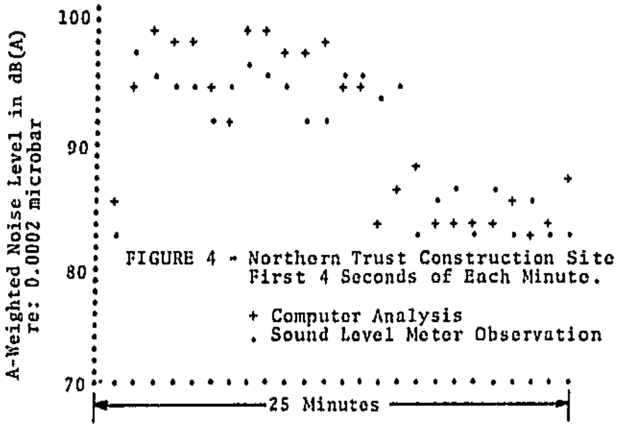

FIGURE 4 - Northern Trust Construction Site First 4 Seconds of Each Minute.

A-Weighted Noise Level in dB(A)
re: 0.0002 microbar

```
55
56
57 **  3
58 *  1
59 ****  8
60 ********  16
61 ************  24
62 *********************  42
63 *********************  41
64 *****************************  58
65 **********************  44
66 *************************  50
67 *************************  49
68 ***************  30
69 *****  10
70 **  3
71 *  1
72
73
74
75
```

FIGURE 5 - Cedar Knolls Construction Site.

```
L(90) =   61
L(50) =   64
L(10) =   68
```

A-Weighted Noise Level in dB(A)
re: 0.0002 microbar

```
 80
 81
 82 ***  5
 83 ******************************************************  108
 84 ******  12
 85 ********  15
 86 ***********************  45
 87
 88
 89
 90
 91
 92 *****  9
 93 *******  14
 94 ****  8
 95 ****************  32
 96 *******************  38
 97 *************************************  73
 98 ***********  21
 99
100
```

FIGURE 6 - Northern Trust Construction Site.

```
L(90) =   83
L(50) =   92
L(10) =   97
```

```
A-Weighted Noise Level in dB(A)
re: 0.0002 microbar

75
76
77
78
79
80
81 *    4
82 ******   29
83 ************************** 138
84 ********************************* 174
85 **********   52
86 *    3
87
88
89
90
```

FIGURE 7 - Seneca Construction Site.

```
L(90) =   83
L(50) =   84
L(10) =   85
```

*

```
A-Weighted Noise Level in dB(A)
re: 0.0002 microbar

65
66
67
68
69
70 ***    3
71 ***********   12
72 ********************   22
73 *****************************   32
74 ********************************************   49
75 ****************************************   44
76 ***************************   30
77 *********************************   37
78 ****************   18
79 **********************   24
80 ****************   18
81 *********   10
82 *****    5
83 ****    4
84 **    2
85
```

FIGURE 8 - Hillquist Construction Site.

```
L(90) =   72
L(50) =   75
L(10) =   80
```

ENVIRONMENTAL NOISE: ASSESSMENT AND IMPACT

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Historically the principal criteria involved in the selection of sites for manufacturing plants have been economic in nature. Little consideration was given to factors involving environmental change. As the condition of the environment degraded under the stress of increasing numbers of manufacturing and associated activities, it became obvious that a reversal of the trend would be required if man and nature were to continue to exist in harmony.

Out of this realization grew the National Environmental Policy Act of 1969 (NEPA), and the Environmental Quality Improvement Act of 1970. The stated purposes of these acts are to "...promote efforts which will prevent or eliminate damage to the environment and biosphere..." and..."provide for the enhancement of environmental quality..."

One of the provisions of the NEPA is one which requires that every recommendation or report on major Federal action contain a detailed statement--referred to as an Environmental Impact Statement or EIS--covering the following: (i) the environmental impact of the proposed action, (ii) any adverse environmental effects which cannot be avoided should the proposal be implemented, (iii) alternatives to the proposed action, (iv) the relationship between local short-term uses of man's environment and the maintenance and enhancement of long-term productivity, and, (v) any irreversible and irretrievable commitments of resources which would be involved in the proposed action should it be implemented.

The environmental features which must be considered in an EIS information draft include the following: (i) Air Quality, (ii) Water Quality, (iii) Land Use, (iv) Noise, (v) Human Environment, (vi) Aesthetics, and (vii) Socio-Economic Factors. Because of the wide range of subject matter covered, the logical approach is to organize an Environmental Task Group (ETG) made up of experts from each of the fields mentioned. In addition, the team should include experts in the areas of information science, logistics, process chemistry, plant design, and other specialists as required by the nature of the proposed installation. No single individual would be likely to possess sufficient expertise in all of these fields to handle all of the investigations required, and if he did, the importance of time in obtaining approval to construct the plant would negate a one-man effort.

In addition to the compilation of the information required for the EIS, an ETG can function as an advisory body to the design group and the plant operating team. In the course of collecting information on the extent to which the construction and operation of the proposed plant will influence the environment, the team members become exceptionally well acquainted with both the existing environmental features and with the characteristics of the plant.

The work of the ETG does not cease when the draft of the EIS information is presented to the plant builder for forwarding to the appropriate agency, but continues in the form of providing advisory services to the designer and operator, monitoring changes in design, construction, and operating procedures to prevent inadvertant adverse environmental impacts, and conducting follow-up surveys to verify predictions of operating effects.

A role of the ETG not discussed here is that of selecting a plant site on the basis of environmental considerations. This role is a very important one since it provides information on proposed alternatives and insures that, of the sites available, the one with the least adverse impact on the environment is known. The discussion in this paper, however, focusses on considerations involved after the site for the facility has been selected.

The intention of this paper is to communicate some of the knowledge which has been obtained in the field of noise as it applies to Environmental Impact Statements and facility construction. Brief descriptions will be given of the techniques for conducting baseline surveys, including methods of selecting measurement locations, techniques for predicting both the plant noise potential and the extent of acoustical treatment required to meet restrictions aimed at protecting the environment, and procedures for predicting the effect of the plant on the noise environment.

METHODOLOGY FOR ENVIRONMENTAL NOISE ASSESSMENT

From Battelle's experience with environmental noise assessment, a methodology has evolved which provides for an orderly flow of tasks with the necessary interaction between the ETG and the designer, builder, and operator. This methodology is shown in flowchart form in Figure 1.

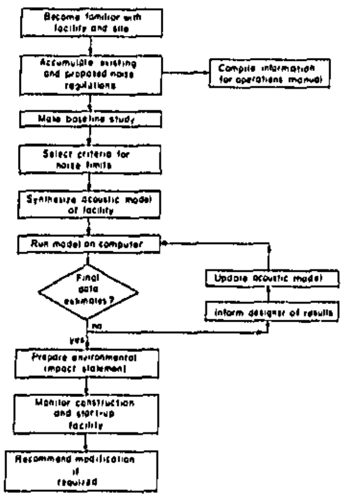

Fig. 1 - Flowchart of methodology used in environmental noise prediction and assessment

Familiarization With Facility Noise Characteristics and Site

Early in the assessment study it is necessary to become generally familiar with both the facility noise characteristics and the site. The former is best accomplished by a meeting with the construction firm that will be building the facility; the latter by making a visit to the proposed site. This background information will provide preparation for making intelligent choices of locations for the baseline noise measurements and insure proper screening of the various regulations and criteria that the installation will be expected to meet during its construction and use. Specific items that should be noted during this initial familiarization are: location with respect to the site of existing major noise sources, general noise climate of the proposed site, and proximity of residential and public areas to the site.

Other sources of information that will aid in familiarization with the site include aerial photographs; and topographic maps. The preliminary visit to the site may also indicate the need for traffic survey information for a nearby interstate highway, or airport traffic information if highway or air carrier traffic seems significant. If noisy industrial plants are adjacent to the proposed site it is appropriate to inquire into their production schedule in order to establish "typical" levels during the baseline study. Even though the nearest dwelling may be a mile or so from the proposed site, an inquiry into possible uses of intermediate property should be made, in order to establish appropriate limits on noise emmissions and to minimize the potential for future litigation.

Collection of Existing and Proposed Noise Regulations

In this phase a search is made of all applicable regulations and criteria the facility would be expected to meet. This includes either existing or pending noise regulations at the Federal, state, and local levels. While most regulations have been drafted to insure a reasonable noise environment for humans, the existence of wildlife adjacent to the site should not be overlooked. This is especially true when mink or other "noise sensitive" animals are found in the area.[1]

Baseline Survey of Site

The baseline noise survey consists of making measurements of noise at selected locations on and near the proposed site. The survey has several purposes. It indicates the present noise climate at the proposed site, and in the case of continuous process plants, may provide the only record of the "before-the-plant" conditions which are necessary to establish an environmental impact reference level. Since all surveys are taken with limited time to meet practical budget limits, it is imperative that the most significant locations and times be utilized.

The familiarization with the proposed installation and its site are valuable in this selection, since the information gained can be used to select the more important locations. Factors to be considered in the baseline survey include the following:

(a) Measurements should be taken at critical locations as determined by the wording of noise regulations. For example, measurements should be taken at locations where the noise levels are now at maximum and minimum values and in locations where the construction engineers indicate the greatest potential for added noise along the property site.

(b) Measurements should be taken in the residential areas nearest to the proposed site. Without property line regulations, the nearest "possible" neighbors may provide a noise design criteria. It is important to note both present residential areas and areas of possible future growth.

(c) Measurement samples should include the maximum and minimum periods of noise around the plant site. For example, while interstate highway noise may dominate the noise climate during the day, interstate noise levels often drop considerably in the early hours of the morning. In the case of a proposed facility expected to have a fairly constant noise level, this may make the noise design criteria for the facility more severe than first anticipated.

(d) Since community noise is typically not a constant value, the techniques of statistical analysis should be utilized to indicate a typical background level (90% level) and a typical maximum level (10% level).

(e) Since it is likely that a noise survey will be taken after the facility is in operation, it is well to consider good vantage points from which to make "before and after" comparisons. These points should be selected for their analysis value. Topographical maps can sometimes be used for this purpose since they show the terrain features adjacent to the site.

Having taken noise data at the various measurement locations and stored the data on magnetic tape, the next step is to reduce the data into the most meaningful form. Although the data could be presented in various ways, the A-weighted cumulative distribution appears to be the most useful. The A-weighting has been found to be very reliable in predicting the human response to noise, while the cumulative distribution takes into account the variability of noise with time in urban environments. This format has also been adopted by the U. S. Department of Housing and Urban Development (HUD) for non-aircraft noise in residential areas in its "guideline criteria".[(2)] Figure 2 shows one such time sample where the various regions of the plot are those adopted by HUD. The vertical scale is divided such that noise with a Gaussian distribution, such as dense highway traffic, will plot as a straight line.[(3)] The values of most interest in the plot are the levels which are exceeded 10, 50, and 90% of the time, which are referred to as the intrusive, median, and ambient levels, respectively.

There are several methods that can be used to convert the tape recorded noise time histories into its cumulative distribution function. However, if digital equipment facilities are available, it may be most advantageous to perform the analysis digitally. Battelle uses this approach in its data reduction, and incorporates a high speed digital plotter to produce the plot shown in Figure 2 as the final output. The overall process is shown in Figure 3 in its current form, and plans are now in progress to refine this system further. Basically the system uses the field recordings, an A-weighted filter and logarithmic converter to obtain a dc signal proportional to the A-level. This signal is then scaled into the range of an analog-to-digital (A/D) converter in which the negative voltage limit of the A/D converter corresponds to a "reference level" which is supplied later to the plotting routine. The resolution of the system is approximately 0.2 dBA using 201 levels and a 40 dB dynamic range. The "logic signal" is used to turn the conversion on and off at appropriate times. Another computerized approach is described in the following reference (4).

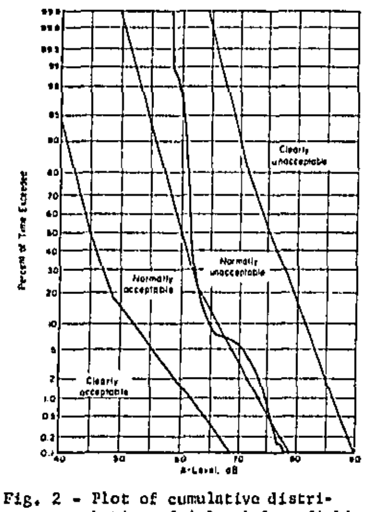

Fig. 2 - Plot of cumulative distribution of A-level from field tape recordings. Subjective ratings are those of HUD for non-aircraft noise sources in residential areas.

Determination of Appropriate Noise Criteria

While at first glance the criterion selection would appear to be a simple task, experience has shown that this often is one of the more difficult parts of the study. Consider the "simple" task of describing the background level at a large site. More likely than not, the noise climate varies considerably around the site, as a function of the activity of neighboring plants and time-dependent processes such as highway traffic. Usually the criterion selection involves determining the most severe conditions which the plant will be expected to meet, and this in turn usually requires that supporting levels be determined in the baseline study.

Synthesis of an Acoustic Model Of The Proposed Facility

In this phase the acoustic characteristics of the facility are assembled together with location data and background levels to form an "acoustic model" of the plant. Battelle has developed a computer program called PREDICT, which accepts these data and then performs the necessary calculations to obtain noise predictions at specified locations, using the appropriate equipment noise estimates and measured background levels.

Working With The Facility Designer

After preliminary results from the acoustic model are obtained, the information is given to the facility designer to use as an input to his design. These results aid him in the selection of individual items of equipment as to the amount of noise abatement that is needed in each component. In addition, the results may indicate whether the size of the land acquisition for the facility is sufficient or whether the noisiest equipment should be re-arranged on the site to obtain more distance between it and the property line. As the designer obtains better estimates of the equipment the model is re-run on the computer. A more detailed discussion of the methods used to predict the noise potential is given later in this paper.

Preparation of the Applicant's Draft Environmental Impact Statement

Having completed the baseline survey, the prediction of the noise potential of the proposed facility, and the computation of noise reduction required to meet the noise criteria established, the next step is to incorporate the information in the Applicant's Draft Environmental Impact Statement (DEIS). The basic organization of the DEIS is based on the five topics required by NEPA (see page 1). The noise portion of the various sections is integrated with the other environmental factors, such as air pollution, water pollution, etc.

PREDICTING THE NOISE POTENTIAL

Perhaps the key element of the methodology described above is the prediction of the noise levels that will exist in the future with the proposed facility operating. The procedure for making these predictions at specified locations consists of accumulating sufficient data to describe the future sound sources and background levels, and then applying mathematical analysis to obtain the desired results. In order to obtain quick and accurate results a computer program entitled PREDICT was developed at Battelle to perform the necessary analysis. The mathematical analysis, along with the data, constitute an acoustic model in which the parameters can be changed at will in order to ascertain the results. It goes without saying, of course, that the results can only be as accurate as the input data. Sources of data include trade journals in the particular industry, including the bibliography with individual articles. Other sources include references works and measurements of similar equipment. The most useful information is that given in terms of raw parameters, since these are usually the only tangible pieces of data available in the early phases of a project. The results of these early "predictions" are usually pessimistic and tend to induce the facility planners to make readily available more accurate information as it becomes known in addition to pointing out problem areas.

Description of Program PREDICT

Program PREDICT is a Battelle-developed Fortran IV computer program used to predict the sound pressure levels in octave bands at chosen locations in the proximity of a major facility installation. Linear or weighted levels can be obtained. The program in its present configuration allows for up to 50 different sound sources and 15 different locations at which a prediction can be made. Basically the program solves the following equation on an octave band basis for each of the sound sources for a given location, and then determines the combined effects of all of the sources on the location:[5]

$$L_p = L_w - 20 \log_{10} r + DI_\theta - A_e - 11$$

where

L_p = sound pressure level inddB at a given location defined by r and θ,

L_w = sound power level of the source, dB re: 10^{-12} watts,

r = distance from source, meters,

DI_θ = directivity index in direction, θ,

A_e = atmospheric attenuation for distance, r, and

11 = $10 \log_{10}(4\pi)$, db.

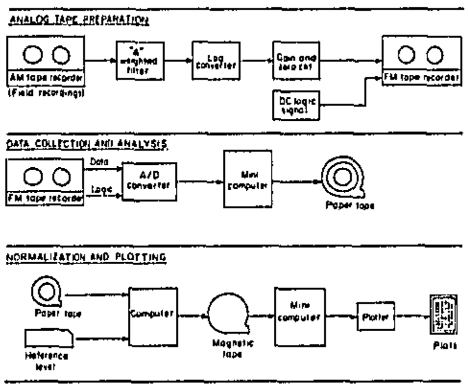

Fig. 3 - Block diagram of data reduction method used to obtain plot shown in Fig. 2.

The different parameters of the equation are determined from raw input data, allowing many locations to be analyzed from the same set of sound source input cards. For example, the distance, r, in the equation has to be evaluated for each source. This computation is made by the program from coordinate data for the source and location. It should be noted that since the locations are evaluated on an octave band basis, the equation is solved on a matrix basis (locations by octave bands) before obtaining the combined result.

DETERMINING THE REQUIRED NOISE REDUCTION

After the sound source and location data has been acquired and the resultant sound pressure levels have been predicted, the pertinent questions are: is the plant quiet enough, and--if not--which pieces of equipment must be quieted, and by how much. In order to answer these questions a criterion must first be established with which to compare the results of the predicted levels. After this has been done, the problem must be turned around in order to determine which pieces of equipment are the worst offenders, and by how much and in what octave bands they must be silenced in order to meet this criterion.

Selecting the Criterion

One of the bases for noise emission criteria is found by considering the most stringent of the noise laws applicable(6) However, a problem with this method occurs if the area in question does not have a realistic ordinance. For instance, if the most stringent ordinance is a local one which was drafted several years earlier on the basis of a noise survey taken at that time, before the new interstate highway, airport, or other noise producer was added, it is possible to find that even before the proposed facility is started the existing noise levels at the property line exceed the allowable ones. In addition to this problem there is usually a time-varying background level (with facilities in urban environments), and therefore the resultant combined contributions also vary with time of day and day of week. One would also suspect that if the legal limit in an octave band was, say 80 dB, and the background was only 60 dB, that the installation's contribution to the noise environment could be as high as 80 dB without incurring problems. However, many ordinances include phrases which require the plant not to "degrade" the environment. Another difficulty with basing calculations on a dB buffer between background and ordinance levels is that future growth around the area of the plant may take away this differential. It can be seen that considerable judgement is required in the establishment of a criterion that is both fair and reasonable to all concerned.

Determining Equipment Noise Reduction Requirements

After selection of a suitable criteria which the plant must meet, it would be possible to use trial-and-error procedures and reduce the sound power levels (e.g. applying noise reduction) until the plant meets the selected criterion. For thirty or forty sources which may be spread out over a several acre plot, however, this would be quite time consuming. For this purpose an optional noise reduction analysis routine was added to PREDICT. The net result is that after all locations have been analyzed an "equipment matrix" contains the necessary reductions to meet a given criterion at all of the locations simultaneously. In this manner it is possible to state which equipment needs quieting, and further to specify this in terms of the reduction in each octave band.

CONCLUSIONS

The use of an Environmental Task Group not only provides a maximum of information in a minimum of time but also provides a degree of expertise that exceeds that of the same experts working alone. The methodology described in this paper illustrates the manner in which a Noise Control Team can not only provide the information required for the EIS but also furnish information needed by the designer in order to minimize the impact of the facility on the acoustic environment.

REFERENCES

(1) Memphis State Univ., Effects of Noise on Wildlife and Other Animals, U.S. Government Printing Office (GPO), Washington D.C., 20404. GPO No. 5500-0055 (1971)

(2) Schultz, T.J. 1971. Noise Assessment Guide Line; Technical Background for Noise Abatement in HUD's Operating Programs. Performed by Bolt, Beranek and Newman. HUD TE/MA 172. 210 pp.

(3) Schmitt, Samuel A., Measuring Uncertainty, Addison-Wesley Publishing, Inc., Reading, Mass., pp 210-214 (1969)

(4) Goodfriend and Associates, Noise from Industrial Plants, GPO, GPO No. 5500-0042 (1971)

(5) Beranek and others, Noise and Vibration Control, McGraw-Hill: New York (1971) pg 165.

(6) For an overview of various regulatory schemes in the U. S. see: George Washington Univ., Laws and Regulatory Schemes for Noise Abatement, GPO, GPO No. 5500-0046 (1971)

DETERMINE BY LITERATURE SEARCH THE METHODOLOGY TO CONDUCT A COMMUNITY NOISE SURVEY

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This paper presents the results of a literature search to determine the methodology to conduct a community noise survey. Essential electronic hardware and software is currently available and is capable of supporting any method chosen.

Despite what may be termed a confusion of facts, the methodology has been determined. The methodology is composed of three elements, combined they form the methodology.

Element	Associated Assumption
1. The physical properties of sound	There are no major indeterminants in measuring the physical properties of sound. Trade-offs occur between accuracy, statistical sampling of events and direct and indirect cost.
2. Acoustics response of individuals	One may select a reasonably valid criteria for subjective response from a number of schools of support. This selection includes the software and hardware technology necessary for its implementation.
3. The individual scaled to the community level	When applied to a specified time period and geographic region, one may model a set of individual response characteristics and generate from this data a scaling response to the community level.

One may choose from any number of methods to derive the basis for conducting a community noise survey. Figure 1 presents a tabulation of some 15 possible choices. Application of the methodology may be based on use of a historical or current method and is illustrated as follows:

Noise Source	The Methodology	Reference
Transportation	dB_A vs community response	2, 15
Aircraft	CNR vs Multiple Classification Analysis (MCA)	11, 14
All	dB_A, CNR, NEF vs numerical value	16

THE PHYSICAL PROPERTIES OF SOUND

The physical properties of a sound wave can be mathematically analyzed; evaluation can be made of its statistical properties and a statistical envelope applied to a series of events.

Significant in data processing are corrections to a standard day to account for meteorological conditions, and may include corrections for temperature, humidity, wind, terrain, altitude and turbulence. Further, within each noise source, there is a statistical spread due to manufacturing, assembly and operational conditions of the equipment. The combination of these variables -- noise source, environment and meteorological conditions, present significant requirements on what has come to be considered the accuracy of the test data.

For example, it is not uncommon to process some 100,000 1/3 octave band sound pressure level data points to derive a three-number (takeoff, sideling and approach) value to determine that aircraft noise certification has been met. (1)

The first assumption for this paper, therefore, becomes:

There are no major indeterminants in measuring the physical properties of sound. Trade-off occurs between accuracy, statistical sampling and direct/indirect cost considerations.

ACOUSTICS RESPONSE OF INDIVIDUALS

The medical aspect of acoustics is covered in the physiological and psychological response of individuals in the following manner:

Individual Response
(State of Health)

Physiological		Psycological
Bio-acoustics	(1)	Psycho-acoustics
Function of auditory mechanism	(2)	Subjective response: annoyance, loudness, pitch
Other: vascular	(3)	Other: sleep
Synergestic effects: vibrations	(4)	Synergestic effects: stress

Although hearing loss and subjective reactions to noise have been recognized since the industrial revolution, it has only been as recent as 1930 (2) that a community noise survey was conducted by a public health agency. It remained until 1961 to witness an International Symposium on Noise sponsored by industry, government and public health officials (3).

An explosion of information in the bio-acoustic and psycho-acoustic sciences has occurred comparatively recently. Some of what has developed are what may be called "schools of support" or, in the words of Botsford (4), "the dB Weighting Game."

To illustrate, essential to any subjective response is the obtaining of an index of human judgements and subsequent mathematical model which attempts to reproduce the information contained in the judgements. This judgement process follows one of four patterns and is presented in the tabular manner which follows:

Judgement Process (5)	Source
1.) The subject is asked to place the stimuli in one of several categories.	Bishop 1967, Pearsons & Horonjeff, 1967
2.) The subject is asked to give an estimate of the relative magnitude of one stimulus relative to a standard.	Stevens 1955, Parnell, Nagel & Parry 1967
3.) The subject is asked to decide which of two stimuli is of greater magnitude.	Kryter 1969
4.) The subject is asked to adjust one stimulus in some relation to another to determine growth rate.	Stevens 1955

An attempt to lend coherence to the diversity of published information was made recently by Kryter (6). This text provides a compendium of bio-acoustics and psycho-acoustics in regard to damage risk criteria, subjective response to noise, environmental noise and its evaluation and other responses to noise. The historical development of various weighting scales is also presented.

The leads to the second assumption, namely: <u>One may select a reasonably valid subjective response from a number of "schools of support." This selection includes the software and hardware technology necessary for its implimentation.</u>

At this point in time, there is no universal agreement within the medical profession on the detail function of the auditory mechanism. The area under question is the region of the sensory-neural response.

Finally, in the area of subjective response, there is no universally accepted one best method as it relates to criteria for a noise survey.

INDIVIDUAL SCALED TO THE COMMUNITY LEVEL

Three interactions occur at the community level in the following manner:

Community Response

Legal — Psychological — Sociological

The community response to noise is effected by the system of laws under which a community functions. A detailed treatment is given of this subject area in references 7, 8 and 9.

The community response from a psychological basis is determined from the subjective response of individuals. Here empirical data, tables and curves are extrapolated exponentially to encompass large population groups, cities and high density magapolis regions.

The community response to noise from a sociological basis is determined by what is referred to as a "social survey." A social survey was conducted in the vicinity of the London Heathrow Airport in 1961 covering residential districts within ten miles of the airport. The sampling population in the area was 1,909 individuals (1).

A more recent study covered the metropolitan areas of Logan International Airport in Boston, O'Hare Internation Airport in Chicago, Dallas International Airport in Dallas, Stapleton International Airpor in Denver, Los Angeles International Airport in Los Angeles, Miami Internation Airport, Miami and Kennedy International Airport in New York. For the seven cities, the sampling population in these areas were a total of 8,207 interviews. (11) This reference presents the use of multiple classification analysis (MCA) as an analytical response method.

A recent book, Future Shock, by Alvin Toffler (12), presents a penetrating argument on the transience of contemporary 20th century cybernated society.

This leads to the third assumption namely:

When applied to a specific time period and geographic region one may model a set of individual response characteristics and generate from this data a scaling response to the community level.

This data becomes high transient as a function of time. Obsolence is at a significant rate and affects the selection of individual response characteristics used for scaling to the community level.

The use of community noise survey data for legal purposes is valid provided it is periodically reviewed in the context of experimental jurisprudence. To illustrate, a recent example has been the rapid increase in numbers and in problems associated with snowmobiles. The noise levels at the operator's ears fall into the region for statistical hearing loss. Further, the community reaction to the destruction of the environment is expressed in storms of protest. Restrictions and enforcement crosses legal boundaries of communities and incorporated areas. The estimated population of snowmobiles in the United States stands at 1.5 million vehicles (13).

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(10) Burns, W., Noise and Man, J. Lippincott Co., 1969.

(11) Community Reaction to Noise, Volume I, Tracor, Inc., NASA CR-1761, July 1971

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(15) Branch, M. D., "Outdoor Noise and the Metropolitan Environment," Appendix A., Case Study of Los Angeles with Special Reference to Aircraft, Department of City Planning, Los Angeles, California, 1970

(16) Housing & Urban Development Circular No. 1390.2 "Noise Abatement and Control."

FIGURE 1

Author *	Method	
	Name	Symbol
Churcher-King	decibels, A. scale	dB_A
Churcher-King	decibels, B, C scale	dB_B, dBc
Kryter	decibels, D scale	dB_D
Fletcher-Munson	sone (linear), phon (log)	sone - phon
French-Steinberg	articulation index	AI
Stevens	Mark VI procedure	sone - phon
Zwicker	energy density	sone/Bark - phon
Kryter	noy (linear), preceived noise in decibels (log)	NOY - PNdB
FAA	effective perceived noise level	EPNL
Kryter	integrated perceived noise level	IPNL
Anon	composite noise rating	CNR
Bishop	noise exposure forecast	NEF
McKennell	noise and number index	NNI
Griffiths-Langdon	traffic noise index	TNI
Robinson	noise pollution level	NPL
Goldstein	environmental noise indexes	ENI

* Generally recognized author. No attempt is made to historically document all individuals and all methods developed within each category. The total of 16 methods listed here could readily be doubled.

UNITED STATES DEPARTMENT OF HOUSING AND URBAN DEVELOPMENT (HUD)
Circular 1390.2 "Noise Abatement and Control", Page 8

EXTERNAL NOISE EXPOSURE STANDARDS

GENERAL EXTERNAL EXPOSURE		AIRPORT ENVIRONMENT				
		NOISE EXPOSURE FORECASTS (NEF)		COMPOSITE NOISE RATING (CNR)		
CONDITION	VALUE IN dB	VALUE	CATEGORY	VALUE		ZONE
				Take-Off/	Run-Ups	
Unacceptable Exceptions are strongly discouraged & require a 102 (2) C environmental statement and the Secretary's approval	Exceeds 80 dB_A 60 minutes per 24 hours Exceeds 75 dB_A 8 hours per 24 hours	>40	C	>115	>95	3
Discretionary (Normally Unacceptable) Approvals require noise attentuation measures, the regional administrator's concurrence and a 102 (2) C environmental statement	Exceeds 65 dB_A 8 hours per 24 hours Loud repetitive sounds on site	30 to 40	B	100 to 115	80 to 90	2
Discretionary (Normally Acceptable)	Does not exceed 65 dB_A 8 hours per 24 hours	---	---	---	---	---
Acceptable	Does not exceed 45 dB_A more than 30 minutes per 24 hours	<30	A	<100	<80	1

NOISE CONTROL IN BUILDINGS

NOISE CONTROL AND THE N.Y.C. BUILDING CODE 1962 - 1972

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In 1962 our firm was engaged to prepare the noise control provisions for a complete new Building Code for New York City. The entire code was written by various specialists under the direction of Polytechnic Institute of Brooklyn staff.

No U. S. Municipality had modern noise control requirements for buildings when the new code was commissioned. In Europe, noise control provisions were limited to specification of acoustical separation and impact transmission between apartments. Noise transmission from mechanical equipment -- airborne or structure borne -- had received no serious consideration in building codes at that time.

It was decided that the New York City Code would provide restrictions against all potential intrusions of noise into a multiple dwelling building that could be legally controlled by the building department with one exception -- plumbing noise. It is important to note that the code's noise control provisions apply only to multiple dwellings such as apartment houses, apartment hotels and school dormitories -- not to office buildings and other commercial structures. Also, that any building code sets forth only minimum standards. In the case of the New York City Code, these minimum noise control standards were written with the knowledge that the apartment buildings would be located in areas with people and vehicular traffic density that provide high background noise levels. Some of the early criticisms, which I will discuss, attempted to make an invalid comparison between this code's requirements and that of European codes. Such a comparison ignored the fact that European noise restrictions applied mainly to buildings located in suburban and rural low noise level environments.

Figure 1 diagrams the noise control provisions of the code. This figure also indicates the criteria that were established after many hours of argument, review, consultation and finally, hearings in the City Council.

The abbreviations noted on this Figure relate to:

STC-Sound transmission class in accordance with ASTM E90-61T (Ref. 1 and 2).

INR-Impact noise rating, as called for by the FHA No. 750 (Ref. 3) based on data obtained in accordance with ISO-R140 (Ref. 4).

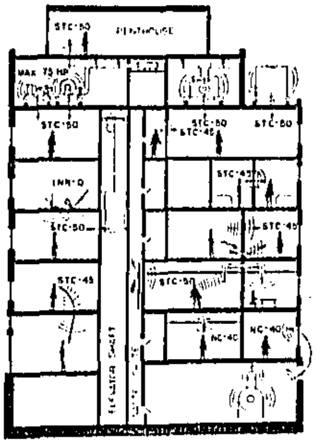
NC-Noise Criteria Curves (Ref. 5).

It may be noted on Figure 1 that STC 50 constructions are required to reduce airborne boiler, elevator shaft, rubbish chute and other mechanical equipment noise transmission to immediately adjacent dwelling spaces. STC 50 seems to satisfy most minimum requirements for sound transmission loss between small mechanical equipment rooms and dwelling spaces. However, optional provision is made for the architect to submit calculations applying tested sound power ratings of mechanical equipment to specific conditions.

Rapid movement of air through ducts, especially at turns and takeoffs, gives rise to noise. It seems likely that in the future, high velocity ducts may be installed in apartment dwellings in the vicinity of occupied rooms. It was, therefore, decided to limit noise generation resulting from the use of ducts with high velocity air flow.

Noise generation by air termination devices such as grilles, registers, diffusers, fan coil units, and induction units was also controlled in the code. (Ref. 6).

Figure 1



Briefly, the noise control provisions were selected to provide an acoustical separation between apartments of about a Class 45. Noise intrusion from mechanical equipment was not to exceed a level in excess of NC 40. Floor constructions were to provide an INR 0 (IIC 51) for impact isolation. Minimum requirements for the vibration isolation of machinery were specified to maintain the aforementioned NC 40 in nearby apartments. Noise radiating mechanical equipment located on a nearby building of any type or use or in or on the multiple dwelling building were restricted in permissible noise output to also maintain NC 40 levels inside the apartments.

This is just a very quick review of the basic criteria; the details that had to be carefully prescribed to avoid bypassing these criteria are contained in the complete code section which was printed in the January 26, 1967 House Congressional Record, pages 669 to 673 by Congressman Kupferman.

Not unexpectedly, this code section received much publicity -- some good, some misleading. The headlines probably typify the range of commentary:

NOISE CAN BE CONTROLLED IN CITY

NOISE CONTROL SOUGHT FOR FLATS

NOISE IS CITY'S NO. 1 ENEMY-MAKER

NEW BUILDING CODE SAYS, "S-s-sh"

CODE PROPOSALS WOULD CUT NOISE

LESS NOISE COMING UP

APARTMENTS TO GET JUST ENOUGH NOISE

The most interesting criticisms of the code were received during the City Council Hearings when the public, citizen organizations and others had the opportunity of having their day in court. During these hearings there were few arguments that the noise control provisions were too stringent. These came later! The opinions voiced were almost entirely on the side that the code was too lenient and were primarily confined to criticism of the STC 45 for partitions and floor constructions which provided the acoustical separation between apartments and of the NC 40 maximum permissible noise levels in apartments from exterior located mechanical equipment.

Responding to this rather one-sided pressure, the City Council increased the required partition and floor construction sound transmission class from STC 45 to STC 50 and reduced permissible noise intrusion from mechanical equipment from NC 40 to NC 35, both changes to become effective in 1972.

ENFORCEMENT

In 1970, meetings were held with the Commissioner of Buildings and members of his staff to discuss how the noise control provisions would be enforced. The procedure decided upon may be described in a simplified manner by indicating the action that may be taken by an architect after he completes his working drawings. The architect and the Building Department Plan Examiner confer to determine whether all code provisions (including noise control) have been met. At this time, the architect may select one of two options:

a) he may submit his drawings with the noise control requirements pertaining to his building noted and obtain approval by submitting evidence that the construction and equipment used throughout the building have been approved by the Materials, Equipment and Assemblies Division of the Building Department. This may be done by referring to approved laboratory tests previously conducted, or by using constructions listed in the Code's Reference Standards.

b) he may decide to have field tests conducted after the building has been completed, but preferably before it is occupied.

During these meetings the Department decided that both field inspection and tests would be conducted by their own staff of Building Inspectors.

The training of a group of 30 plan examiners and building inspectors was started in 1970. This educational effort included 8 days of concentrated study of basic acoustical terminology, acoustical properties of materials, laboratory test procedures and field inspection, but not field testing procedures. Those attending were selected for reasons of special aptitude and/or engineering degrees and included several night student candidates for doctorate degrees in engineering. The results of this course have been deemed to be successful, especially for the plan examiners who are enforcing the portion of the noise control provisions that pertain to the drawing stage of the building.

The second course was conducted in March, 1971, for 11 Building Inspectors and consisted of training in field measurement of transmission loss, impact transmission loss and sound pressure levels. Equipment purchased by New York City was used both in classroom demonstrations and in actual multifamily dwellings recently completed and unoccupied.

Some of those attending the course seemed to have difficulty with the equipment and also with the calculations leading to the single number ratings used in the Code -- Sound Transmission Class, Impact Noise Rating and Noise Criterion (NC) Levels.

RECENT CRITICISMS

Since the code went into effect there have been but a few criticisms that the noise control provisions are too lenient.

There have been a number of complaints that the requirements are too difficult and expensive to achieve in light of present building technology. Specifically, building material manufacturers, builders, and architects have come forward with complaints that specific provisions of the noise control section are too stringent -- mainly using two arguments; the number of available constructions meeting the requirements of the code are too few and not sufficiently diverse, and the constructions are too expensive. By the time the code became effective in 1971, building manufacturers had provided sufficient partition and floor constructions that met the required STC 45; however, when this requirement was increased in 1972, very few economical STC 50 constructions were available.

In response to these complaints, the Building Department granted a moratorium until April 30, 1973 for the code requirement of 0 INR (IIC 51) because of the present limited number of floor constructions that can meet this requirement. A similar moratorium is in effect for the requirement that apartment entrance doors meet an STC 35. The reason in this case is that there was also a one hour fire resistance requirement for apartment entrance doors and there were no door designs available that would meet both requirements. An industry association is presently having fire tests conducted on a specially designed STC 35 door.

Some manufacturers of air conditioning equipment have expressed difficulty in meeting with the sound power level limitations for exterior located mechanical equipment. These were written so as not to exceed an NC 40 in an apartment with an open window in direct line of sight of the mechanical equipment. As of 1972 these maximum permissible power levels have been reduced to achieve NC 35 levels inside the apartment. Again, the Building Department has relaxed the severity of these noise control provisions by temporarily exempting 3 ton and smaller through-the-wall air conditioning units from this requirement. Part of this problem of noise from exterior mechanical equipment seems to be a direct result of the building density in New York City. It is frequently almost impossible to use distance as an adequate attenuator for noise from even moderately noisy exterior located mechanical equipment because space is not available in small courtyards and on roof offsets. The only remaining methods of attenuation are power consuming silencers, or free standing noise barriers. Of course, with time, manufacturers' research efforts toward less noisy equipment will pay off.

To my knowledge, no criticism has been received of the vibration isolation specifications, nor has requirements for duct lining, partition and floor construction required for airborne isolation of noise from mechanical equipment located inside the same building or maximum permissible noise levels from fan coil units, grilles, registers, diffusers and the like been considered unworkable.

With the hope that it is a favorable sign, no complaints have been registered by tenants occupying the multiple dwellings that have been completed under the new code.

NOISE CONTROL PROVISION MODIFICATIONS

Since the code went into effect in 1971, the Building Department has made a number of administrative changes. Probably the most important of these has been to cease in the attempt to have building inspectors conduct field measurements except in the case of housing built by the New York City Housing Authority. Instead, the owner of the building must now engage the services of an approved acoustical laboratory to make the field measurements required by the noise control provisions. 25% of the apartments in a multiple dwelling must be measured and the Building Department must be notified of the precise location of the measurements and the date the measurements are to be made. The results of such field measurements shall not fail by more than 2 points as is noted in the code.

The Code's Reference Standards which were last updated in September 1967 have again been brought up to date incorporating recently conducted test data of partition and floor constructions and also the ISO R 1680 Test Code for the Measurement of the Airborne Noise Emitted by Rotating Electrical Machinery (1970) which will permit rating the sound power of very large equipment before it is installed and also permit making field measurements after the installation of large and small mechanical equipment.

There are some changes we, as writers of the noise control provisions, would like to see made to the code. One is a provision for control of plumbing noise. However, this must await a test procedure for laboratory and field measurement. This work is now under way in ISO/TC43/SC2 and ASTM E33.

Although, as of this writing, the requirements for vibration isolation of mechanical equipment seem to be accepted, I now believe this can be improved.

We would also like to change the procedure used for testing the impact noise isolation of floor constructions and update the Impact Noise Rating; however, since work is presently under way in both ISO/TC43 and ASTM E33 it may be wise to await the completion of these revisions to existing ISO and ASTM Standards.

Two years of usage are not sufficient to determine all the faults of a comprehensive pioneer code such as New York City's and we hope to continue to observe as to how it may be improved. In retrospect, however, this assignment has, during the last 10 years, led to many interesting experiences and concepts. In addition, it has reinforced our belief that the challenge imposed on the building industry by this and other codes has the beneficial side effect of encouraging research to produce better and quieter buildings.

REFERENCES

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3. Federal Housing Administration Bulletin No. 750 entitled, "Impact Noise Control in Multifamily Dwellings" prepared by Bolt, Beranek and Newman, Inc.

4. International Organization for Standardization Recommendation No. R140-1960 entitled, "Field and Laboratory Measurement of Airborne and Impact Sound Transmission".

5. L. L. Beranek, "Criteria for Office Quieting Based on Questionnaire Rating Studies", JASA, Vol. 28 No. 8 and "Revised Criteria for Noise in Buildings" Noise Control Vol. 3 No. 1.

6. American Society of Heating, Refrigerating and Air-Conditioning Engineers No. 36B-63 entitled, "Method of Testing the Acoustical Performance of Terminal Units".

EXPERIMENTAL EVALUATION OF A SIMPLE METHOD FOR ESTIMATING SOUND TRANSMISSION CLASS IN BUILDINGS

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INTRODUCTION

The simplified procedure for estimating Sound Transmission Class (STC) of wall in buildings, first suggested by Siekman, Yerges and Yerges (1), has been investigated under laboratory conditions. This simplified test procedure is intended to provide a simple and inexpensive test procedure for determining compliance with contract specifications or building codes. The standard field test procedure ASTM E 336-67T (2) is far too complicated and expensive for such applications, because a trained crew is required to take as many as 400 individual readings. A great need exists for a simple and accurate method to determine the STC of an existing partition in a building. The availability and standardization of such a test procedure can be expected to have a great impact on the degree of acoustical privacy built into new construction, because compliance with contract specifications and building codes could then be economically tested. It should be noted that the simplified test procedure is not intended to replace ASTM E 336-67T.

An experimental evaluation of the simplified test procedure of Siekman, Yerges and Yerges was conducted on three walls under laboratory conditions. Specifically, the following aspects of the simplified procedure were evaluated.

1. Validity and accuracy of using a single third point to determine the sound pressure level.
2. The capability of a single speaker at a third point (with and without the use of rotating vanes) to produce a diffuse sound field in the source room.
3. The overall accuracy of the procedure as compared to the results obtained from ASTM E 90-70 (3).

SIMPLIFIED TEST

The simplified procedure involves using the normalized difference in A-weighted sound pressure levels (SPL) between the two sides of the partition to be tested. The STC as estimated by the simplified method is given by

$$STC = L_s - L_r + 10 \log (S/A_r) \qquad (1)$$

where L_s is the A-weighted sound pressure level in the source room, L_r is the A-weighted SPL in the receiving room, S is the size of the wall separating the source and receiving rooms and A_r is the absorption of the receiving room. The values of L_s and L_r are each determined by single measurement made at a third point on any of the four long diagonals of each test room. The noise source is a single small speaker excited by pink noise and capable of producing over 100 dB(A) in a small room. The speaker is to be located at a third point of the source room and faced away from the test partition toward a corner of the room. See the paper by Siekman, Yerges and Yerges for more details of the simplified procedure.

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EXPERIMENTAL PROCEDURE

In order to eliminate certain complications present in field testing, the experimental evaluation was run under laboratory conditions using the facilities of the Riverbank Acoustical Laboratory. These facilities conform to the requirements of ASTM E 90-70. Detailed description of the test rooms is beyond the scope of this paper, however, both test rooms are large (approximately 6,000 cubic feet each), hard surfaced and have rotating diffusers. Isolation between the two test rooms is sufficient to permit measurement to STC 70. The edges of the test walls were carefully sealed to prevent leaks.

A single speaker with sufficient power capacity for the simplified test was not available. Alternately, four Jensen 8-inch (P8-PC) speakers were mounted in a square array and placed in a small acoustic enclosure. In some tests, the normal speaker arrangement at Riverbank of separate low and high frequency speakers mounted in the rear corners of the test room were also used. The single speaker was located at a lower rear third point and pointed toward the furthest lower rear corner of the receiving room.

The diffusers consisted of three glass fiber panels, two mounted at an angle in the top of the chamber and one mounted vertically near the bottom of the chamber. Several tests were run with the diffusers rotating and other tests with the diffusers not rotating and the lower vane removed (called no diffusers). Of the eight possible microphone positions in the receiving room, only six were used due to the presence of the non-rotating diffusers in the upper portion of the test room. In the source room five microphone positions were used, the six mentioned above less the position at the speaker location.

All SPL measurements were made to the nearest half dB with a B&K Type 2203 Precision Sound Level Meter. A 1-inch condenser microphone with a long cable connecting to the sound level meter was used.

THIRD POINT DATA

Three basic questions arise concerning the use of a single third point measurement to determine the sound pressure level. First, can statistically valid measurements be made at the third points, second is a single measurement sufficient to determine the SPL to a desired degree of precision, and third what is the effect of rotating vanes upon measurements at the third points. Each of these three questions will be considered separately, and all conclusions are contained in the section labeled Conclusions. The data in this section was obtained using the single third point speaker. The wall in place was the metal office wall, and except as noted there were no rotating diffusers.

In an attempt to determine whether statistically valid measurements can be made at the third points, a large number of A-weighted SPL measurements were made in both the source and receiving rooms. A total of 11 (5 third point and 6 other) and 13 (6 third point and 7 other) SPL measurements were made in the source and receiving room, respectively. The results of these measurements are summarized in Table I. The measurements labeled total involve the average of either 11 or 13 measurements. The statistical precision (90 percent confidence) of all averages was greater than or equal to $\pm$ 0.6 dB except for the average of the third point data in the receiving room where it was $\pm$ 0.8 dB.

Table I. Summary of A-weighted Sound Pressure Levels in dB.
Third point speaker and no diffusers.

ROOM	THIRD POINT			TOTAL		
	MIN	AVE	MAX	MIN	AVE	MAX
Source	107.5	108.1	108.5	107.5	108.3	109.
Receiving	77.	78.1	79.	76.5	78.3	79.

Next, the data in Table I can be used to determine the validity of using a single third point measurement to determine SPL. Since, according to Eq. 1 the difference in SPL is required, the validity will be evaluated in terms of differences. The smallest, average and largest differences of the third point data are 28.5 dB, 30 dB, and 31.5 dB, respectively. Thus, use of single SPL measurements will yield an estimate for STC with a statistical precision of no better than ± 1.5 dB.

Last, the effect of rotating diffusers upon the third point data was evaluated. In this case, no measurements could be made at the upper third points due to the rotating diffusers. Thus, only three and four measurements were made in the source and receiving rooms, respectively. The results of these measurements are summarized in Table II. The results for no diffusers are given in Table I.

Table II. Effect of Rotating Diffusers Upon Third Point A-Weighted Sound Pressure Level in dB. Third point speaker.

ROOM	MIN	AVE	MAX
Source	108.5	109.	109.5
Receiving	77.5	77.9	78.

SPEAKER AND DIFFUSERS

The use of a single speaker at a third point as required by the simplified procedure has been evaluated. The existence of a diffuse sound field in both the source and receiving rooms is necessary for the validity of the standard test procedures, and likewise for the simplified procedure. Direct evaluation of the degree to which sound field approaches a diffuse sound field is very difficult (4). For the purposes of this evaluation, two methods have been employed to evaluate the validity of using a single speaker at a third point, without a rotating diffuser, to produce the necessary diffuse sound field, or at least a sufficient approximation thereto. A characteristic of diffuse is a uniform SPL except near walls or other surfaces. The data of Table I indicates a rather uniform sound field in both the source and receiving rooms. While admittedly a uniform A-weighted SPL is generally not an accurate measure of the existence of a diffuse sound field, it is an indication of a sufficiently diffuse sound field for this application.

Also, the effect of speakers and rotating diffusers upon transmission losses, as measured using the standard laboratory test ASTM E 90-70, were evaluated. Tests were run on a metal office partition with different combinations of speakers and rotating diffusers or vanes. These tests are: first with the standard speakers and rotating diffusers, second with the standard speaker and without rotating diffusers, and third with a single speaker at a third point and without rotating diffusers. The transmission loss curves in third octaves for these three tests are given in Fig. 1. The STC ratings given in Fig. 1 were obtained using ASTM E 413-70T (5).

TRANSMISSION LOSSES

The STC of three walls was evaluated using the standard procedures of ASTM E 90-70 in conjunction with ASTM E 413-70T (5) and estimated using the simplified procedure. The standard procedure makes use of the standard speakers and rotating diffusers while the simplified procedure makes use of a single speaker at a third point and no diffusers. The results of these tests are summarized in Table III. The STC estimated by use of the simplified procedure is normalized, 10 log (S/A_r), using the absorption at 1,000 Hz. as specified by the simplified procedure. The one third octave normalized transmission loss curves and corresponding reference curves are given in Figs. 2, 3, and 4.

Table III. Results of Standard and Simplified Test Procedures

Wall	ASTM 90-70	Simplified Test
Metal office wall	STC 38	STC 34
Wood stud wall	STC 39	STC 34
Composite metal wall	STC 56	STC 55

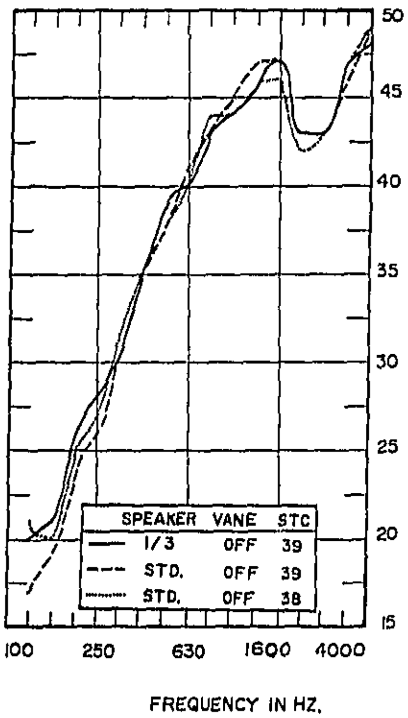

Fig. 1. Effect of speakers and rotating vanes upon transmission losses.

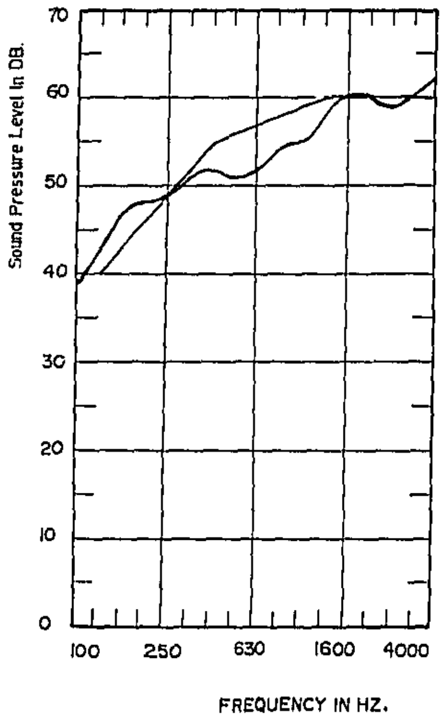

Fig. 2. Transmission loss curves for composite metal wall.

CONCLUSIONS

Several conclusions can be drawn based upon the results obtained. Great care must be exercised in the use of these conclusions as they are derived from the limited application of a proposed field test procedure in the laboratory. Several conclusions and associated discussions are listed below. In each case, the accuracy of the simplified procedure refers to its accuracy when compared to the results of a standard ASTM E 90-70 test.

1. Measurement of SPL at third points gives statistically valid results.

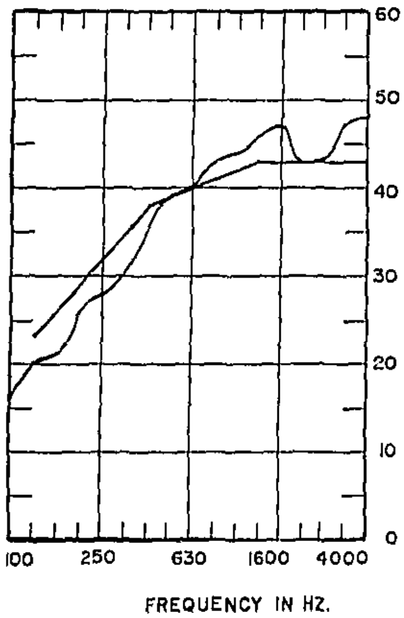

Fig. 3. Transmission loss for metal office wall.

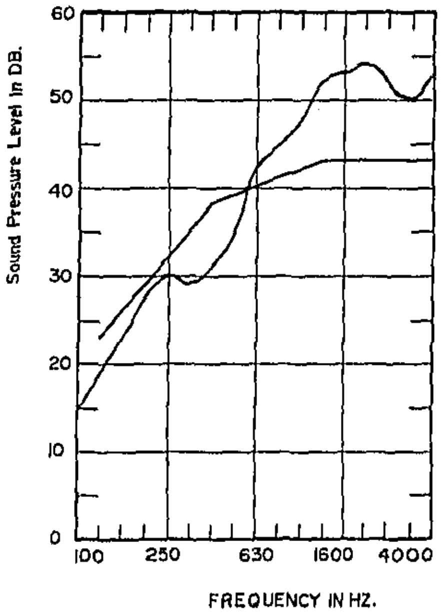

Fig. 4. Transmission loss for wood stud wall.

2. A single third point measurement gives results ($\pm$ 1.5 dB) that within the accuracy of $\pm$ 2 dB claimed (without supporting test data) by the authors for 95 percent of their tests. However, under field conditions, a single third point measurement cannot be expected to yield measurements with the above level of statistical precision.
3. The presence or absence of rotating diffusers appears to have little effect upon the results of A-weighted SPL measurements.
4. The use of a single speaker at a third point appears to produce a sound field sufficiently diffuse for the purposes of the simplified test procedure.
5. The accuracy of all tests of $\pm$ 6 dB claimed by the authors are supported by the data presented above.
6. The accuracy of $\pm$ 2 dB for 95 percent of partition constructions cannot be substantiated from the data presented above. In two of the cases presented above, the STC estimated by the simplified procedure was four or five dB low. In these two tests, two conditions occurred simultaneously. First, the partition had an STC of 38 or 39 and hence would be unacceptable for use in most constructions. Second, in each case, the breakdown (deviations below the reference curve) occurs at lower frequencies. Note also that large deviations of opposite sign can occur when a large coincidence dip exists.
7. Lower frequencies appear to be a problem. This is exhibited by the two cases mentioned above where lower frequency breakdown occurred. While the normal A-weighting curve is down 19 dB at 100 Hz., the A-weighting with pink noise is down only about 9 dB at 100 Hz. Thus, the results from the simplified procedure may be effected by low transmission losses at 100 Hz. and below. Tests to evaluate this effect were inconclusive.

RECOMMENDATIONS

Obviously, the most important recommendation is that further experimental evaluation, particularly in the field, of this and other simplified test procedures is urgently needed. Based on the results of the experiments reported, the following additions or modifications to the simplified test procedure are suggested.

1. In each test room, three third point A-weighted SPL measurements be made if the difference between the largest and smallest is 6 dB or less. If this difference is 7 dB or more, then five third point measurements are to be made.
2. To avoid shielding of a microphone by an inexperienced operator, all measurements are to be made using a tripod and a ten-foot extension cable between the microphone and sound level meter.
3. A high pass filter with a cutoff frequency of about 100 to 112 Hz. may be needed, depending upon results of future investigations on the effect of high pass filters as described in the seventh conclusion.
4. While the simplified procedure does provide an estimate of STC, the procedure actually measures the Noise Isolation Class (NIC), as defined by ASTM E 336-67T. STC applies only to the transmission of a single partition with no flanking, while the NIC gives a rating of the normalized noise reduction between two rooms. The results of the simplified procedure should be labeled as NIC (6).

ACKNOWLEDGEMENT

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THE MEASUREMENT OF ACOUSTICAL FLANKING IN BUILDINGS

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INTRODUCTION

It is well known that the transmission loss performance of a sound barrier measured in the field is very often substantially inferior to its measured laboratory value, (1-3). This is true for both walls and floors. The primary reason for this lack of performance is known as flanking transmission by which acoustical energy is able to bypass or 'flank' the primary barrier. This may take place in two distinct ways. Energy may be able to 'leak' through a hole or crack around the edge of the barrier giving rise to acoustical flanking, while energy may also be propagated along other structures joining the two rooms separated by the partition (for example, continuous side walls or floor structures) and then be re-radiated on the other side of the barrier. This type of transmission is known as structural flanking. Studies of the field performance of sound barriers have shown (2) that in most cases one really measures the performance of the flanking paths rather than the sound rating of its dividing structure in typical room-to-room tests. However, methods for the determination of such flanking transmission are, to date, somewhat clumsy and time consuming and usually involve the erection of a second barrier covering the partition being evaluated. Furthermore, this method does not usually give anyinformation about individual flanking paths if more than one exist.

Recently correlation techniques have been applied to several acoustical measurements (4) including the determination of absorption coefficients (5), radiated sound power (6), wave propagation in structures (7) and more recently to measure the sound transmission loss of a partition (8).

This paper describes the application of a correlation technique to measure the characteristics and importance of the acoustical flanking paths existing between two rooms separated by a partition.

THEORY OF MEASUREMENT

The cross correlation of two different time varying signals $x(t)$ and $y(t)$ is defined by;

$$r_{xy}(\tau) = \lim_{T\to\infty}\frac{1}{2T}\int_{-T}^{T} x(t)\ y(t+\tau)\ dt, \qquad (1)$$

where τ is the time delay imposed on $x(t)$ before it is multiplied by $y(t)$.

If $x(t) = y(t)$ then the result of the above integral yields the auto correlation function r_{xx}. Upon taking the Fourier transform of Equation (1), and denoting frequency functions by capitals, the following result is obtained.

$$R_{xy}(\omega) = X^{*}(\omega)\ Y(\omega), \qquad (2)$$

Putting $Y(\omega) = H(\omega)\ X(\omega)$ gives

$$R_{xy}(\omega) = X^{*}(\omega)\ H(\omega)\ X(\omega), \tag{3}$$

where $H(\omega)$ is the frequency response of the system and $X^{*}(\omega)$ is the complex conjugate. This also follows from the fact that if a white noise signal $x(t)$ is applied to a linear system and then cross correlated with the output $y(t)$ the result is the impulse response of the system $h(t)$, i.e., $r_{xy}(t) = h(t)$. Taking the transform yields $R_{xy} = F[h(t)] = H(\omega)$. This is true only for a uniform infinite bandwidth source. A correction for the source can be made by considering the transform of the auto correlation of the source signal;

$$R_{xx}(\omega) = X^{*}(\omega)\ X(\omega). \tag{4}$$

Therefore the system frequency response may be determined from the ratio of the Fourier transforms of the cross and auto correlation functions as follows:

$$H(\omega) = R_{xy}(\omega)/R_{xx}(\omega). \tag{5}$$

Schomer (8) shows that a better correction for the source spectrum variations may be made by repeating the cross correlation in the absence of the system under test. This however, is not always possible in real situations. In the case of determining the sound transmission loss of a partition between two rooms a source of white noise is created on one side of the partition and monitored by two microphones, one in each room. The microphone close to the source yields the input signal to the partition $x(t)$ while the other microphone gives the 'response' signal $y(t)$. By measuring both $R_{xy}(\omega)$ and $R_{xx}(\omega)$ and using equation (5) the frequency spectrum of the acoustical power transfer between the two microphones (i.e., cross power spectrum) can be evaluated. This includes both the direct path and possible flanking paths. However, by proper truncation of the cross correlation function the cross power spectra of chosen paths can often be evaluated. In real rooms, however, it is often difficult to obtain a smooth correlation function with clearly identifiable peaks due to the many reflected paths within the rooms and to the preferential exciation of room modes. If one is only concerned with the relative importance of the flanking paths as compared to the direct transmission then it is sufficient to eliminate the source microphone and take $x(t)$ as the input to the acoustical source. A correction for the frequency response of the source may then be later applied to the resulting cross power spectra so obtained. It is found that this method of using only one microphone gives a more useful correlogram in real rooms which can be more easily truncated and is ideally suited for the determination of the relative importance of flanking transmission.

EXPERIMENTAL SET-UP

A pair of model rooms were constructed using 1" thick plywood as shown in Figure 1. These had volumes of 65. and 49.2 cubic feet respectively. The dividing partition was constructed from 1/8" thick aluminum and rigidly clamped around its edges by an aluminum frame. To eliminate the effect of short delay reflections and room modes, both rooms were lined with 2" thick fiberglass. The whole structure was vibration isolated from the supporting floor by six neoprene isolators placed around the perimeter.

The acoustical source used in these measurements was a University Sound ID-60 trumpet driver whose frequency response had been previously measured in an anechoic room. A receiving microphone was placed in Room 2 at 7" from the partition and 24.5" from the horn driver and on its axis. As previously stated, it was found that better correlograms could be obtained by directly taking the input signal to the horn driver as the $x(t)$ signal for correlator, rather than by the use of a separate source microphone. The microphone signal from Room 2 was amplified and fed directly into a PAR 101A correlator as $y(t)$. Each of the signals were then sampled at intervals of 10 μsec and the correlogram plotted by an X-Y plotter. The resulting correlator output was then digitized and fed into a computer where the desired Fourier transforms were calculated using an FFT technique.

RESULTS

Figure 2 shows the measured normal incidence transmission loss of the dividing 1/8" aluminum panel with the partition well sealed all around its perimeter by closed cell neoprene tape and caulking compound. The measured values agree well with the normal incidence mass law and would therefore tend to indicate only a small amount of acoustical flanking, if any.

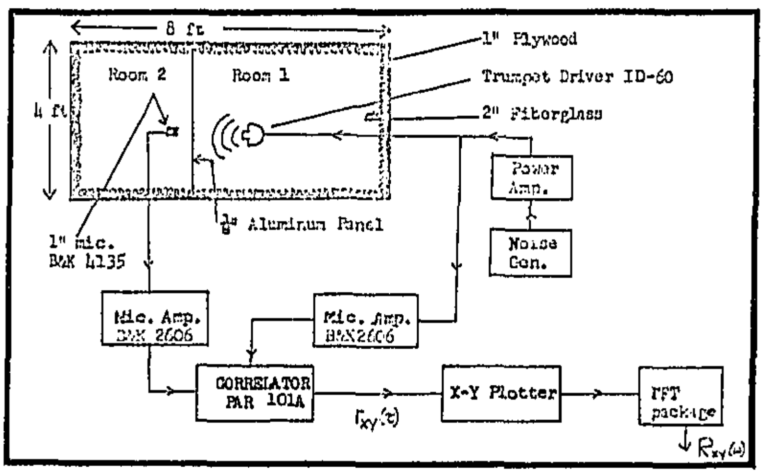


FIGURE 1. Experimental model and equipment layout.

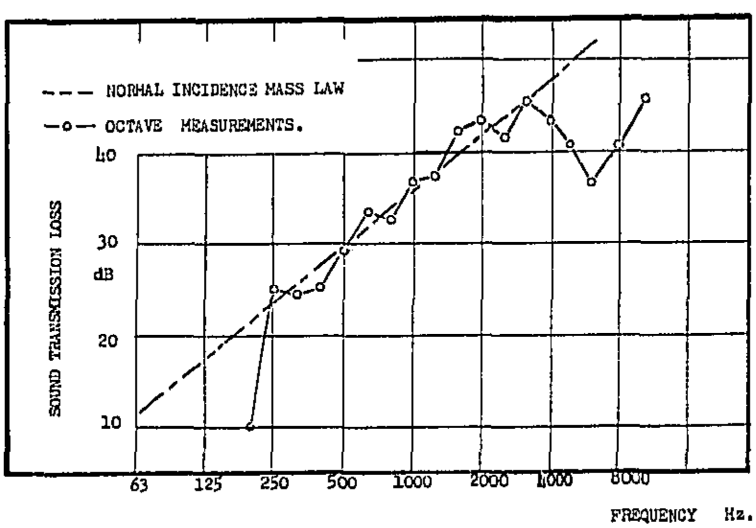


FIGURE 2. Measured transmission loss of dividing partition.

The corresponding cross correlation function for τ = 0 to 8 msec obtained between the receiving room microphone and the input to the horn driver is shown in Figure 3a. The peak at 2.3 msec corresponds to the direct transmission path through the partition (allowing for a 0.3 msec delay due to the horn driver response) while the remaining portion of the correlogram contributes additional energy into the reception room via other paths. No definite flanking paths could be easily identified from this correlogram. In order to demonstrate the method previously described a known small acoustical flanking path was then introduced into the system. This was achieved by removing a 7" length of the sealing neoprene tape from one edge of the panel clamping frame. This resulted in a small air gap 7" long, approximately 1/16" wide and 1.7" deep. The resulting cross correlation function is shown in Figure 3b. Once again the direct transmission peak is clearly evident at 2.3 msec. However, a strong peak can also be seen at 5.3 msec. This time delay corresponds exactly to the acoustical time delay between the horn driver and the microphone via the introduced edge flanking path. The subsequent decaying peaks are thought to be due to a tone burst effect at the narrow slit forming the flanking path. The frequency of the tone burst is seen to be 3400 Hz and this corresponds fairly well with the half wavelength frequency of the flanking slit at 3900 Hz.

FIGURE 3

Cross Correlations before and after flanking

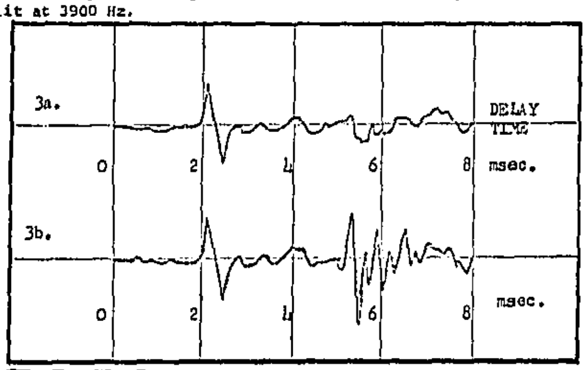
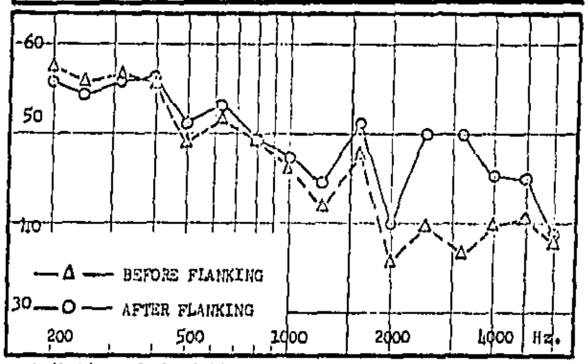


FIGURE 4

Normalized cross power spectra before and after flanking.

dB re 10^{-2} volts.



In order to determine the contribution of the deliberately introduced flanking, cross power spectra were evaluated from each of the correlation functions shown in Figure 3 for τ = 0 to 8 msec. In each case the correlogram was digitized into 800 points using a Gradicon graphical digitizer. An additional number of zero value data points were then appended to this set to give a total of 2048 (i.e., 2^{11}) data points in all. This gives an upper frequency limit of 12,500 Hz to the Fourier transform. The discrete transform was then computed using an FFT package developed by Rackl (9). For such a data set the resulting bandwidth is given by:

$$\Delta f = (2N.\Delta T)^{-1}, \text{ Hz} \quad (6)$$

where N is the number of data points and ΔT the sampling period. For the present case $\Delta f = (2 \times 2^{11} \times 10^{-5})^{-1} = 25$ Hz and the Fourier transform therefore gives a very narrow band cross power spectra and subsequently tends to be rather resonant in appearance. In order to facilitate a better comparison, the resulting power spectra were averaged over 1/3-octave bands. These averaged spectra are shown in Figure 4 and give the relative power contribution before and after the introduction of the flanking path.

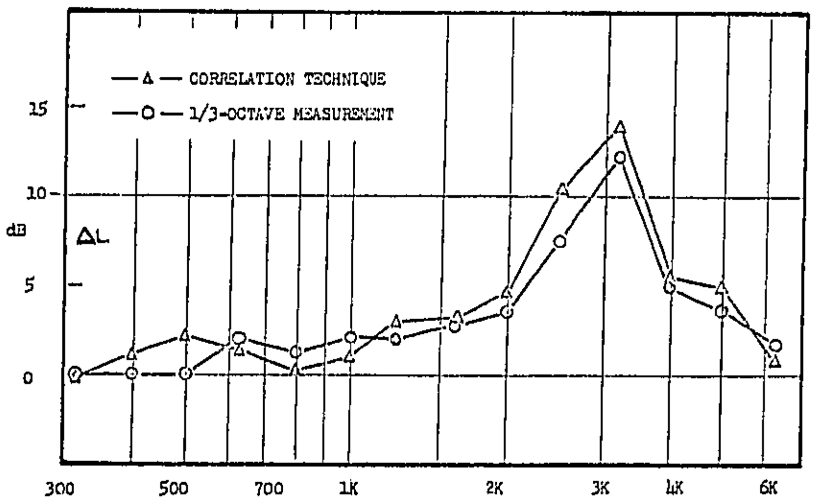


FIGURE 5. Increase in sound pressure level ΔL in Room 2 due to introduced flanking path.

The difference between these two spectra gives the expected increase in sound level due to the deliberately introduced flanking. This difference is also shown in Figure 5 and compared to the actual measured increase in sound pressure levels recorded in 1/3-octaves in the receiving room. There appears to be good agreement between the curve obtained from conventional measurement and the data obtained from the Fourier transforms of the cross correlation functions.

ACKNOWLEDGEMENTS

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THE EFFECTIVENESS OF BARRIERS UNDER EXTENDED CEILINGS

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SUMMARY

Under extended ceilings, sound propagates with noticeably less reduction with distance than predicted by spherical spreading loss. This is caused by build up of reflections from the floor. At great distances the build up can be as much as 11 db above the spherical spreading prediction. With the presence of walls there is an additional build up of reflections which can add another db or two at great distances. In the cases where it is desired to use a partial barrier to reduce the noise in selected areas, the reflections from the walls and ceiling reduce or eliminate the shadowing effect at great distances. The shadowing obtained by barriers appears nearly independent of size except in the region close to the barrier. The addition of absorbent ceilings and walls greatly increases the shadowing. Wall and ceiling reflections completely eliminate any advantage to use of multiple barriers. Partial enclosures benefit more from the use of absorbent room ceilings than from roofs on the enclosure.

ANALYSIS TECHNIQUE

The data for this paper was generated by a computer program written to answer the question of the effectiveness of meeting OSHA requirements with alternatives to total enclosures. The algorithm used was a three dimensional sound image technique illustrated in Figure One.

Figure One shows the first four images in the ground plane for the X direction and the first two images for the Y direction. The wall labeled A in the X direction, is repeated alternately with the wall B. Effectively the original floor is flipped over repeatedly to produce the images. The distance the sound ray travels to the image of the receiver is calculated by the difference of the coordinates of the points in the X, Y plane. As a ray is reflected from a wall, appearing to pass through in the diagram, it is reduced by the reflection coefficient. By adding the number and types of walls the ray encounters accounts for the reduction by absorption. The total picture is constructed by repeating the image process in three dimensions. In the example of Figure One even the first few orders of image produce an extremely complicated ray path. By adding a section to detect the presence of a barrier image on the ray path, the contribution of rays blocked by barriers is eliminated. This assumes the barriers wholly absorb reflections. In most cases the additional path length the barrier reflections would have to travel to the receiver would make their contribution insignificant. The wave diffracted over the top of the barrier in the first order image is in many cases significant. The first order diffracted wave contribution was calculated using the method given in Beranek's Noise and Vibration Control, Page 175. It was assumed that in most circumstances the barrier height would be the limiting factor and that width could be adjusted for the maximum effectiveness. A secondary computer investigation found that where the barrier width was four or more times wider than the vertical intersection height the barrier side diffractions were insignificant. Diffraction around the barrier images was assumed to be insignificant with respect to the contribution of the direct rays.

ACCURACY VERIFICATION

With a complex program it is a problem to determine whether it is working properly and if so whether the assumptions reduce the accuracy of results. The program was checked for function by hand calculation of a test situation for the first two images. The first two images give a three dimensional matrix containing 64 sections. Hand calculations of the non interfering cases agreed exactly with the computer prediction proving the functioning was correct. The number of reflections could conceivably be infinite so it was necessary to provide a logical cutoff to the calculations. By comparison to Hale Sabine's formula for the sound propagation under extended ceilings, a cutoff was found to be the elimination of reflections that contribute less than 0.1 db to the total sum. By extending the calculations the worst case error was 1 db low for highly reflecting walls.

The accuracy of the assumptions was tested by simulating a field test run for Owens-Corning Fiberglass by Geiger Hamme. The field test took place in a 15'x30'x8'-10" high test chamber completely divided at the 10 foot distance in the 30 foot direction by a 5 foot barrier. The walls and barrier were heavily draped to produce high absorption. The floor was carpeted and the ceilings consisted of standard acoustic tile. The source

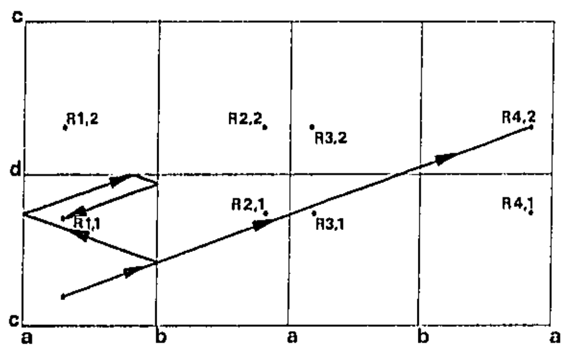

figure one:
diagram of image technique

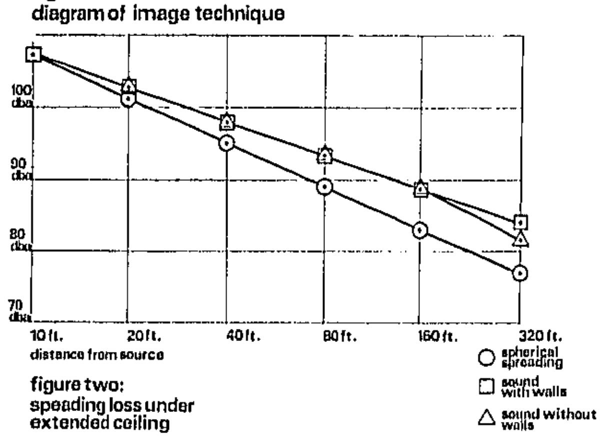

figure two:
speading loss under
extended ceiling

was located 6 feet from the barrier with receivers at 9,12,15,and 18 feet from the source on the opposite side of the barrier. Both the source and receivers were 4 feet above the carpet. Where the absorption coefficients of the materials were not available from Owens-Corning they were estimated from the literature. Comparison of the computed predictions with the test measurements in the third octaves from 500 to 2000 Hertz found the predictions to average 0.7 db high with a range of plus and minus 3 db and a standard deviation of 1.46 db. Since the program did not take into account the variation of material absorption with angle this was considered to be sufficient accuracy.

RESULTS AND DISCUSSION

Our client's 440'x540'x38' high factory floor was used for this work. The sound source consisted of chipping operation noise at a level of 117 dbA at 3 feet. The source was located 200 feet from the wall in the 440 foot direction and 120 feet from the wall in the 540 foot direction. The height of the source was 3 feet with the receivers at 5 feet. The computer output consisted of the direct, refracted, and reflected components of the noise in addition to the total sum. The reported results are in dbA. As a design aid various combinations of wall and floor ceiling absorption coefficients were run. These tables are available from the author, however the application may be limited in rooms of different dimensions.

Figure Two shows the noise drop off with distance in the bare room. At the 320 foot distance the reflection build up would be 8 dbA above that expected for simple spherical spreading loss. To determine the effect of the walls and the possible benefits of adding absorption to the walls a second run was made with no wall reflections. As can be seen the wall reflections only effect the noise at distances of 160 feet or greater, amounting to 2 dbA at 320 feet. The absorption coefficients used for this case and the rest to be discussed unless stated otherwise, were those for Galbestos walls, type S metal roof decking, and 0.1 to 0.2 for the floor. The latter was used as an estimate of the effect of machine clutter.

Figure Three is a breakdown of the components of the room noise. The direct sound was not shown but it can be seen that from about one ceiling height out the floor and ceiling reflections are dominant. The lower curve for the wall reflection component is almost independent of distance from the source. Hand calculations of a simple geometric nature confirmed that the wide floor area does indeed produce this result where the source is far from the walls. When the source is close to a wall the near in wall reflection strength approaches the direct component. At all positions of the source the 320 foot reflection strength remains at about 83 dbA.

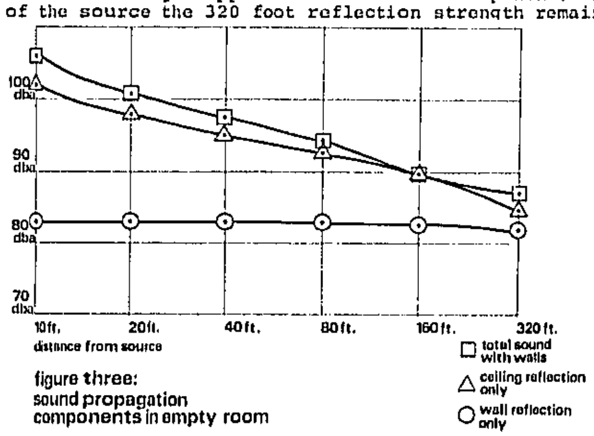

figure three:
sound propagation
components in empty room

Figure Four shows the shadow effect of four different size barriers placed at 5 feet from the source. Examination of the effect of barrier placement found that the maximum reduction occurred as the barrier was placed closer to either the source or receiver. The greatest area shadowed occurred with the barrier close to the source. All subsequent analysis was made on this basis. As shown in Figure Four the largest barrier was most effective. Going from a 10 to 20 foot barrier had the greatest barrier to barrier improvement. From the analysis it would appear that on a cost basis a 20 foot barrier is most efficient.

Figure Five shows the components of sound behind a 10 foot barrier. The sound diffracted over the barriers is an insignificant part of the total. The sound from the ceiling and floor is dominant out to about one ceiling height, beyond that the wall reflections become dominant. It would appear that the addition of absorption to the ceiling would be very effective out to one ceiling height with absorption on the walls necessary to reduce the level at great distances.

The lower curve on Figure Six shows the effect of ceiling absorption. There is a fairly uniform reduction in level. Close in the diffracted wave is important. At middle distances the reduction is greatest since the floor ceiling reflections are dominant, and at great distances the wall reflections produce a lower limit. With gains of 4 to 6 dbA it would appear worthwhile to add ceiling absorption. This example is very pessimistic in that the absorption coefficients for a Type H-F acoustideck were used and which had its maximum absorption in the 500 to 1 KH Hertz octaves whereas the source had its maximum strength in the 2 and 4 KH Hertz bands. A better match would produce more dramatic noise reduction.

The effectiveness of multiple barriers under acoustic ceilings is illustrated in Figure Seven. Figure Seven shows the noise behind three barriers located at 5, 35, and 155 feet. With an ordinary ceiling the levels are comparable to the effect of one barrier at 5 feet. With the acoustic ceiling the levels from 40 feet out drop dramatically indicating that the floor ceiling reflections are highly important.

Figure Eight shows the components of the sound behind the multiple barriers. Because of the position of the source the wall reflections are more nearly constant with distance. The floor ceiling reflections drop off dramatically behind the second barrier at 35 feet. A logarithmic sum made of the wall and ceiling reflection does not add up to the total at great distances, due to the rays being reflected in a cork screw fashion from both walls and ceiling floor.

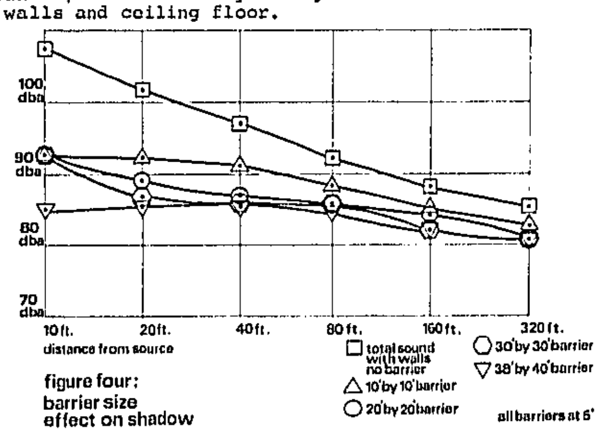

figure four:
barrier size
effect on shadow

Figure Nine shows the effects of a booth. A three sided booth with no roof is only 2 dbA more effective than a 10 foot barrier. The addition of an acoustic ceiling gains as much as 10 dbA appearing beyond one ceiling height. Addition of a booth roof in the presence of an ordinary ceiling is noticeably less effective than an acoustic ceiling except where the refracted sound is dominant.

CONCLUSIONS

The performance of a barrier under an extended ceiling is much less than predicted from simple diffraction theory. The floor ceiling, and wall reflections produce in various combinations the lower limit of performance. The addition of absorption to the wall and ceiling can produce dramatic improvements to the barrier performance. Multiple barriers in a hard room are only slightly more effective than a single barrier close to the source. Addition of absorption to the ceiling will produce a great improvement in the middle distances and wall treatment reduces the limit at great distances. The use of a booth without a roof is highly effective with an area acoustic ceiling but in the presence of ordinary ceilings a booth roof is necessary to obtain noticable improvements over a simple barrier. Wall treatment would have a great effect. Placement of a barrier in front of the open side of the booth would be more effective than ceiling treatment when the booth has a roof.

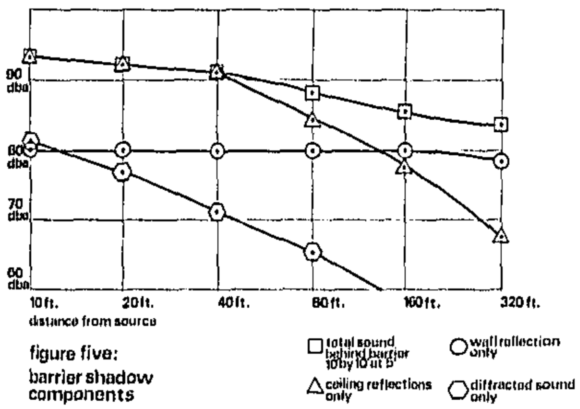

figure five:
barrier shadow components

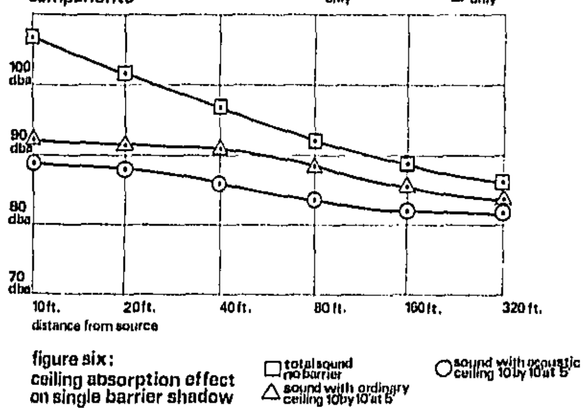

figure six:
ceiling absorption effect
on single barrier shadow

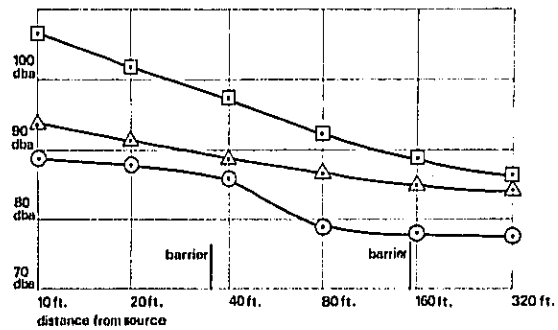

figure seven:
ceiling absorption effect on
multiple barrier shadow

□ total sound no barriers
three barriers 10by10' at 5, 35, 155' with:
△ ordinary ceiling
○ acoustic ceiling

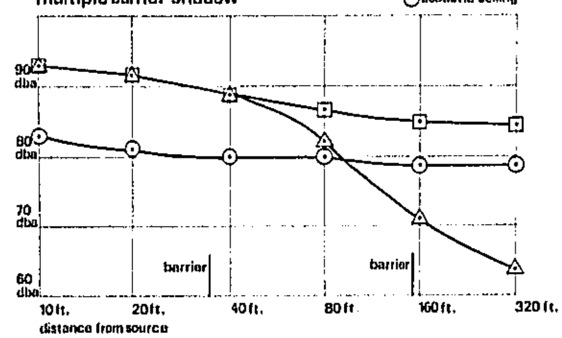

figure eight:
multiple barrier
shadow components

□ total sound behind barrier 10'by10' at 5, 35, 155'
△ ceiling reflections only, ordinary ceiling
○ wall reflections only

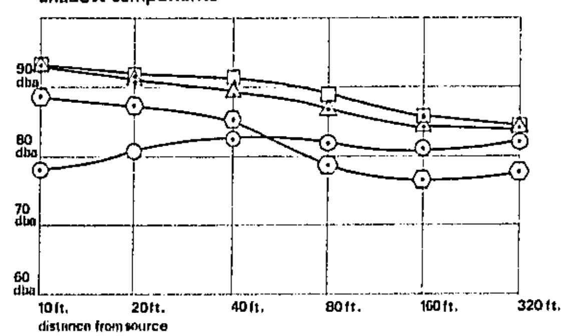

figure nine:
partial enclosure
shadow

□ single 10'by10' barrier at 5'
△ three sided booth no roof ordinary ceiling
○ three sided booth with roof ordinary ceiling
⬡ three sided booth no roof acoustic ceiling

THE PERFORMANCE OF ACOUSTIC BARRIERS

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ABSTRACT

The insertion loss provided by interposing an acoustical barrier of infinite lateral extent between a sound source and a receiver under free field conditions has been discussed by several authors.[2-4] Recently, Tatge has obtained relatively simple and useful formulas for predicting the insertion loss of such barriers when under free field conditions. This paper extends the work of Tatge to include the effects of (a) finite-sized barriers and (b) room reverberation on the insertion loss. A novel method for obtaining the room absorption in the field is also presented.

1. INTRODUCTION

Several years ago, the system concept was introduced as a generalized approach to solving a noise control problem.[1] In principle, this concept stated that the reduction of noise from a source could be effected by either reducing the noise at the source, along the path of propagation, or at the listener. To the noise control engineer, however, the only means opened to solving the great majority of industrial noise control problems is to modify the "path", as for example by completely enclosing the noisy machine, by adding acoustical absorption to the room in which the machine is located, by resiliently coupling the machine to its support and attached structures or by erecting a wall between the machine and the area where noise reduction is desired. The latter approach, namely, the control of industrial noise by acoustic barriers, will be dealt with in this paper.

The insertion loss (the amount by which the sound pressure level at a specified point is reduced) achieved by using a barrier results from the shadow cast by the barrier. The intensity inside the shadow zone results from the diffracted waves from the edges of the barrier. Historically, the exact solutions to the problems of diffraction on a semi-infinite plane due to an incident plane wave and an incident cylindrical wave have been obtained by Sommerfeld[2] and Macdonald[3], respectively. Later, Redfearn[4] obtained an approximate solution for the problem when the incident wave is spherical. All three solutions are expressed in terms of Fresnel Integrals. Redfearn's paper points out that acoustical absorption on the source side of the barrier has a negligible effect on the noise reduction. Our experimental results support this prediction in a free field.

While most of the published theoretical work heretofore has been addressed to the problem of diffraction over a semi-infinite barrier in a free field, a few studies have considered special cases for finite-sized barriers. For example, Sommerfeld[5] has obtained a solution for the field on the central axis behind a circular disc which predicts that "There is no darkness anywhere along the central perpendicular behind an opaque circular disc except immediately behind the disc." More recently, Maekawa[6] has suggested that the noise reduction provided by finite-sized rectangular barriers can be obtained by summing the diffracted field from each edge of the barrier where the diffracted field from any edge of the barrier is considered to be the same field as would be diffracted if that edge were infinitely long. This approach, however, breaks down when barriers other than those of rectangular shape are considered. To the writers' knowledge, no one has yet obtained a general solution for the noise reduction provided by screen of arbitrary shape.

The use of barriers to control machine noise in factories presents some special problems. For example, one would expect that, due to reflection from other surfaces in the room, the noise reduction in closed spaces would be less than that in the free field. It is the aim of this paper to consider this aspect of the barrier problem, namely, the effect of room reverberation.

2. THEORY

Our approach to the problem of finding the insertion loss due to a barrier basically considers the mean squared sound pressure at the receiver location to be sum of the mean squared sound pressures from the reverberant field plus the diffraction field from the barrier. The justification for this approach lies in the fact that most industrial noise sources are incoherent. Sound transmission through the barrier is assumed negligible.

2.1 Mean Squared Pressure Before Inserting the Barrier

According to Embleton[7] the mean squared sound pressure at the observation point before inserting the barrier can be expressed approximately as

$$|p_o|^2 = \rho c W \left(\frac{Q}{4\pi r^2} + \frac{4}{S_o \alpha_o}\right) \quad (1)$$

where ρc = characteristic impedance of the air
W = source sound power
Q = directivity factor
r = distance from the source to the receiver location
α_o = mean sabine absorption coefficient of the room
S_o = total room surface area

The two terms inside the parentheses represent the sum of the direct and reverberant fields.

2.2 Mean Squared Pressure After Inserting the Barrier

After inserting the barrier the total mean squared sound pressure at the receiver location can be written

$$|p|^2 = |p_r|^2 + |p_d|^2 \quad (2)$$

where $|p_r|^2$ = mean squared sound pressure of the diffuse field
$|p_d|^2$ = mean squared sound pressure of the diffracted field

2.2.1 Mean Squared Pressure In the Diffuse Field

After the barrier is inserted, the mean squared sound pressure at the point of observation due to the diffuse field can be calculated by considering that the barrier divides the original room into two rooms which are acoustically coupled by the reverberant energy which passes over and around the barrier perimeter (Figure 1).

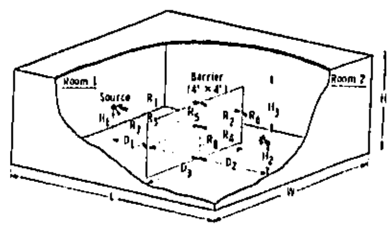

Figure 1—Acoustically coupled rooms formed by the insertion of a barrier

Similar approaches have been used by Bolt and Ingard[1] and Mankovsky.[8] Assuming that the sound source radiates constant power, then the energy balance for the source room requires that

$$W - W_1 - W_2 + W_3 = 0 \quad (3)$$

where W = sound power radiated by the source
W_1 = sound power absorbed in the source room
W_2 = sound power reflected from the source room to the receiver room
W_3 = sound power reflected from the receiver room back to the source room

According to diffuse room theory, the dissipated and reflected powers are related to the relevant mean squared sound pressure by

$$W_1 = |p_1|^2 \frac{S_1 \alpha_1}{4\rho c} \quad (3a)$$

$$W_2 = |p_1|^2 \frac{S}{4\rho c} \quad (3b)$$

$$W_3 = |p_2|^2 \frac{S}{4\rho c} \quad (3c)$$

where $|p_1|^2$ = mean squared sound pressure in the source room
α_1 = mean absorption coefficient for the source room
S_1 = surface area in the source room
$|p_2|^2$ = mean squared sound pressure in the receiver room
S = open area between the barrier perimeter and the room walls floor and ceiling. (In Figure 1, $S = HW - D_3H_3$.)

When these values are substituted into Equation (3) the following results:

$$4\rho cW = |p_1|^2 (S_1\alpha_1 + S) - |p_2|^2 S \quad (4)$$

Similarly, the energy balance for the receiver room requires that

$$W_4 + W_3 - W_2 = 0 \quad (5)$$

where
W_4 = sound power absorbed in the receiver room

According to diffuse room theory, W_4 can be written as

$$W_4 = |p_2|^2 \frac{S_2\alpha_2}{4\rho c} \quad (5a)$$

where

α_2 = mean absorption coefficient for the receiver room
S_2 = surface area for the receiver room

After substituting Equations (5a), (3b), and (3c) into Equation (5), the mean squared sound pressure in the source and receiver rooms are seen to be related by

$$|p_1|^2 = |p_2|^2 \left(\frac{S_2\alpha_2 + S}{S}\right) \quad (6)$$

Substituting Equation (6) into Equation (4) and rearranging terms, one finds that the mean squared diffuse sound pressure in the receiver room is

$$|p_2|^2 = 4\rho cW \frac{K_1K_2}{S(1 - K_1K_2)} \quad (7)$$

where

$$K_1 = \frac{S}{S_1\alpha_1 + S} \quad (7a)$$

$$K_2 = \frac{S}{S_2\alpha_2 + S} \quad (7b)$$

2.2.2 Mean Squared Pressure Due to Diffraction

Barriers having lengths which are much greater than the barrier height (i.e., $D_3 >> H_3$ in Figure 1) can, for all practical purposes be considered as infinitely long. The insertion loss obtained from free field measurements or such barriers can be compared directly with theories which deal with diffraction by a semi-infinite screen as, for example, the approximate theories of Redfearn(4) and Kirchhoff(9). Maekawa(6) has performed extensive measurements on the insertion loss provided by such barriers. Although the theories of Redfearn and Kirchhoff fitted Maekawa's data reasonably well, the use of these theories to compute the insertion loss involves Fresnel integrals which, although tabulated, are not readily familiar to the typical engineer who is faced with the solution of an industrial noise control problem. Moreover, the polynominal approximations to these integrals are somewhat cumbersome. Tatge(10) has suggested that the mean squared sound pressure at the receiver location, as depicted by Maekawa's data, can be represented by a simple expression, namely

$$|p_d|^2 = |p_f|^2 \frac{1}{3 + 20\, N_1} \quad (8)$$

where

$|p_d|^2$ = mean squared sound pressure at the receiver location due to diffraction over the top of the barrier
$|p_f|^2$ = free field mean squared sound pressure at the receiver location before inserting the barrier
N_1 = Fresnel Number for the diffraction path over the top of the barrier

According to Tatge, Wells has obtained a slightly different expression for the mean-squared diffracted sound pressure which is approximately

$$|p_d|^2 = |p_f|^2 \frac{1}{1 + 10\, N_1} \quad (8a)$$

where the terms have the same meaning as in Equation (8).

The free field mean squared sound pressure before inserting the barrier is related to the source power, W, by

$$|p_f|^2 = \frac{\rho cWQ}{4\pi r^2} \quad (8b)$$

where ρc, Q, and r have the same meaning as in Equation (1).

The Fresnel Number is given by

$$N_1 = \frac{2\delta_1}{\lambda}$$

where λ is the wavelength and δ_1 is the difference between the path over the barrier top and the direct path. Using the nomenclature of Figure 1,

$$\delta_1 = R_1 + R_2 - R_3 - R_4.$$

For finite sized rectangular barriers, the mean squared sound pressure at the receiver location can be estimated by summing the intensity diffracted from each edge. For the barrier of Figure 1, diffraction occurs at three edges (i.e. the top and sides). Using Rathe's(11) expression for the diffraction at each edge, the mean squared sound pressure at the receiver location is

$$|p_d|^2 = |p_f|^2 \left[\frac{1}{3 + 20\ N_1} + \frac{1}{3 + 20\ N_2} + \frac{1}{3 + 20\ N_3} \right] \qquad (9)$$

where N_1, N_2, and N_3 are the Fresnel Numbers. ($N_1 = 2\delta_1/\lambda$, $N_2 = 2\delta_2/\lambda$ and $N_3 = 2\delta_3/\lambda$).[1] Again, from Figure 1:

$\delta_2 = R_5 + R_6 - R_3 - R_4$ and

$\delta_3 = R_7 + R_8 - R_3 - R_4$

In general, the mean square sound pressure due to diffraction under free field conditions can be written as:

$$|p_d|^2 = \frac{\rho c W Q D}{4\pi r^2} \qquad (10)$$

and the insertion loss is

$$\text{I.L.} = 10 \log_{10}(D) \ \ \text{dB} \qquad (10a)$$

where D is the diffraction coefficient. According to Rathe's representation,

$$D = \Sigma \frac{1}{3 + 20\ N_i} \qquad (10b)$$

and according to the Wells' approximation,

$$D = \Sigma \frac{1}{1 + 10\ N_i} \qquad (10c)$$

where N_i is the Fresnel Number for the particular path in question.

3. INSERTION LOSS IN A SEMI REVERBERANT ROOM

After inserting the barrier, the total mean squared sound pressure at the receiver location is, according to Equations (2), (7), and (10),

$$|p|^2 = \rho c W \left[\frac{QD}{4\pi r^2} + \frac{4\ K_1 K_2}{S(1 - K_1 K_2)}\right] \qquad (11)$$

The insertion loss is defined as $10 \log_{10} (|p/p_o|^2)$ and, according to Equation (1) and (11) is

$$\text{I.L.} = 10 \log_{10} \left[\frac{\dfrac{QD}{4\pi r^2} + \dfrac{4\ K_1 K_2}{S(1 - K_1 K_2)}}{\dfrac{Q}{4\pi r^2} + \dfrac{4}{S_o \alpha_o}} \right] \ \text{dB} \qquad (12)$$

It is of interest to consider some special cases of Equation (12).

Case 1

Both rooms are nearly perfectly absorbing before and after inserting the barrier.

In this case, particularly when S is much smaller than S_1 and S_2, then $K_1 = K_2 \simeq 0$ and $S_o\alpha_o >> 4$ so that the insertion loss becomes

$$\text{I.L.} \simeq 10 \log_{10}(D) \ \ \text{dB} \qquad (12a)$$

which is to be expected on the basis of Equation (10b).

Case 2

Both rooms are nearly perfectly reflecting before and after inserting the barrier.

In this case, $K_1 = K_2 \simeq 1$ and $S_o \alpha_o \simeq 0$ so that the second terms inside the brackets in both the numerator and denominator become very large compared to the first terms. The insertion loss becomes

$$\text{I.L.} \simeq 10 \log (1) \simeq 0 \ \text{dB} \qquad (12b)$$

Case 3

Source room made absorptive and receiver room made reflective after inserting the barrier.

In considering this case, it is convenient to write the second term of the numerator inside the brackets as

$$\frac{4\ K_2}{(S_1\alpha_1 + S)\ (1 - K_1 K_2)}$$

Since $K_1 << 1$ and $K_2 \simeq 1$, the bracketed term is approximately

$$\frac{4}{S_1\alpha_1 + S}$$

and the insertion loss is

$$I.L. = 10 \log_{10}\left[\frac{\frac{QD}{4\pi r^2} + \frac{4}{S_1\alpha_1 + S}}{\frac{Q}{4\pi r^2} + \frac{4}{S_o\alpha_o}}\right] \text{ dB} \qquad (12c)$$

which indicates, among other things that the insertion loss depends on the amount of absorption in the source room rather than the lack of absorption in the receiver room.

Case 4

Receiver room made absorptive and source room made reflective after inserting the barrier.

Owing to the symmetry of K_1 and K_2 in Equation (12), it follows from Equation (12c) that the insertion loss for this case is

$$I.L. = 10 \log_{10}\left[\frac{\frac{QD}{4\pi r^2} + \frac{4}{S_2\alpha_2 + S}}{\frac{Q}{4\pi r^2} + \frac{4}{S_o\alpha_o}}\right] \text{ dB} \qquad (12d)$$

4. EXPERIMENTS

4.1 Insertion Loss in a Free Field

The experimental insertion losses for free field conditions were obtained by measuring the sound pressure level (SPL) produced by a small unhoused centrifugal blower impeller (i.e., ILG standard sound source) placed in an anechoic chamber both with and without a 4' x 6' x 1" compressed wood-chip barrier. The microphone was pulled at constant speed by an electric motor along the central axis of the barrier and the octave band SPL's were recorded on a level recorder. A sketch of the experimental arrangement is shown in Figure 2. The automatically plotted data are shown in Figure 3 as the SPL vs. distance, together with the calculated insertion loss according to Equation (10b). Actually, the insertion loss using the Wells' diffraction coefficient fits our data better than the insertion loss obtained using Rathe's diffraction coefficient.

The theoretical results of Redfearn indicate that absorption on the barrier has a minor effect on the free field insertion loss. To test this result, we measured the SPL of an 3/8 electric hand drill in an anechoic chamber (1) without a barrier, (2) using a 4' x 4' x 1" hard surfaced, compressed wood chip barrier and (3) using the same barrier as in (2) above but faced with a 1" layer

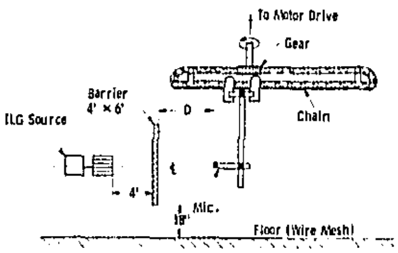

Fig. 2—Sketch of the experimental arrangement used to obtain the free field insertion loss for a finite sized barrier

of acoustically absorptive material. The absorptive material, Owens Corning "Unpainted Linear", is used commercially for suspended ceilings or wall treatments and has absorption coefficients of 0.08, 0.24, 0.68, 0.91, 0.96 and 0.94 at 125, 250 500, 1000, 2000, and 4000 Hz respectively according to the Acoustical and Insulating Materials Association. The results (Figure 4) show that the absorptive layer gives an average 1.5 dB increase in the insertion loss for the octave band center frequencies between 63 and 8000 Hz.

4.2 Insertion Loss in a Semi-Reverberant Room

The experimental insertion loss under semi-reverberant conditions was obtained by measuring the SPL at various distances from an ILG sound source in empty offices for three cases (shown in Figure 5) both with and without a 4' x 4' x 1" barrier.

The room absorption was obtained by measuring the SPL (without the barrier) at various distances from the source. At these larger distances, the ratio between the free field mean squared pressure Equation (8a) and the total mean squared sound pressure Equation (1) is

$$\left|\frac{p}{p_f}\right|^2 = 1 + \frac{16\pi r^2}{QS_o\alpha_o} \qquad (13)$$

which, when expressed in logarithmic form represents the difference between (a) the SPL for the source in a semi-reverberant room and (b) the SPL for the source in a free field. That is

$$\Delta SPL = 10 \log_{10}\left(1 + \frac{16\pi r^2}{QS_o\alpha_o}\right) \text{ dB} \qquad (13a)$$

By rearranging terms in Equation (13a), the room absorption, $S_1\alpha_o$, is

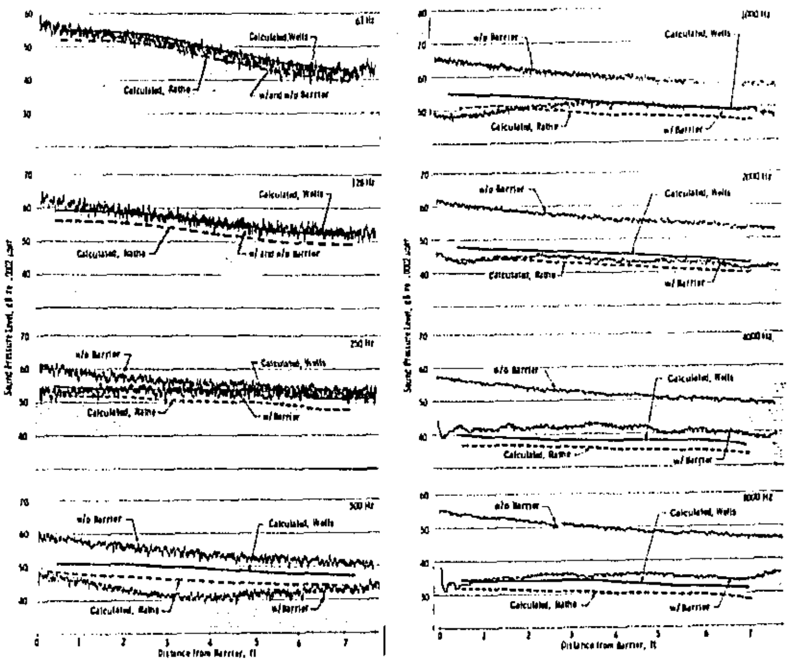

FIG. 3 – Sound pressure level vs distance from the barrier

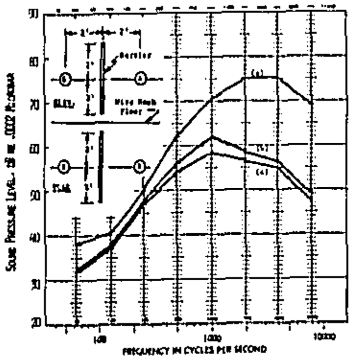

FIGURE 4 – Octave band SPL spectrogram for a 3/8 inch electric drill when used (a) without and (b) with an acoustically hard and (c) an acoustically soft barrier.

$$S_o\alpha_o = \frac{16\pi r^2}{Q[\log^{-1}(\frac{\Delta SPL}{10}) - 1]} \qquad (13b)$$

Near the sound source (say 1-2 ft), the reverberant field can be neglected and the mean squared sound pressure is given by Equation (8a) which, in turn, can be used to compute the source power, W. At larger distances from the source, the mean squared sound pressure is given by Equation (1). In determining the room absorption for factories, we have found this approach to be more direct than calculations based on published absorption coefficients and surface areas. In fact, the latter procedure is often impossible to implement in practical situations because absorption coefficient data is unavailable for some materials presently used in factory construction. If one plots the SPL vs. distance measured in a room on semi-log paper and then draws a straight line at a slope of -6 dB/doubling of distance through (or slightly below) the SPL measured closest to the source to represent the free field SPL, then, at any distance, the difference between the measured SPL and the line representing free field SPL can be used in Equation (13b) to determine the room absorption. The essential features

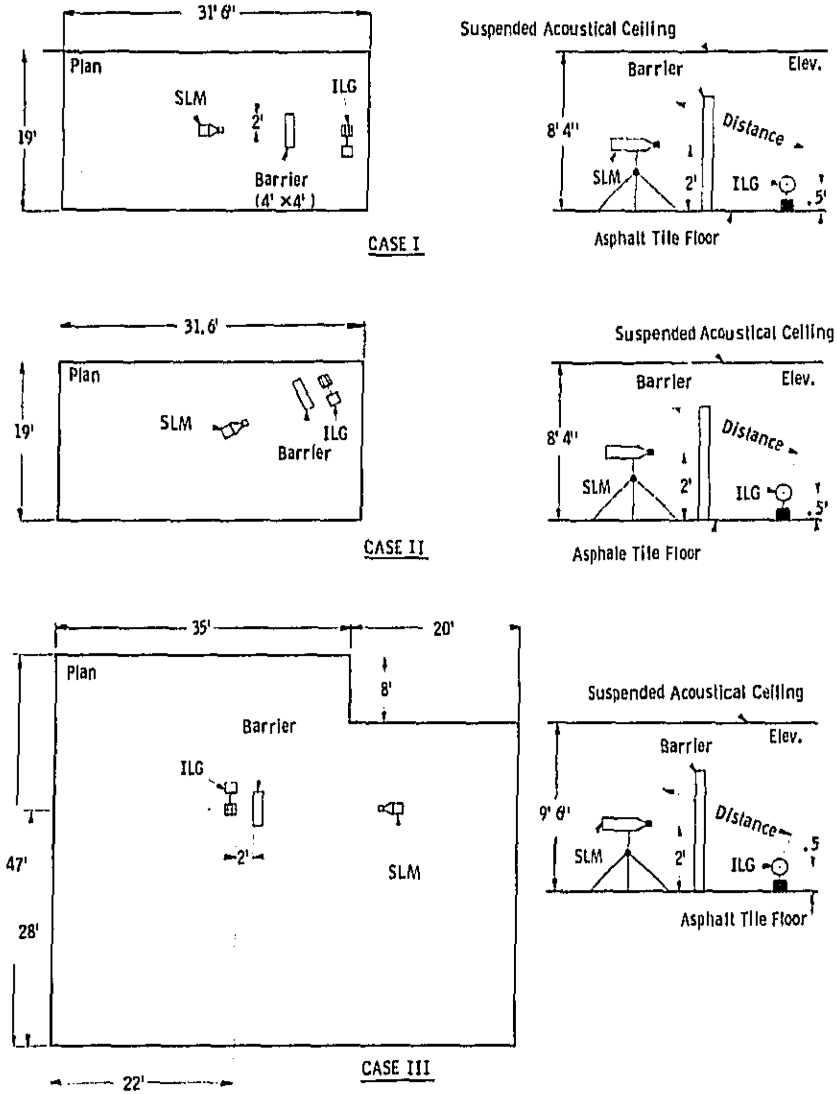

Figure 5—Arrangement used to collect data for the insertion loss of barriers in semi-reverberant rooms

of this method are illustrated in Figure 6.

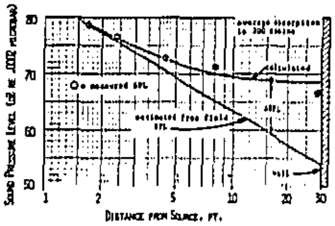

FIGURE 6 - Graph showing the use of SPL vs. distance measurements to determine the room absorption.

In most cases the room absorption obtained at various distances will be different owing to theoretical deficiencies inherent in Equation (1). For this reason, it is recommended that the absorptions as computed for various distances be averaged. When this procedure is used, Equation (1) describes the SPL vs. distance in a room reasonably well as shown by the "calculated" curve of Figure 6.

The measured and calculated insertion loss (using Rathe's diffraction coefficient) at various distances from the source for the three conditions previously mentioned are shown in Figures 7 through 9. These results are seen to be in reasonable agreement with the theoretical predictions of Equation (12), particularly at high frequencies. The estimated standard deviation between the measured and calculated insertion loss is 3.6 dB.

It is to be noted that in some low frequency cases, the measured insertion loss is negative. This is likely due to the fact that the constant source sound power assumption in our theory is not valid in these instances. Rather, the radiation impedance, and consequently the radiated sound power is increased owing to the confinement of the sound source by the reduced volume of the source room. The data of Figures 7 through 9 support this conclusion where the tendency towards negative insertion losses can be seen to increase as the size of the source room decreases. In addition, acoustical resonances may also be responsible for the measured negative insertion losses.

The tendency toward negative insertion

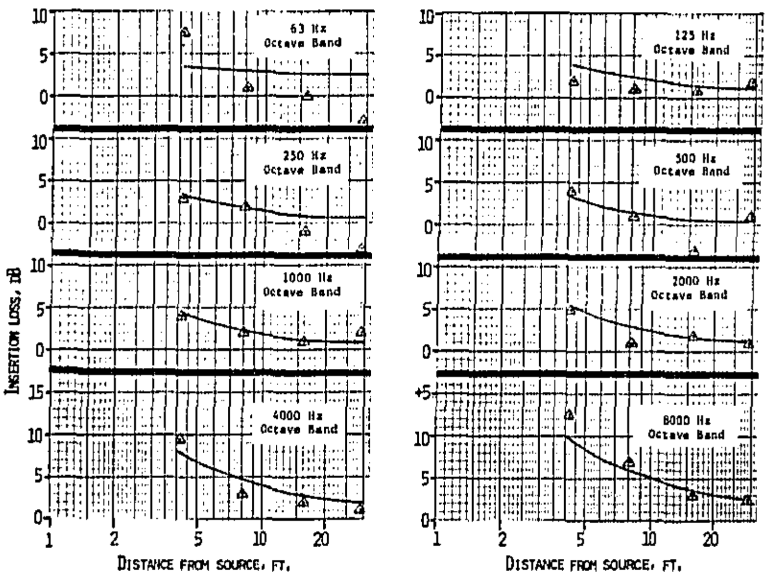

FIGURE 7 - Measured (triangles) and calculated (solid line) insertion loss in a semi-reverberant room for case I.

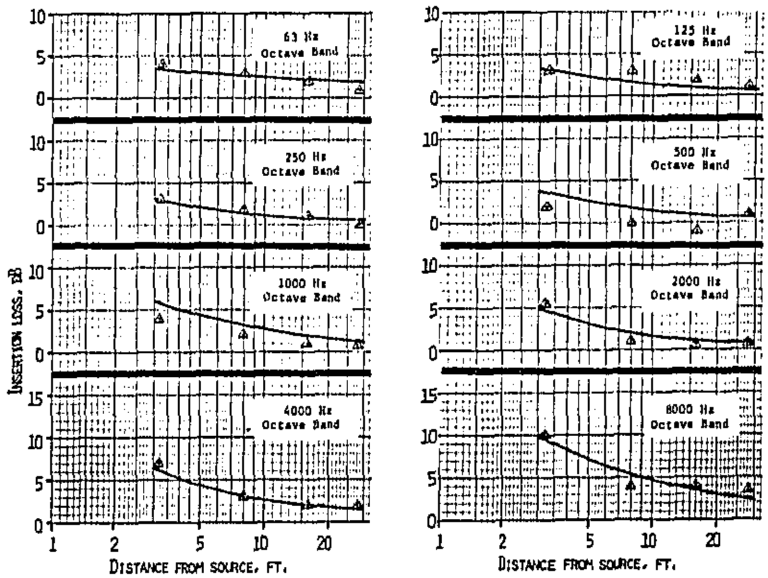

FIGURE 8 - MEASURED (TRIANGLES) AND CALCULATED (SOLID LINE) INSERTION LOSS IN A SEMI-REVERBERANT ROOM FOR CASE II.

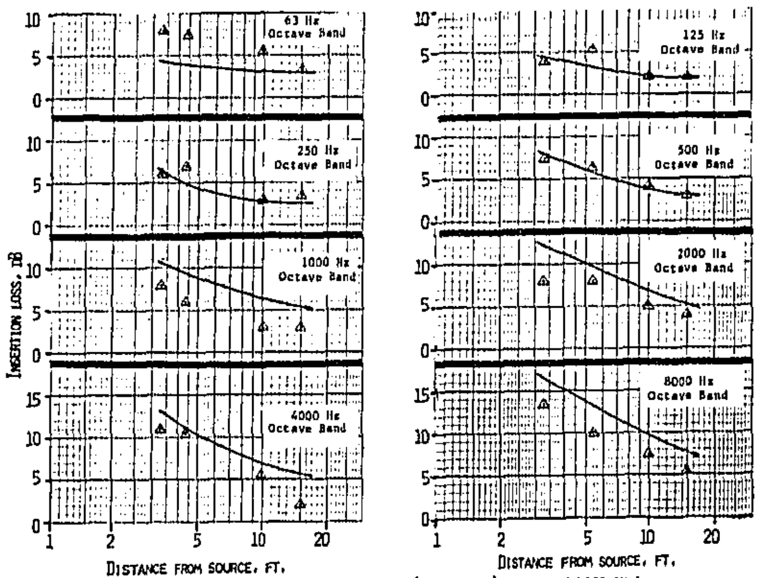

FIGURE 9 - MEASURED (TRIANGLES) AND CALCULATED (SOLID LINE) INSERTION LOSS IN A SEMI-REVERBERANT ROOM FOR CASE III.

losses can be suppressed by making the source room wall surfaces more acoustically absorptive.

5. SUMMARY

The insertion loss of a finite sized barrier can be estimated reasonably well for both free field and semi-reverberant conditions. The room absorption, which is required in arriving at the estimate, can be determined from measurements of the SPL at various distances from the source. The insertion loss calculation procedure presented herein will be useful to persons concerned with reducing machinery noise levels in industrial plants.

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ACOUSTICAL DESIGN OF A LOW PRESSURE LOSS ATTENUATING BEND FOR NOISE CONTROL IN AIR CONDITIONING DISTRIBUTION DUCT SYSTEMS.

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INTRODUCTION

Low-frequency attenuation of air conditioning noise within the distribution flow duct system has been a difficult and complex problem. Almost without exception, any appreciable amount of noise reduction in the frequency range of 50 to 500 Hz was achieved only at the expense of substantial cost and large weight penalty. At the same time, it is well known that in most low-pressure, low-velocity airflow duct systems, a bend of some type-such as a Y, or a T, or a 90° angle, behaves areodynamically as effectively as a piece of long, straight ducting.

The purpose of the present study was to design a special low-pressure loss bend that was effective both from aerodynamic and acoustical standpoints, thus providing a natural attenuation at the low design frequency without the penalty of required static pressure loss in the distribution system. In addition, necessary steps were taken to ensure that the proposed bend design was economical and practical in its application.

DISCUSSION

Most common type of air conitioning noise encountered in distribution duct systems whether in building construction or passenger aircrafts, is of a broad-band nature. In many cases, the excessive noise from various air conditioning machines, is partially attenuated by long sections of distribution duct system. In addition, one can further reduce the high frequency noise in these ducts by properly lining the interior duce surfaces with commercially available acoustical materials. But the fact still remains that it is practically impossible to attenuate low-frequency noise transmission and propagation in these ducts, effectively and economically, without adding excessive weight and cost to the air conditioning system.

The proposed low-pressure attenuating bend was designed as an 'acoustic rejection filter' (notch filter and or T-bridge) in the transmission line between the (noise) source and termination (sink). An appreciable reduction in noise propagation downstream from the bend is obtained, with maximum attenuation of 12 dB at 250 Hz design frequency.

In the proposed bend, two channels have been provided about a profiled centerbody. The path lengths along the center lines of the two channels differ by $\lambda/2$ (or 180° phase angle shift), where λ is the acoustic wave-length corresponding to the blade passage frequency of the compressor fan in most ventilating systems. Thus, if one assumes one-dimensional propagation of a discrete frequency noise along the two channels, the outlet plane of the filter becomes a low impedance point. When viewed from upstream, the impedance mismatch between the bend and the connecting duct, therefore, results in a poor power (or energy) transfer to the downstream duct, that is, a high attenuation in dB at the design frequency. A low pressure loss (Δ p) was obtained by proper aerodynamic profiling of the centerbody and the interior walls of the bend.

TEST PROCEDURE AND RESULTS

Figure 1 shows the general test set-up for acoustical evaluation of the proposed attenuating bend. To introduce sound into the airstream of the quiet air source (muffler), a nozzle-type noise source was installed 9 feet upstream of the attenuating bend (Reference 1). With airflow of 2,000 to 8,000 ft/min (in increments of every 2,000 ft/min) sound pressure levels were recorded at four preselected test locations. Four Bruel & Kjaer 1/2 inch condenser microphones were flush-mounted to the walls of the ducts including the test bend. A common instrumentation scheme[1] was employed for the entire test measurements, as shown in figure 2. Two static pressure probes were installed in the test duct system for measuring pressure drop (Δp) across the bend.

Two sets of data were recorded, one with and the other without the artificial (nozzle) noise source upstream of the test bend (Figure 4); Two separate plots are included here, one showing the variation of pressure drop (Δp) in inches of water versus airflow velocity in ft/min (Figure 3), and the other showing attenuation in decibels (dB) versus preferred octave-band frequencies at various airflow velocities (Figure 5). The attenuation in dB is the difference in SPL between two flush-mounted microphone readings, one located upstream and the other downstream of the attenuating test bend. Figure 2 shows the schematic of the attenuating bend with actual dimensions and sample computation for the low design frequency.

CONCLUSION

The test results indicate clearly the acoustical effectiveness of the attenuating bend design, which not only gives appreciable dB attenuation at the low design frequency but also meets the low pressure-drop (Δp) requirements of the air conditioning distribution duct system. Additional attenuation could be achieved by lining the interior side walls of the bend with commercially available noise-control materials, (Reference 2).

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Figure 1-Test Set-up Schematic for the Attenuating Bend

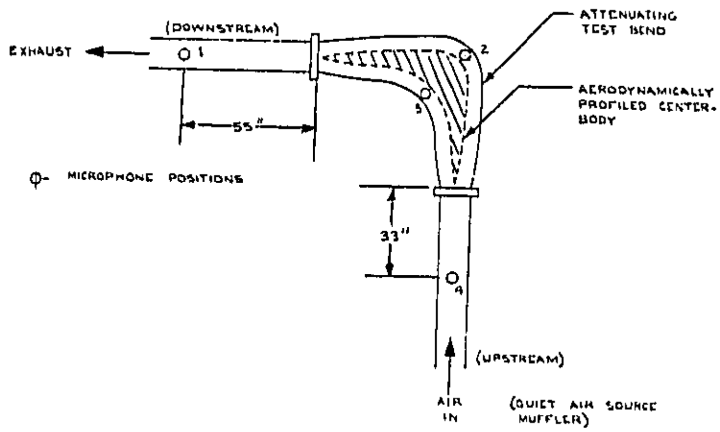
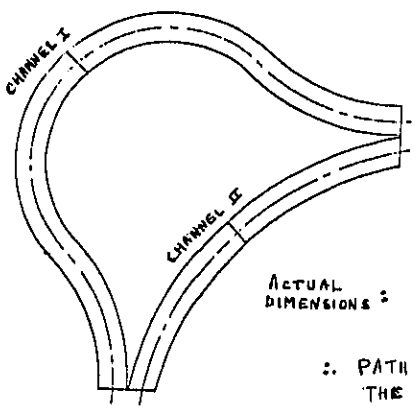


Figure 2-Attenuating Bend Geometry



ACTUAL DIMENSIONS: PATH-LENGTH OF CHANNEL I = 46.50" = d_I
PATH-LENGTH OF CHANNEL II = 24.25" = d_{II}

∴ PATH DIFFERENCE BETWEEN THE TWO CHANNELS = $(\Delta d) = 22.25''$

BUT $\Delta d = \frac{\lambda}{2} = 22.25'' = 1.86$ ft.

∴ $\lambda = (2 \times 1.86) = 3.72$ ft.

∴ $f = \left(\frac{c}{\lambda}\right) = \left\{\frac{1,128}{3.72}\right\} = 300$ Hz

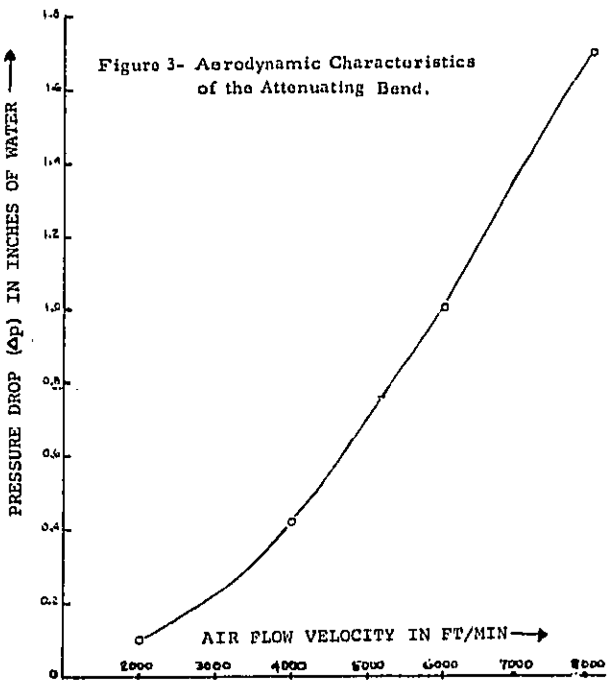


Figure 3- Aerodynamic Characteristics
of the Attenuating Bend.
PRESSURE DROP (Δp) IN INCHES OF WATER
AIR FLOW VELOCITY IN FT/MIN
0
2000
3000
4000
5000
6000
7000
8000

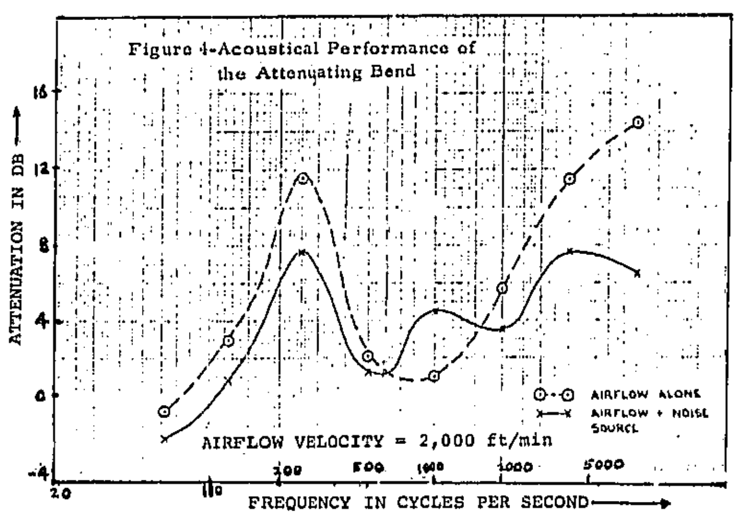


Figure 4-Acoustical Performance of
the Attenuating Bend
ATTENUATION IN DB
16
12
8
4
0
-4
AIRFLOW ALONE
AIRFLOW + NOISE SOURCE
AIRFLOW VELOCITY = 2,000 ft/min
FREQUENCY IN CYCLES PER SECOND

Figure 5-Acoustical Performance of the Attenuating Bend

AIRFLOW VELOCITY = 4,000 ft/min

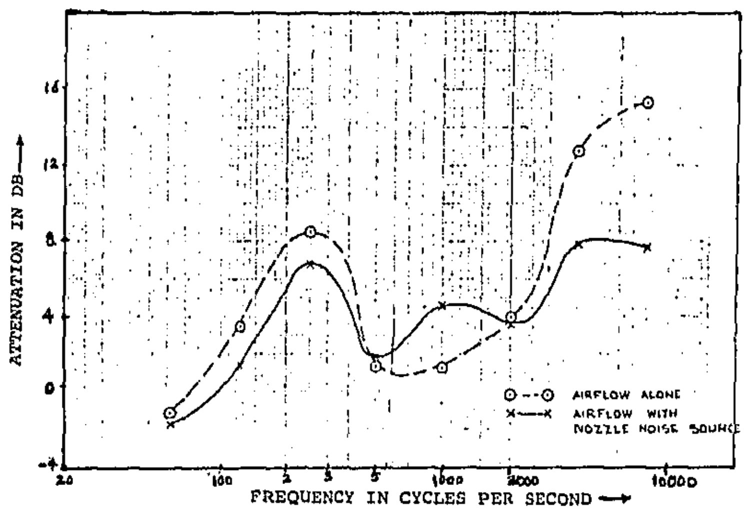
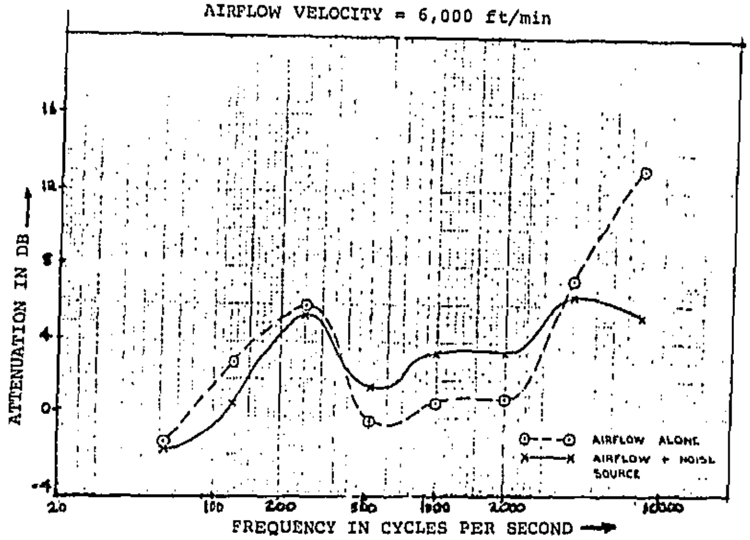


Figure 6-Acoustical Performance of the Attenuating Bend

AIRFLOW VELOCITY = 6,000 ft/min



NOISE DIFFRACTION: SUGGESTED ESTIMATION PROCEDURES

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In the present paper, formulas, curves, and procedures are outlined for the estimation of sound pressure levels at locations which are partially shielded from the sound source by a barrier. Existing literature[1,2] suggests that all techniques commonly used to estimate diffracted sound pressure levels are based either on the Fresnel-Kirchoff diffraction theory[3] or on Sommerfeld's exact solution[4] for a semi-infinite screen of negligible width. Such techniques would seem appropriate only if the thickness of the barrier is small compared to a wavelength. Although Maekawa[5] has recently suggested a technique appropriate to wide barriers based on the replacement of a wide barrier by an "equivalent" screen, this technique appears to have little theoretical basis and to often give erroneous estimates. The alternative procedures presented here are based on the asymptotic theory of sound diffracted by a right-angled wedge[6,7] and on Keller's geometrical theory of diffraction.[8-10]

In the interests of brevity, the theoretical development is omitted here and only results are given, the emphasis being on the computational procedure, the hope being that such a presentation will be of maximum benefit to practicing noise control engineers. A more archival article is planned for separate publication.

Procedures given here are restricted to the case when the listener lies in a shadow zone. Any listener location is said to lie in a shadow zone if the straight line (direct path) from source to listener passes through the barrier. If sound cannot pass through the barrier, then any sound received in the shadow zone must have traveled along a path from source to listener which is not straight and is accordingly said to have been diffracted. Generally, sound pressure levels in a shadow zone are less (by as much as 20 dB or more) than what would be expected were the barrier not present.

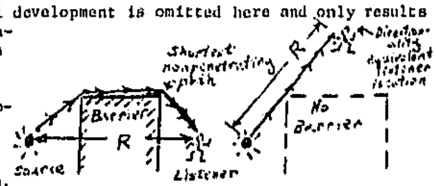

Fig. 1. Definition of directionality equivalent listener location.

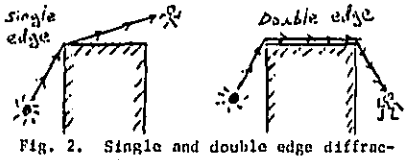

Fig. 2. Single and double edge diffraction.

A. DEFINITION OF EXCESS ATTENUATION

The excess attenuation (in dB) due to a barrier may be defined as follows (see Fig. 1) for an omni-directional source. Let $(L_p)_{NB}$ be the sound pressure level which would be measured at the same point were no barrier present. Then excess attenuation (EA) is defined such that

$$(L_p)_B = (L_p)_{NB} - (EA) \qquad (1)$$

The fact that sound propagation is a predominantly linear phenomenon implies that (EA) should be independent of the strength of the source, although it will certainly de-

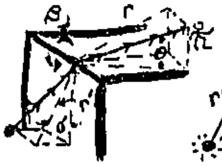

Fig. 3. Symbols for single edge diffraction.

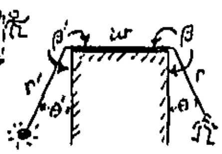

Fig. 4. Symbols for double edge diffraction.

pend on the source location and source frequency spectrum.

If the source is directional, then Eq. (1) will still define the excess attenuation in such a way that (EA) is independent of source directionality characteristics, providing $(L_p)_{NB}$ is interpreted as the sound pressure level which would be measured in the absence of the barrier at a "directionality equivalent" listener location. This "directionality equivalent" location is at the same radial distance from the source as the actual listener location but is along a direction which may be somewhat different. The direction is defined to be that which is the initial direction of the shortest possible (crooked) line connecting source to listener which would not penetrate a barrier (Fig. 1).

Since diffraction is a frequency dependent phenomenon, the excess attenuation (EA) will in general vary with frequency. In general (EA) increases with frequency (implying more effective shielding) since higher frequencies diffract less than lower frequencies. If the source is broadband, it may often be appropriate to estimate the actual (EA) by that which might be expected for a pure tone source with some representative frequency. If the source spectrum is flat over the audible range, the choice of f = 1000 Hz would appear to be a reasonable choice of representative frequency.

B. GEOMETRICAL PARAMETERS IN DIFFRACTION

Given that the listener lies in the shadow zone of a barrier, the estimation of the excess attenuation (EA) in general requires a knowledge of a few elementary parameters characterizing the arrangement of source and listener positions with respect to the barrier location. A key concept is that of the shortest nonpenetrating path, which is the shortest (with sharp bends allowed) line going from source to listener which does not penetrate the barrier. Two cases are of common interest and are accordingly discussed here. (See Fig. 2.) The first case (single edge diffraction) is where the shortest nonpenetrating path touches the barrier at just one point, this point lying on an edge of the barrier and corresponding to a kink in the shortest nonpenetrating path. The second case (double edge diffraction) is where a segment of the path lies on a face of the barrier, the end points of this segment being at edges of a side of the barrier. (Note that we specifically exclude curved barriers here. However, reasonable estimates of escess attenuation by curved barriers can conceivably be made if one replaces the curved barrier by an equivalent rectilinear barrier.)

In the case of single edge diffraction, (Fig. 3) we define parameters r, θ, r', θ', β, L, and R as follows. The parameter r is the radial distance (shortest) of the listener from the diffracting edge, while r' is the corresponding radial distance of the source from the diffracting edge. The angle θ is the angle between the plane passing through listener location and diffracting edge with the plane which forms the side of the wedge nearest to

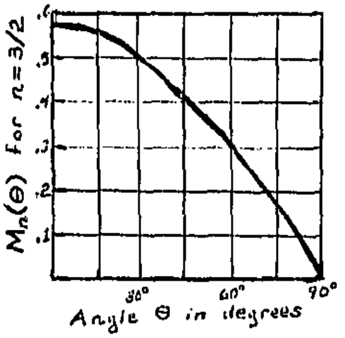

Fig. 5. Angle dependent factor $M_n(\ )$ defined in Eq. 10 for the case n=3/2.

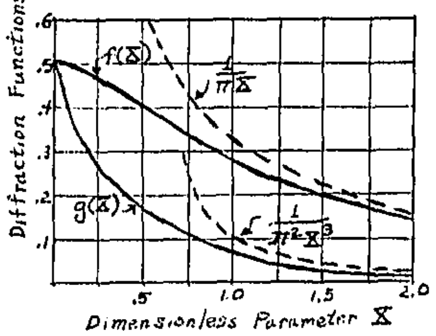

Fig. 6. Diffraction functions f(X) and g(X) versus X along with plots of their asymptotic limits.

the listener. Similarly, θ' is defined as the angle between plane through source location and edge with wedge side nearest to source. The angle β is the exterior angle of the wedge. Thus, if the corner is rectangular, the wedge interior angle is 90° and β would be 360°-90° or 270°. If the diffracting body is modeled as a thin screen of negligible thickness, then the interior angle would be 0° and β would be 360°. The parameter L is the length of the shortest nonpenetrating path, while R would be the length of the hypothetical straight line connecting source to listener which penetrates the barrier. (See Fig. 3). Note that if the diffracting edge is taken as the z axis and the z coordinates of the source and listener are taken as z' and z respectively, then

$$L = [(r + r')^2 + (z - z')^2]^{1/2} \tag{2}$$

$$R = [(r)^2 + (r')^2 - 2rr'\cos(\theta' + \theta - \beta) + (z - z')^2]^{1/2} \tag{3}$$

Also, one should note that the listener lies in the shadow zone only if

$$\theta < (\beta - 180°) - \theta' \tag{4}$$

In the case of double edge diffraction, the only case considered here to be of interest is where both edges are parallel (Fig. 4). Parameters of interest are r, θ, r', θ', w, β_1, β_2, L and R. The parameter r is the radial distance (shortest) of the listener from the nearest diffracting edge. The angle θ is the angle between plane-passing through listener and nearest diffracting edge with side of barrier nearest to listener, while θ' is the angle between plane passing through source and its nearest diffracting edge with side of barrier nearest to source. The width of the barrier between the two diffracting edges is denoted by w. The exterior angle of the edge nearest to source is denoted by β_1, while the exterior angle of edge nearest to listener is denoted by β_2. As in the case of single edge diffraction, L is defined as the length of shortest nonpenetrating path connecting source to listener, while R is the length of the shortest barrier penetrating path (i.e., the direct path) connecting observer to listener. If either diffracting edge is taken as the z axis such that the z coordinates of source and listener are z' and z, respectively, then,

$$L = [(r + r' + w)^2 + (z' - z)^2]^{1/2} \tag{5}$$

$$R = [(z' - z)^2 + \{w - r\cos(\beta_2 - \theta) - r'\cos(\beta_1 - \theta')\}^2 + \{r\sin(\beta_2 - \theta) - r\sin(\beta_1 - \theta')\}^2]^{1/2} \tag{6}$$

One should also note that, in order for the conditions of double diffraction to hold, he should have both

$$\theta' < \beta_1 - 180° \tag{7a}$$

$$\theta < \beta_2 - 180° \tag{7b}$$

C. GENERAL FORMULAS FOR EXCESS ATTENUATION IN SINGLE EDGE DIFFRACTION

Let λ be the free air wavelength of the sound radiated by the source, i.e., $\lambda = c/f$ where f is the frequency in Hertz and c = 1130 ft/sec is the speed of sound. Another definition of interest is the wedge angle index

$$n = \beta/180° \tag{8}$$

where β (measured in degrees) is the exterior wedge angle defined in the previous section.

The general procedure for calculating excess attenuation is as follows. One first calculates the dimensionless quantity

$$\gamma = (n)\left(\frac{2rr'}{\lambda L}\right)^{1/2} \tag{9}$$

He also should obtain the two functions $M_n(\theta + \theta')$ and $M_n(\theta - \theta')$ where

$$M_n(\theta) = \frac{\cos(180°/n) + \cos(\theta/n)}{\sin(180°/n)} \tag{10}$$

$$= \frac{2\sin\{[(n-1)180° - \theta]/2n\}\sin\{[(n-1)180° + \theta]/2n\}}{\sin(180°/n)} \tag{10'}$$

Here one merely replaces θ by $\theta + \theta'$ or $\theta - \theta'$ in the argument above to obtain $M_n(\theta + \theta')$ or $M_n(\theta - \theta')$. For convenience of computation, we give in Fig. 5 a graph of $M_n(\theta)$ versus θ for the case n = 3/2 (right angled edge). If β = 270°, one may find $M_n(\theta + \theta')$ or $M_n(\theta - \theta')$ by simply picking the appropriate value of M_n from the graph. [For example, if θ = 10°, θ' = 25°, one looks on the graph for M_n(35°) and M_n(-15°). Note that $M_n(-\theta)$ is the same as $M_n(\theta)$.]

Once γ, $M_n(\theta + \theta')$, $M_n(\theta - \theta')$ are obtained, one uses these to evaluate two diffraction parameters, $\underline{X}_+$ and $\underline{X}_-$, defined as

$$\underline{X}_+ = \gamma M_n(\theta + \theta') \quad (11a)$$

$$\underline{X}_- = \gamma M_n(\theta - \theta') \quad (11b)$$

For each of these latter two numbers, one evaluates functions $f(\underline{X}_+)$ and $f(\underline{X}_-)$ as well as $g(\underline{X}_+)$ and $g(\underline{X}_-)$. The values of $f(\underline{X})$ as well as $g(\underline{X})$ for either value $\underline{X}_+$ or $\underline{X}_-$, of the argument $\underline{X}$ may be found from the graphs in Fig. 6 if $\underline{X} < 2.0$. If $\underline{X} > 2.0$, it is sufficient to take

$$f(\underline{X}) = 1/(\pi \underline{X}) \qquad \underline{X} > 2.0 \quad (12a)$$

$$g(\underline{X}) = 0 \qquad \underline{X} > 2.0 \quad (12b)$$

for the purpose of finding excess attenuation with $\pm$ 0.5 dB. In typical applications one may generally expect to find that the computed values of $\underline{X}_-$ are invariably larger than 2.0 while $\underline{X}_+$ is larger than 2.0 unless the observer is very near the edge of the shadow zone. (This presumes the value of γ in Eq. (9) is greater than, say, 5.)

Once $f(\underline{X}_+)$, $f(\underline{X}_-)$, $g(\underline{X}_+)$, and $g(\underline{X}_-)$ have been evaluated, the excess attenuation due to the barrier may be readily calculated from the formulas

$$(EA) = -10 \log_{10} Q \quad (13)$$

where

$$Q = (R/L)^2 (1/2) [\{ f(\underline{X}_+) + f(\underline{X}_-) \}^2 + \{ g(\underline{X}_+) + g(\underline{X}_-) \}^2] \quad (14)$$

If both $\underline{X}_+$ and $\underline{X}_-$ are larger than 2 (which typically should hold unless the listener is very close to the shadow zone boundary) the above formulas reduce to

$$(EA) = +10 \log_{10} \left\{ \frac{rr'L}{\lambda R^2} \right\} - L'_n(\theta, \theta') \quad (15)$$

where $L'_n(\theta,\theta')$ is an angle dependent factor (in dB) given by

$$L'_n(\theta,\theta') = 20 \log_{10} \left\{ \frac{1}{2n\pi} \left[\frac{1}{M_n(\theta+\theta')} + \frac{1}{M_n(\theta-\theta')} \right] \right\} \quad (16)$$

In Fig. 7, contours of equal $L'_n(\theta,\theta')$ for $n = 3/2$ are plotted in a θ versus θ' diagram. The nature of the figure is such that one may pick out values (with some extrapolation) of $L'_n(\theta,\theta')$ for given θ and θ'. Thus, for right angled edges, one need not directly compute $L'_n(\theta,\theta')$ from Eq. 16. If one wishes to use Eq. (15) and Fig. 7 without first checking on the values of $\underline{X}_+$ or $\underline{X}_-$, he should bear in mind that (EA) should always be

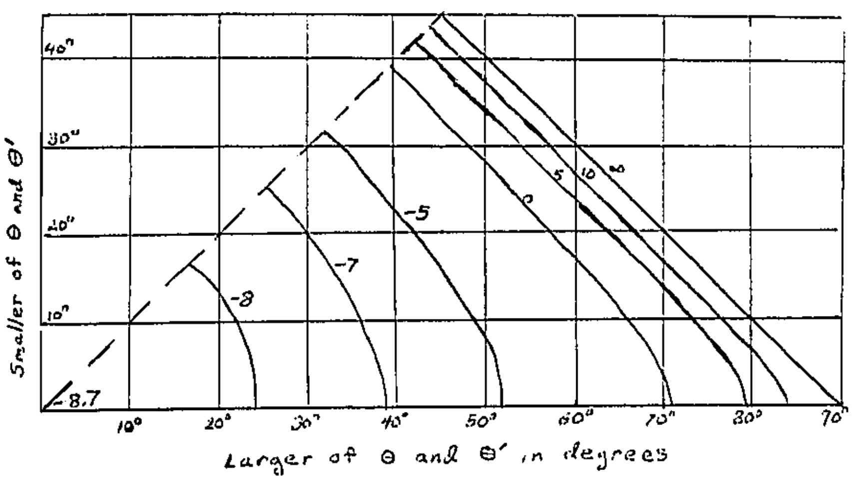

Fig. 7. Contours of constant $L_n(\theta,\theta')$ vs. θ and θ'. Note that $L_n(\theta,\theta')$ is in dB and is as defined in Eq. 16 above.

greater than 0. If Eq. (15) gives a negative value than the use of Eq. (13) and (14) would certainly be indicated.

D. GENERAL FORMULAS FOR EXCESS ATTENUATION IN DOUBLE EDGE DIFFRACTION

As in the previous section, we let λ be the free air wavelength of the sound radiated by the source. We also define two wedge angle indices n_1 and n_2 where

$$n_1 = \beta_2/180° \tag{17a}$$

$$n_2 = \beta_2/180° \tag{17b}$$

where β_1 and β_2 are the two exterior angles. (See Sec. B and Fig. 4). Other pertinent definitions are the dimensionless quantities

$$\gamma_1 = (n_1)\left\{\frac{2r'[w+r]}{\lambda L}\right\}^{1/2} \tag{18a}$$

$$\gamma_2 = (n_2)\left\{\frac{2r[w+r']}{\lambda L}\right\}^{1/2} \tag{18b}$$

and the quantities

$$A_1 = \gamma_1 M_{n_1}(\theta') \tag{19a}$$

$$A_2 = \gamma_2 M_{n_2}(\theta) \tag{19b}$$

where, as in Eq. (10),

$$M_n(\theta) = \frac{\cos(180°/n) + \cos(\theta/n)}{\sin(180°/n)} \tag{20}$$

Here $M_{n_1}(\theta')$ may be obtained from the above expression with an appropriate substitution of n_1 for n and θ for θ'. For convenience in computation, we give a plot of $M_n(\theta)$ versus θ in Fig. 5 for the special case $n = 3/2$ (right angled edges).

Another parameter needed for the computation is

$$B = \frac{1}{2}\left\{\frac{w(w+r+r')}{(w+r')(w+r)}\right\}^{1/2} \tag{21}$$

which may be considered as characterizing the barrier width. (It is presumed that w/λ is greater than, say, 3.)

Although the exact solution for double edge diffraction is not known, very good approximate expressions have been derived for the limiting cases of either A_1 or A_2 identically zero or for both A_1 and A_2 large. With these limiting cases as a guide, we have devised an extrapolated expression which should be approximately valid for all cases of interest. The suggested expression for excess attenuation is

$$(EA) = -10 \log_{10} Q \tag{22}$$

where

$$Q = (R/L)^2[f^2(A_>) + g^2(A_>)][f^2(BA_<) + g^2(BA_<)] \tag{23}$$

where $A_>$ is the larger of the two quantities A_1 and A_2 and $A_<$ is the lesser. (Both should be positive.) Here the functions $f(\underline{X})$ and $g(\underline{X})$ are as plotted in Fig. 6. They should alos be approximated by Eqs. (12) if the argument is greater than 2.

If both BA_1 and BA_2 are large compared to 1 (greater than 2 should be sufficient) then the previous formulas given for excess attenuation reduce to

$$(EA) = +10 \log_{10} \left\{\frac{wrr'(w+r+r')}{\lambda^2 R^2}\right\} - L_{n_1}(\theta') - L_{n_2}(\theta) \tag{24}$$

where $L_n(\theta)$ is given by

$$L_n(\theta) = -20 \log_{10}\left\{n\pi\, M_n(\theta)\right\} \tag{25}$$

In order to facilitate the use of Eq. (24) in rapid estimations, a graph of $L_n(\theta)$ is given in Fig. 8 for $n = 3/2$ (right angled edge). We suggest that one may take $L_n(\theta)$ directly from the graph rather than use Eq. (25). If one ever seeks to use Eq. (24) directly without first checking the magnitudes of BA_1 and BA_2, he should bear in mind that (EA) should always be greater than 6 dB for double edge diffraction. If Eq. (24) gives a number less than this or a negative value, then the use of Eqs. (22) and (23) is warranted.

Generally, if the first term in Eq. (24) is greater than, say, 30, and both of the $L_n(\theta)$ are less than, say, 12, one would expect Eq. (24) to be a very good approximation. The chief limitations in the case of right angled barriers occur whenever θ or θ' is close to 90° (such that a singly diffracted ray passes very close to the listener.)

ACKNOWLEDGEMENT

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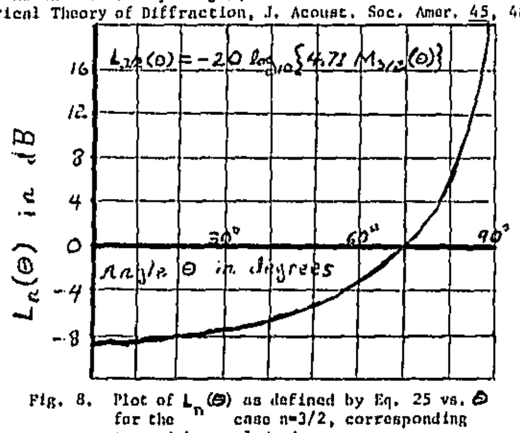

Fig. 8. Plot of $L_n(\theta)$ as defined by Eq. 25 vs. θ for the case n=3/2, corresponding to a right angled edge.

MATERIALS FOR NOISE CONTROL

COMBINED EXTENSIONAL AND CONSTRAINED DAMPING FOR BROAD TEMPERATURE AND FREQUENCY PERFORMANCE

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INTRODUCTION

Laminated constructions to control resonant vibrations have been available and in use for many years. One of the more common types of laminate uses is an elastomeric damping material between two metal sheets as shown in Fugure (1). When this combination undergoes bending deformations, the polymeric core is sheared between the two metal plates or beams. Because of the inherent characteristics of the polymer, some of the energy of deformation will be dissipated as heat. This results in a structure with a controlled resonant response.

The energy dissipation or damping properties of the geometry of laminates of this type are frequency dependent; however, as is well known, the properties of polymeric materials are also frequency dependent. The stiffness of polymers increases slowly with the frequency of excitation. This has the effect of modifying the coupling between the structural portions of the laminate such that the inherent frequency dependence of the laminate is improved. That is, the range of high energy dissipation of the laminate is broadened due to the frequency dependence of the polymeric core in contrast to that expected if the core had a single value of stiffness. In addition, because of the temperature dependence of all polymeric materials, the damping properties of the laminates are also temperature-dependent.

The results of our work presented here demonstrate ways of considerably broadening both the temperature and frequency ranges of laminated configurations. This has been achieved through the combined use of several polymeric damping layers, each with its own particular properties as a function of temperature and frequency.

The general configuration under discussion in this paper is shown in Figure (2). There are two outer metal layers with two different elastomeric damping materials sandwiched in between. The entire assembly is rigidly bonded together at all interfaces.

GENERAL CHARACTERISTICS

The generalized mechanical properties of a polymeric damping material as a function of temperature and frequency are shown in Figure (3). All polymeric damping materials have these general characteristics. The useful temperature/frequency range is, of course, where the material loss factor (a measure of the damping it contains) is at or near its peak.

We have found it very useful to employ two or more different damping materials with different temperatures of peak damping; thus, one of the materials should be acting at or near its optimum temperature when the laminate is exposed to a relatively wide range of temperatures. The equations which determine the composite shear modulus and loss factor of the two combined damping layers are as follows:

$$G_o = \frac{G_2G_3[G_3H_2(1+\beta_3^3) + G_2H_3(1+\beta_2^2)]\ (H_2+H_3)}{(G_3H_2+G_2H_3)^2 + (G_3H_2\beta_3 + G_2H_3\beta_2)^2} \qquad (1)$$

$$\beta_o = \frac{\beta_2G_3H_2(1+\beta_3^2) + \beta_3G_2H_3\ (1+\beta_2^2)}{G_3H_2(1+\beta_3^2) + G_2H_3(1+\beta_2^2)} \qquad (2)$$

where G_i = Shear Modulus of the i-th layer
β_i = Material Loss Factor of the i-th layer
H_i = Thickness of the i-th layer

Study of these equations shows that it is not satisfactory for proper design to simply choose materials with high loss factors in the proper temperature. From the equations, it is clear that the shear modulus and thicknesses of the layers must also be taken into consideration.

For example, if one of the layers is taken to contain no damping ($\beta_2=0$) equations (1) and (2) reduce to:

$$G_o = \frac{(H_2+H_3)(1+\beta_3^2)\ G_3}{H_3\ \left[1+(1+\beta_3^2)\ \frac{G_3/H_3}{G_2/H_2}\right]\ \left[1+\frac{\beta_3^2}{1+(1+\beta_3^2)\ \frac{G_3/H_3}{G_2/H_2}}\right]} \qquad (3)$$

$$\beta_o = \frac{G_2\beta_3H_3}{G_2H_3 + G_3\ H_2\ (1+\beta_3^2)} \qquad (4)$$

Equations (3) and (4) show that the stiffness and damping of the combined layers are still fairly complex functions of the thickness and shear modulii of the two layers. The following illustrates what can happen:

Case I $\qquad G_3 \ll G_2$

$$\beta_o = \beta_3$$

$$G_o = \frac{(H_3+H_2)\ G_3}{H_3}$$

Case II $\qquad G_3 \gg G_2$

$$\beta_o = 0$$

$$G_o = \frac{(H_3+H_2)\ G_2}{H_2}$$

It can be seen that in Case I the composite damping characteristics are identical to those of layer (3), while in Case II there will be no damping produced by the composite layer. These limiting conditions occur because all the motion is forced to occur in one of the layers only.

The ideal situation would be to have all the motion occur in the layer which under the input conditions has the highest damping. In practice, this cannot be accomplished. The best that can normally be achieved is to use two materials which have nearly the same stiffness characteristics and different temperatures of peak damping. Small differences in stiffness characteristics can be made up by adjusting the thicknesses of the different layers. Figure (4) is an illustration of the properties of two different damping materials and the properties they produce when combined. The softer material has been used in a thinner layer to make up for the thickness differential. It is clear from the figure that the combination of the materials would produce a composite material which had broader temperature range capability.

OPTIMIZATION

The equation which predicts the composite loss of a laminate of the type under consideration is:

$$\eta_c = \frac{2\beta_o y g}{1 + (2 + y)2g + 4(1 + \beta_o^2)(1 + y)g^2}$$

where:

η_c = composite loss factor of the laminate
β_o = loss factor of the combined damping material
y = geometrical parameter
g = frequency parameter

From standard methods, the maximum composite loss factor will occur when:

$$g_{opt} = \frac{1}{2(1 + y)^{1/2}(1 + \beta_o^2)^{1/2}}$$

Where: $$y = \frac{3h_4\,[1 + 2\,(h_2 + h_3) + h_4]^2}{(1 + h_4)(1 + h_4^2)}$$

$h_i = H_i/H_1$

H_i = thickness of the i-th layer

H_1 = thickness of the first layer of the laminate

If substitutions of these relations are made into the equation for composite loss factor, this loss factor can then be optimized with respect to geometry. Figure (5) is a plot of these optimized geometrical terms for a laminate which contains two materials in its core. Both materials have a finite stiffness, but one of them contains no damping. Picking a laminate which fits on one of these curves produces a laminate which has a maximized loss factor.

BROAD FREQUENCY RANGE PERFORMANCE

The damping treatments discussed above all use two shear materials. It is possible to construct a laminate using one extensional damping material and one shear damping material. An extensional damping material is one which, when directly applied to a structure, damps the bending deformations of the structure by undergoing alternate extension and compression. No constraining layer is applied to this type of material. The basic difference between a shear material and an extensional material is that the extensional material is of much higher rigidity.

The advantage in using both types of materials in combination is that a laminate made from the combination canbe used to increase the useful temperature and/or frequency range. The geometry of an extensional damping treatment is independent of frequency, so neglecting the frequency sensitivity of the material itself at any given temperature, the composite loss factor of such a damped structure will be a constant. Referring to Figure (2), consider the case where layer 2 is an extensional material and layer 3 is a shear material. Figure (6) now shows a comparison of the composite laminate and a laminate using only the shear material. It is clear that the frequency range of high damping is increased.

The effective temperature range of the overall laminate may be increased by using an extensional material and shear material of different temperatures of peak damping. Again referring to Figure (2), if layer 1 is the structure to be damped, layer 2 is the extensional material and layer 3 is the shear material, the proper selection of the materials can be used to get broad temperature range performance. The key is to select an extensional material which has a temperature of peak damping that is higher than that of the shear material. Selection by this criterion results in the following. At the lower end of the temperature range, the extensional material is rigid and contains very little damping. At these temperatures, a very high percentage of the damping is provided by the shear material. At the high end of the useful temperature range, just the opposite is true. Nearly all of the damping will be provided by the extensional material . The shear material at this temperature is very soft and contains very little damping. At these temperatures, the upper two layers are essentially "going along for the ride", and performance of the entire damping treatment is approximately the same as if only the extensional material was used.

TEST DATA

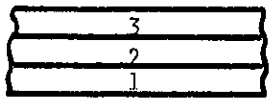
To illustrate the properties previously discussed, several tests were performed. All testing was completed with sample laminates tested on a B&K complex modulus apparatus.

Figure (7) shows a comparison of two laminates tested as a function of temperature. All data points are taken at a frequency very near 100 Hz. The curve marked "single material" is a conventional three-ply laminate made with a damping material which has peak damping at approximately $10^{\circ}F$. The data marked "two materials" uses two different materials: one with peak damping at $10^{\circ}F$ and another with peak damping at $80^{\circ}F$. For the sample with two materials, the combined thickness was kept the same as that of the single material used in the conventional laminate. For these tests no attempt was made to optimize the configuration; however, the improvement in overall performance is obvious.

The same two laminates were tested as a function of frequency at $60^{\circ}F$. The temperature was chosen to demonstrate the improvement in frequency dependent performance. Both materials have reasonably good damping properties at $60^{\circ}F$. Also, at $60^{\circ}F$ the material which peaks at $80^{\circ}F$ acts as extensional material and the $10^{\circ}F$ material acts as a shear material. The performance improvement shown by this test appears in Figure (8).

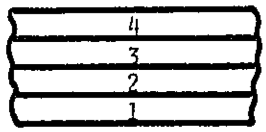
CONCLUSION

This paper has discussed methods for using two polymeric damping materials to substantially broaden the temperature and frequency range of constrained layer laminates. Both theoretical relationships and test results have been shown to illustrate the useful advantages of this concept. This work can clearly be extended to laminates with more than two polymeric layers at the cost of increased theoretical and practical complexity.



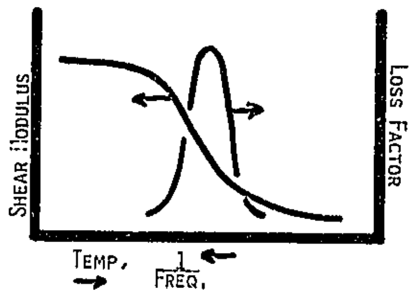
1,3 Structural Skin
2 Polymeric Damping Material

F I G U R E 1

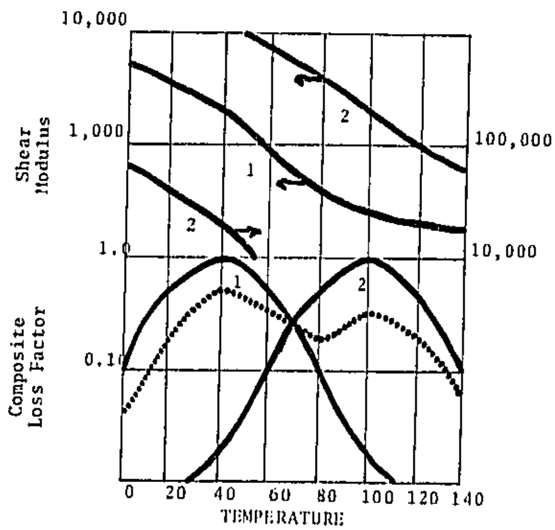


1,4 Structural skin
2 First Damping Material
3 Second Damping Material

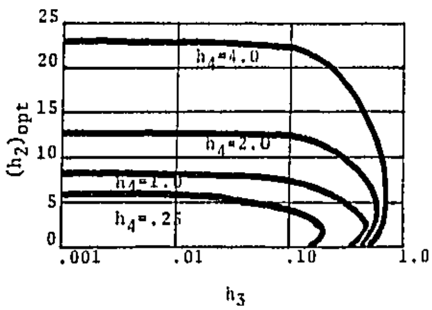
F I G U R E 2



F I G U R E 3

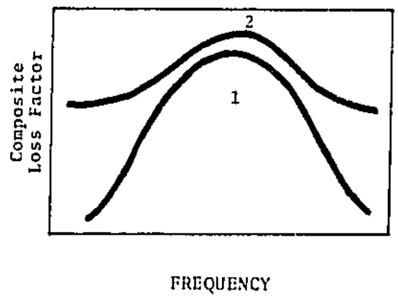


F I G U R E 4



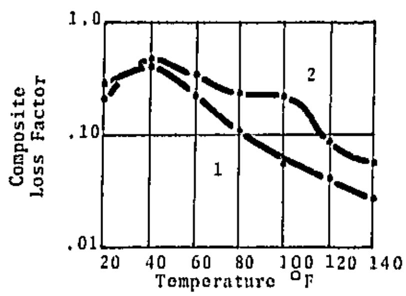
G_2 = 10,000 psi.
G_3 = 1,000 psi.
β_3 = .50

F I G U R E # 5



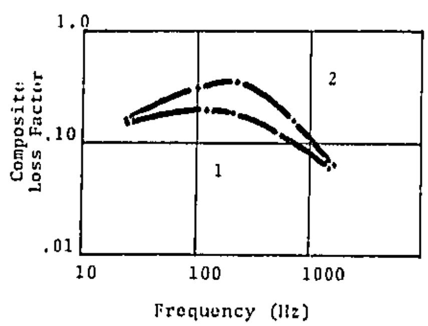
1 Shear Material Only
2 Extensional and Shear Material

F I G U R E 6



1 Single Materials
2 Two Materials

F I G U R E 7



1 Single Material
2 Two Materials

F I G U R E 8

LAMINATED METAL COMPOSITES FOR INDUSTRIAL NOISE CONTROL APPLICATIONS

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Layered composites of planar sheet metal adhesively bonded together using viscoelastic adhesives (i.e. constrained layer laminates) have been known for some time as efficient and effective internally damped materials of construction. While patent references describing this concept first appeared during the 1930's and early 1940's (1-4), patents are still (currently) being issued on variants of the constrained layer laminate principle (5-9). Considering this apparent volume of research and development activity, commercial products based on the constrained layer laminate concept, although they have been and are being offered (10-15), have not yet attained full prominence as noise control materials. While progress is being made, it is believed that the full potential of effectively using and exploiting the principle of constrained layer laminates as materials for noise control has not yet been fully realized.

Systematic research studies have been conducted on the relations between the viscoelastic properties of the interlayer materials and laminate performance (16-18). In other work, the postformability and fabricatability of laminated metal composites were investigated (19) (20). In most cases, studies have concentrated on the effect of polymer viscoelastic properties on laminate damping properties. In further work, different geometric arrangements of planar ply laminate configurations have been examined; an example of such a geometric variation is the so-called spaced damper (21) (22).

Classically, the first study of the dynamic mechanical properties of viscoelastically damped layered sandwich panels was done by Oberst (23) and by Van Oort (24); this work was continued by Oberst and Schommer (17). A composite materials analysis of the vibration damping of a metal beam containing a constrained damping layer was done by Kerwin (25) with the conclusion that during the flexural vibration of the composite beam, shear deformation is induced into the viscoelastic interlayer leading to effective damping. Imparting shear deformation in the viscoelastic layer still remains the dominate mechanism in the engineering design of internally damped structures, products, and devices employing the constrained layer laminate principle.

Most studies on constrained layer laminates have been performed on metal laminates whose plies are planar. However, a review of the literature reveals that laminates prepared using metal plies of different surface geometry can be designed to have unique, multifunctional properties. It has been shown that the mechanical stiffness, vibration damping and acoustical properties of such laminates can be altered by varying the ply surface geometry. For example, laminates employing embossed or striated metal plies have been found to increase the vibration damping effectiveness of metal laminates involving thin viscoelastic interlayers (26). Such laminates can be designed into structural panels that have a decorative surface which is scratch resistant and is inherently vibration damped (27).

In other reported work (28), laminates prepared with woven screens embedded in the viscoelastic layer have been shown to improve vibration damping properties over laminates containing no screens. Other laminates made using perforated metal plies have been observed to have excellent acoustical (e.g. sound transmission loss) properties and mechanical stiffness, for their light weight (29). These laminates have been reported to have a greater strength to weight ratio that solid core metal laminates.

Laminates with corrugated metal plies were reported as early as 1941 (30). These composites were intended for use in aircraft structural applications. A noise control laminate composed of a combination of a corrugated aluminum core sandwiched (using adhesive) between a layer of perforated steel and a planar ply of sheetmetal has recently been developed (31). The special core material is described as a stable, naturally vented, rigid core structure which is made of cross laminated corrugations. If this core material is cut into sheets along a certain plane, and enclosed in the sandwich configuration previously described, the structure resembles a partially-partitioned resonator panel (32). In view of uniqueness of this material, and structural integrity, some acoustical absorption tests were conducted on a sample. In the initial examination, data comparing the experimentally measured random (33) and normal (34) incidence acoustical absorption properties of this material indicated that it can function as a fairly narrow band acoustical absorber. The random incidence acoustical absorption, α_R, data showed an absorption peak occurs in the vicinity of 3000 Hz; α_R = 0.85 to 1.0). The breadth of this peak can be illustrated by reviewing the data: 1000 Hz, α_R = 0.22 to 0.40; 2000 Hz, α_R = 0.52 to 0.92; 4000 Hz, α_R = 0.74 to 0.89; 5000 Hz, α_R = 0.35 to 0.52. It is expected that through changes in configuration, a panel having a much broader absorption peak could be designed.

The normal incidence acoustical absorption measurements were obtained using a Brüel and Kjaer impedance tube. The data were found to parallel those of random incidence with regard to the position of the absorption peak. The peak, however, was narrower and less intense. Upon reviewing the overall capabilities of this new material, it is judged that this configuration has a unique combination of sound absorption, heat dissipation, self-venting, and structural properties which could be developed into an ultimately suitable material for industrial machinery enclosure applications.

This abstract presents a brief review of the nature of laminated metal composites: Their use in industrial noise control applications (plastic scrap grinders, turbine generator housings and hydraulic pump enclosures, etc.) is well known. It is predicted that such noise control materials will become of increasing utility and value in solving in-plant and OEM noise and vibration problems.

ACKNOWLEDGMENT

The author wishes to thank Mr. John Zalas for performing and interpreting the acoustical absorption measurement herein described.

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ATTENUATION OF NON-METALLIC PANELS

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INTRODUCTION

This paper presents measured attenuation data on non-metallic panels, some with 1" thick absorptive foam. The attenuation of these panels cannot be easily calculated because some are nonisotropic, nonhomogenous and their damping is unknown. The paper also provides attenuation data of acoustically "tight" enclosures made from some of the measured, bare panels.

MEASUREMENT PROCEDURES

Figure 1 shows a cutaway side view of the measurement conditions. The 1.08 m^3 pit is excited by a 2" diameter impeller running at 10,000 RPM. The change in the reverberant field sound pressure level in the 180 m^3 reverberation room when a panel is placed over the pit is defined in this paper as panel attenuation. The average random incidence absorption coefficient of the wall surfaces in the pit and in the reverberation room is less than 0.025 in the 1 kHz octave band. Most of the panels measured should show about the same absorption when the pit side of the panel is not lined with foam. Low frequency absorbers which decrease the reverberation time by 4 times at 100 Hz but not measurably at 8 kHz are installed in the 180 m^3 room. The horizontal top edge of the pit had a 1/16" thick, 1" wide rubber tape on which each particular panel was laid. To prevent leakage of sound, a 1/2" thick strip of sealer was used on the edges of the panel. Edge sound leakage was checked meticulously by listening through a 1" ID vinyl tube. The pit is a linearly scaled-down version (approximately 7 times) of the reverberation room. However, while the reverberation room is a parallelepiped, the surfaces of the pit are splayed to enhance the random character of the reverberant field.

Figure 2 shows the not quite rectangular dimensions of the panels. This shape, rather than a rectangular shape, was chosen to decrease the Q of panel eigenfrequencies and also to achieve more eigenfrequencies for a given bandwidth. This results in better averaging and repeatability of the data.

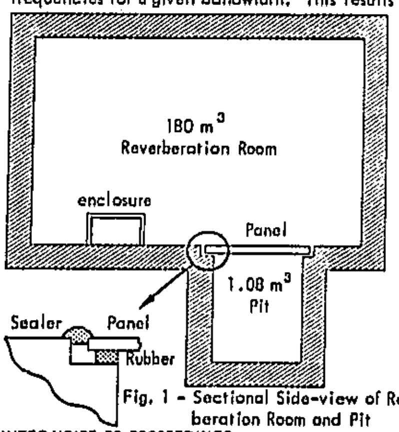

Fig. 1 - Sectional Side-view of Reverberation Room and Pit

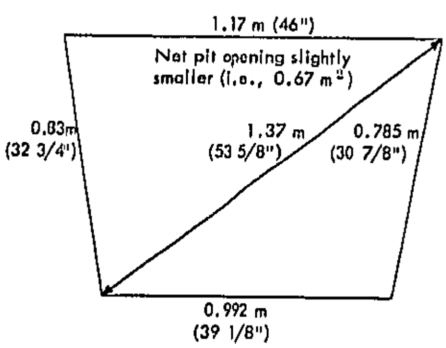

Fig. 2 - Panel Dimensions

Figure 3 shows the 250 through 8,000 Hz octave band sound pressure levels generated in the reverberant field of the 180 m^3 room by the 10,000 RPM source in the pit in the top curve; the lower curve shows the sound levels when a 0.047" thick, cold-rolled steel panel is placed on the pit.

The solid curve in Figure 4 shows the difference between the two curves in Figure 3; i.e., the attenuation of the 0.047" steel panel. The dashed curve shows the attenuation of this steel panel when the reverberation room is excited by a reference sound source blower and the reverberant field in the pit is measured. Reciprocity is within measurement accuracy for the 500 to 8,000 Hz octave bands. In octave bands below 500 Hz, reciprocity fails because the wavelengths approach or exceed the panel and pit dimensions (insufficient or no modes per octave band) and a cavity resonance of the pit/panel also occurs whose frequency and Q is controlled by the particular panel and the pit compliance.

Table I presents data for the panels measured. The lowest frequency f_c where wave coincidence can occur in the panels was calculated according to

$$f_c = \frac{1.16\,c^2}{h}\sqrt{\frac{\rho}{E}}$$

where:

f_c - lower critical frequency at which wave coincidence occurs (Hz)

h - panel thickness (inches)

c - speed of sound in air (feet/second)

ρ - density (pounds mass/cubic foot)

E - Young's modulus (pounds force/square foot)

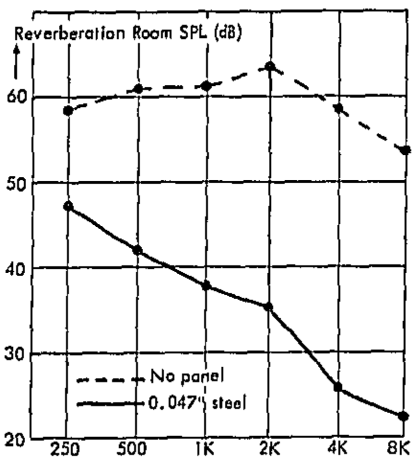

Fig. 3 - Reverberation Room Sound Pressure Level (SPL) with and without Steel Panel Over Pit

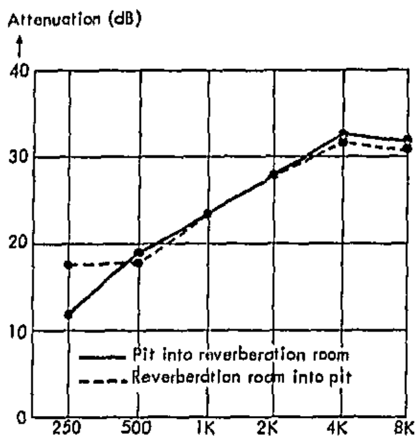

Fig. 4 - Reciprocity Data; Energy from Pit into Reveration Room and Vice Versa

TABLE I – PANEL MATERIALS MEASURED

Panel Material	Young's Mod ($lbsf/ft^2 \times 10^6$)	Density (lbs/ft^3)	Thick (inch)	Weight (lbs.)	f_c (kHz)	dBA and Octave Band (Hz) Attenuation Ref. 0.047" Steel Panel* dBA	250	500	1000	2000	4000	8000
Steel **	4300	480	0.047	17.0	10.3	0(24)	0(12)	0(18.5)	0(23.5)	0(28)	0(32)	0(31.5)
Aluminum	1400	168	0.160	20.0	3.1	1.0	2.0	0	0	2.0	11.0	0
"			0.500	65.7	1.0	+3.5	+4.0	+4.5	+3.5	+2.0	+3.5	+6.0
Superplastic (SP)	1150	320	0.068	15.0	10.9	1.0	0.5	1.0	1.0	1.0	1.0	1.0
SP heat treated	1730	320	0.070	15.5	8.9	2.0	1.0	1.5	2.0	2.0	2.0	3.0
ABS Foam	13.7	40.5	0.375	11.4	6.9	2.0	0	2.0	2.0	2.5	2.0	4.0
" "	13.7	42	0.750	23.8	3.4	+1.5	+3.0	+2.5	+3.0	+2.0	3.0	3.0
ABS sheet	43.2	91	0.125	8.6	17.1	5.0	·3.5	4.5	4.5	5.0	4.0	3.5
Foam core	10	8.6	0.185	1.2	7.4	17.0	9.0	14.5	17.5	20.0	22.5	17.5
Glass-filled	220	119	0.125	11.2	8.7	3.5	2.0	2.5	3.0	4.0	3.5	7.0
Polyester	220	127	0.250	23.0	4.4	0.5	+1.0	+0.5	+0.5	0.5	8.5	0.5
"Plexiglas"***	64.8	84	0.125	7.9	13.0	5.5	3.0	6.0	6.5	7.0	6.0	5.0
"	64.8	74	0.250	14.0	6.5	3.0	3.0	1.0	1.5	2.5	+1.0	4.5
Polyester	17	32	0.375	9.1	5.4	6.0	4.0	4.0	5.5	7.5	13.5	11.0
"	35	43.5	0.375	12.3	4.4	4.5	2.0	2.0	3.5	4.5	12.0	7.0
"	65	55.0	0.375	15.5	3.6	2.5	1.0	1.0	1.5	3.5	13.5	13.5
"	17	39.6	0.750	22.4	3.0	3.0	+1.5	1.0	1.0	5.5	10.5	5.0
"	27	44.0	0.750	24.9	2.5	2.5	+1.0	0.5	1.5	4.5	10.0	2.5
"	65	55.0	0.750	30.9	1.8	1.0	+2.0	+1.0	1.5	5.0	6.0	4.5
PVC	61.3	95.5	0.250	18.0	7.4	0	+2.5	0.5	+0.5	0.5	2.5	4.0
Safety glass	1440	138	0.285	29.6	1.6	+1.5	+5.0	+3.0	+2.0	+2.0	4.5	+3.0

* Digit without sign means less attenuation, a + means more attenuation than 0.047" steel.
** Figures in paranthesis are panel attenuation for 0.047" steel; for other materials add or subtract as applicable.
*** Registered trademark, Rohm and Haas Co., Philadelphia, Pa.

In Figure 5, the attenuation data from Table I for three different panels are compared: steel, skinned polyurethane, and acrylonitrile butadiene styrene (ABS) sheet (not foam). Note that the weight and thickness of the panels are different. The skinned polyurethane demonstrates how detrimental low densities are to the attenuation, resulting in a low critical frequency for the polyurethane panel. Conversely, the ABS sheet shows relatively high attenuation for its weight throughout the octave bands because its critical frequency is about 17 kHz. Other non-metallic panels showing fairly high attenuation are glass-filled polyester (GFP), polyvinyl chloride (PVC), ABS foam, "Plexiglas" and safety glass.

Increasing the thickness (i.e., the weight of the same panel) by a factor of 2 resulted in an increase of 2.74 to 5.72 dB in attenuation at low frequencies (500 Hz octave band) for aluminum and Plexiglas, respectively. The average figure for all materials was 3.66 dB while theory predicts 5.5 dB*. The primary reason for this discrepancy is probably attributable to the panel boundary conditions which may dampen thinner, "live" panels (e.g., steel, aluminum) more than thicker, "dead" (e.g., ABS) panels. However, such boundary conditions may often exist in actual machine cover panels. Metallic panels of course have the additional advantage of providing electromagnetic shielding.

Figure 6 shows the excess of measured vs. calculated transmission loss (TL) figures according to the plateau method; this was one of the reasons why this study was undertaken. For octave bands below 2 kHz, the measured transmission loss is 5 to 8 dB higher than calculated. For octave bands above 1 kHz, the measured TL may be 3 to 14 dB higher than calculated according to the plateau method.

Because absorptive materials very often are used for lining the internal side of machine panels, the increase in panel attenuation with 1" thick absorptive foam with a density of 6 lbs/ft^3 was measured,

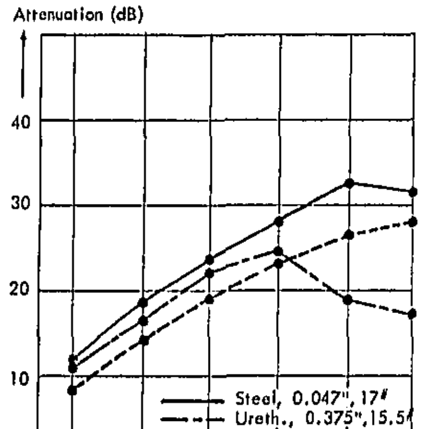

Fig. 5 - Attenuation of Three Panels of Differing Material, Thickness and Weight

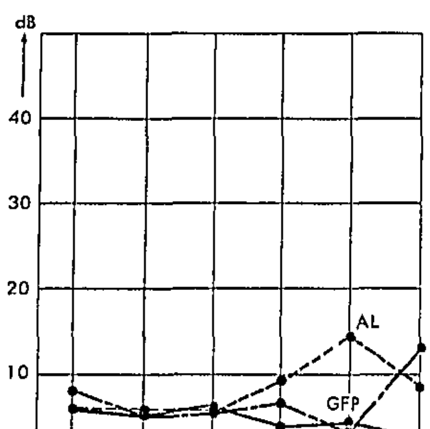

Fig. 6 - Excess of Measured vs. Calculated Transmission Loss (Plateau Method) for Three Panels

* See for example: Leo F. Beranek, "Noise and Vibration Control," McGraw Hill: 1971.

Figure 7 shows this increase for (a) steel, (b) aluminum and (c) glass-filled polyester (GFP). The dashed curves show the increase when the foam sheets are relatively loosely attached to the panels with double coated tape, and the solid curves show the increase in attenuation when the foam was glued onto the panel. The data clearly illustrates the importance of a good bond between the panel and the foam — at least for this type of relatively high density foam (6 lbs/ft^3). Note the 15 dB increase in attenuation of the aluminum panel in the 4 kHz octave band. This is the band in which (see Table I) this panel shows a 11 dB dip in attenuation as compared to the steel panel. The addition of the foam when glued onto a panel has four effects: first, it will reduce the diffusivity of the sound field impinging on the panel — i.e., more direct (approximation of normal incidence) sound will strike the panel; second, the intensity of the sound field striking the panel will decrease; third, the foam lining (total weight 4.3 lbs.) will noticeably dampen the "live" panels; and fourth, the foam will increase the mass of the "composite" panel. The third effect primarily accounts for the high increase in attenuation of the aluminum panel as shown in Figure 7b for the 4 kHz octave band.

Of further interest is scaling between panel attenuation data and attenuation of tight enclosures built from the panels. If the absorptivity of the internal surfaces of a tight enclosure is low, the intensity of a reverberant sound field will predominate at the internal surfaces. Doubling the volume of, e.g., a cubicle enclosure, then means a 1.59 times increase in internal surface area, but the reverberant sound field intensity would be reduced by the same amount. The product of average reverberant sound field intensity and enclosure surface area should, therefore, remain constant when all dimensions of a given enclosure are changed in the same proportion. The enclosure attenuation can thus be assumed to be independent of enclosure volume. For the same conditions, this is also true if the absorptivity of the internal enclosure surfaces is high. In this case, the direct sound field intensity will predominate at the internal surfaces. Increasing the enclosure volume will decrease the direct average sound field intensity at the internal enclosure surfaces, again making the product of average intensity and surface area the same; consequently, the enclosure attenuation should remain the same. At this point it should be added that the attenuation of a panel decreases as the reverberant character of the impinging sound field increases. The decrease in panel attenuation going from, e.g., an enclosure with high absorptivity (field incidence) to one with low absorptivity (random incidence) can be as much as 5 dB.

The attenuation that a tight enclosure provides also depends upon noise source location, its radiation impedance and the impedance of surfaces in source proximity. These factors are assumed constant in this paper.

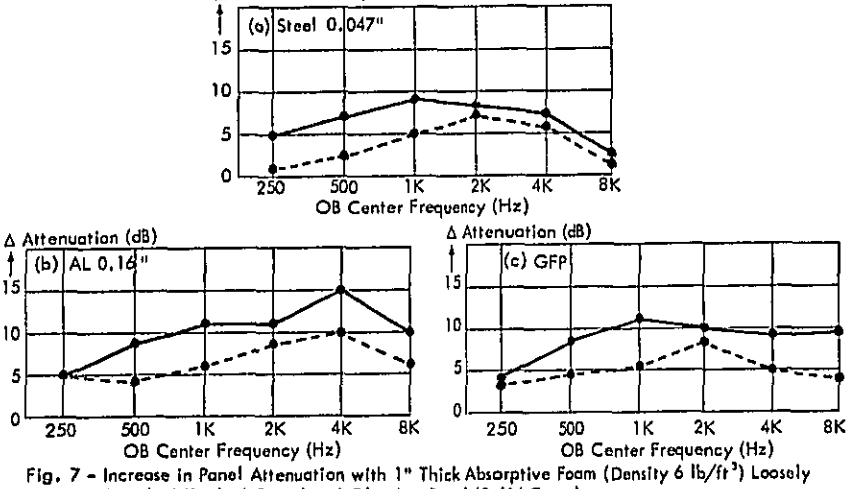

Fig. 7 - Increase in Panel Attenuation with 1" Thick Absorptive Foam (Density 6 lb/ft^3) Loosely Attached (Dashed Curve) and Glued to Panel (Solid Curve)

Table II shows the difference between bare panel and bare enclosure attenuation measured with the 10,000 RPM source located towards the center of an enclosure (dimensions: 0.26 m x 0.41 m x 0.69 m.) The enclosure had no bottom and was placed on the floor in the 180 m^3 reverberation room (see Fig. 1). To prevent leakage of sound a sealer was used along the bottom edge of the enclosure. While relatively small, the data in Table II are an additional set of scaling values to be added to the panel attenuation data in Table I to more precisely estimate the attenuation of a tight enclosure. Table II data for steel should be added for "live" metallic panels; the data for GFP should be added for non-metallic panels. Even with the limited set of data in Table II, it should be safe to estimate that the accuracy with which the attenuation of bare enclosures can be assessed is within ± 1.5 dB for the 500 through 8,000 Hz octave bands. For the 250 Hz band the size of the panel and measurement setup is too small for this accuracy; the accuracy may be about ± 3 dB. The ± 1.5 dB accuracy is considerably better than attenuation based on, e.g., calculations according to the plateau method. More precise predictions of enclosure attenuations can thus be made.

TABLE II — PANEL-ENCLOSURE ATTENUATION (dB)

		Octave Band Hz				
Material	250	500	1K	2K	4K	8K
Steel 0.047"	-1.5	+1.5	-0.5	-1.0	0	+0.5
GFP 0.16"	+0.5	+2.5	+1.0	+1.0	+2.0	-0.5

CLASSIFICATION AND PERFORMANCE OF COMMERCIAL VIBRATION ISOLATORS

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INTRODUCTION

The use of vibration isolation systems to interrupt transmission paths of noise and vibration has recently become a major aspect of engineering design. Vibration isolation systems comprise a well established product area. However, these products have not been classified by type and manufacturers have not adopted uniform test procedures to obtain product data. Comparison of products is possible, therefore, only by review of manufacturer's catalogs. This investigation provides a classification of various commercial vibration isolators and an independent comparison of performance based on uniform testing procedures.

A simplistic approach is usually followed in design for vibration isolation. An isolator is selected such that the resonant frequency of the system is much less than the forcing frequency. The stiffness coefficient, k, of the isolator is the characteristic of primary importance. Dissipation by an isolator, characterized by the damping factor, z, is important near resonance but adversely affects transmission in the usual design range of frequency.

CLASSIFICATION

A large variety of prefabricated isolator systems, obtained from seven major manufacturers was employed for the study. Specific company names are not revealed because of prior agreement with firms who supplied systems for comparative testing. Significant differences in quality and performance of products of the same type from individual manufacturers was obvious in many cases. Classification of isolators is presented in Chart 1. The sketches are intended to generally describe particular types of isolators and do not represent a specific product. The classification is given by letter and number designation, "A", "A1", "B", "B1", etc.

STIFFNESS TESTS

An Instron load-deflection instrument was employed to obtain stiffness data. It appeared that many manufacturers obtained k values by compressing isolators in excess of specified maximum loading and using a linear approximation. This tends to underestimate the value for k and imply a lower resonant frequency. In the present study, the best approximation to k was assumed to be the value found from compression and relaxation of the isolator in the specified load range as illustrated in Figure 1. Sample results are presented in Table 1 together with dynamic test data.

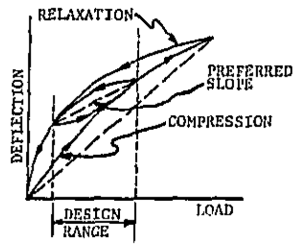

FIGURE 1: STIFFNESS DETERMINATION

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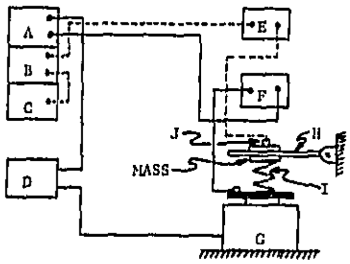
VIBRATION TESTS

In dynamic tests, the base of the isolator was attached to a shaker table and loaded by attachment of a weight to the top of the isolator. Accelerometers were attached to both the mounting base and the top. Constant acceleration of the mounting plate was attained through compressor circuitry (servo-control loop) at an oscillator for a frequency sweep of 2 to 2,000 Hz. An acceleration amplitude difference was monitored on a graphic level recorder which provided transmissibility, TR, directly and indicated resonant frequencies. See Figure 2.



LEGEND
— SERVO CONTROL LOOP
--- SYSTEM OUTPUT LOOP FOR DATA RECORDING
A BEAT FREQUENCY OSCILLATOR/COMPRESSOR
B AUDIO FREQUENCY SPECTROMETER
C GRAPHIC LEVEL RECORDER
D SHAKER PRE-AMPLIFIER
E PRE-AMPLIFIER WITH INTEGRATION CIRCUITS
F PRE-AMPLIFIER
G SHAKER UNIT WITH MOUNTING TABLE
H STABILIZER SYSTEM
I TYPICAL VIBRATION ISOLATOR
J TYPICAL ACCELEROMETER

FIGURE 2: TEST SET-UP

DAMPING FACTOR

The sharpness of resonance technique[2] was used for obtaining damping factors. The transmissibility, TR, as a function of forcing frequency, ω, with resonant frequency, ω_n, and damping factor, z, as parameters for a single degree of freedom system is

$$TR = \frac{\sqrt{1 + (2z\omega/\omega_n)^2}}{\sqrt{[1 - (\omega/\omega_n)^2]^2 + [2z\omega/\omega_n]^2}} \qquad (1)$$

At the half-power points (3db down) for resonance, $\omega = \omega_n$,

$$TR = \frac{\sqrt{1 + 4z^2}}{2\sqrt{2}\, z} = \sqrt{\frac{1 + 4z^2}{8z^2}} \qquad (2)$$

The frequency ratios at the half-power points are found by equating (1) and (2). Omitting terms which involve z raised to powers greater than four, the ratios are

$$\frac{\omega_{1,2}}{\omega_n} = \sqrt{\frac{1 + 2z^2 - 8z^4 \pm 2z\sqrt{1 + 9z^2 - 40z^4}}{1 + 4z^2 - 32z^4}} \qquad (3)$$

The frequency ratios are found from the laboratory test TR plot, Figure 3, and z can be obtained, with computer aided calculation, from equation 3. Typical data are presented in Table 1.

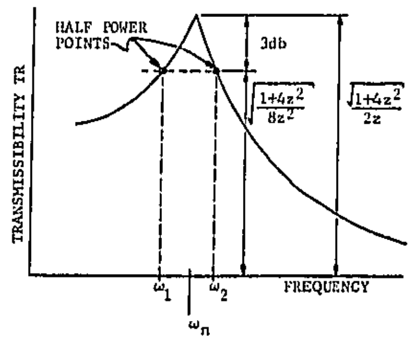


FIGURE 3: HALF POWER POINTS AT RESONANCE

TEST RESULTS

The data presented in Table 1 are typical of results found for the entire test program[3] in which more than a hundred isolators were tested. Tests on four samples of the same type and from the same supplier were made whenever possible. Agreement of results between groups of samples from a particular manufacturer was generally good with deviations of less than 20 percent in most cases. All samples were tested within the load range specified for the product. Stiffness coefficients supplied with samples are tabulated for comparison with test data.

TABLE 1
PERFORMANCE DATA

ISOLATOR TYPE	k_{MFGR} (LB/IN)	k_{EXP} (LB/IN)	TEST LOAD (LB)	RESONANT FREQUENCY (Hz)	DAMPING FACTOR
A	45	45	27.5	6.5	0.02
A	50	53	32.0	5.0	0.00
A	75	66	32.4	5.0	0.00
A	77	83	27.1	6.0	0.00
A	200	195	32.0	6.3	0.02
A1	175	162	42.7	22.0	0.34
B	25	30	27.0	3.0	0.19
B	50	53	22.0	5.0	0.00
B	77	80	27.0	4.0	0.00
B	100	101	33.5	5.9	0.11
B	102	84	27.5	6.0	0.00
B1	55	50	35.5	6.0	--
C	112	278	42.0	13.0	0.07
C	180	525	42.0	18.0	0.18
C	300	410	42.0	20.0	0.17
C	500	945	42.0	20.0	0.12
C	790	795	41.0	19.5	0.15
C1	650	1113	42.0	21.0	0.19
C2	750	800	37.0	19.0	0.05
D	660	2290	42.0	29.5	0.09
D	1300	2100	42.0	30.5	0.11
D	3000	4700	42.0	40.0	0.40
D1	1330	1925	42.0	33.5	0.31
D2	360	655	42.0	19.0	0.15

DISCUSSION

The stiffness coefficients, k, obtained experimentally were in approximate agreement with catalog data for only 17 percent of the isolators tested. The majority of suppliers gave lower k values than obtained by testing. Although 60 percent of the isolators had k values larger than indicated for the product, lower k values were observed in almost one-fourth of the tests. Agreement between experimental and catalog data was good for A and B type isolators whereas all k values for C and D types were larger than values given.

The spring isolators demonstrated "wave effects" with excessive noise generation at one or more frequencies. The lowest frequency inducing the wave effects occurred between 150 and 350 Hz. The A type isolators also had noise generation at points of connection of spring coils with upper and lower plates and between lower coils during heavy loading. Noise generation from inner part suspensions was evident in B type isolators. Noise pads, commonly adhered to bottom mounting plates were generally ineffective. The intensity of airborne noise generation was sufficient to require protective "ear muffs" for operators during testing.

Unlike the spring systems, isolator types C and D did not display resonances comparable to theoretical undamped natural frequency calculations. For these types, resonance consistently occurred at a higher frequency than predicted. This is in disagreement with single degree of freedom theory where damping shifts the resonance peak to a frequency lower than the undamped natural frequency. Dynamic excitation of the system apparently increased the effective stiffness. In view of the extent of discrepancy in predicting resonance, it is recommended that designers use stiffness coefficients roughly four times greater than those given in manufacturers catalogs for C and D type systems.

The data of Table 1 indicate that dissipation in isolators cannot be assumed negligible in many isolator systems. When damping factors are not provided by suppliers for isolator systems, the results from this study can be used to estimate values. In the same cases, manufacturers should probably amend their catalog data to suggest the possible range of damping factors, available from Table 1, which might exist in their products. The user of the product should be made aware if damping is not negligible since isolation design could be significantly affected. For example, at $\omega = 4\omega_n$ and $z = 0.2$, the force transmitted is approximately double the value for no damping.

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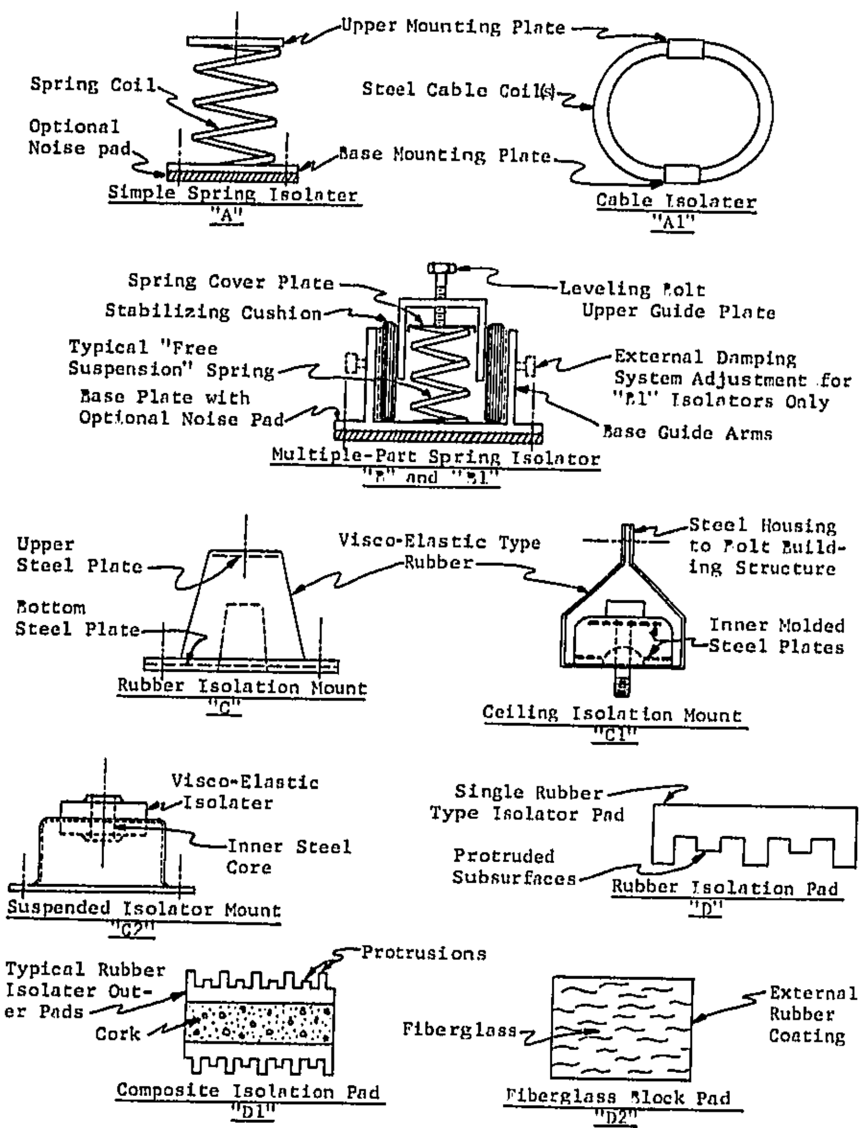
1. "Buyer's Guide, Noise and Vibration Control Materials", Vol. 5, No. 7, Sound and Vibration, July, 1971.

2. W. T. Thompson, Vibration Theory and Applications, Prentice-Hall Inc., Englewood Cliffs, N. J., 1965.

3. S. G. Harvey, Investigation of Performance Involving Typical Commerical Vibration Isolation Systems, Master's Thesis, University of Florida, 1972.

CHART 1

CLASSIFICATION OF VIBRATION ISOLATORS



NOISE REDUCTION PROPERTIES OF SELECTED PIPE COVERING CONFIGURATIONS

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INTRODUCTION

A selected group of nineteen pipe coverings of various materials and configurations was tested under identical conditions of noise generation within the pipe. The candidate coverings represented those treatments considered noise control standards as well as new products with apparent potential.

The purpose of these tests was to evaluate the noise reduction properties of each pipe covering over a wide range of frequency and to assess the relative performance of these treatments with regard to the installed cost of materials and application for each.

The results of these tests provided guidelines for the selection and application of pipe coverings for noise control on the basis of noise reduction and cost of installation so that the most effective, least expensive covering could be selected for each pipe noise radiation problem. A detailed discussion of the tests and results follows.

DESCRIPTION OF TEST EQUIPMENT AND CONDITIONS

A typical materials handling system including fan, piping and fittings was installed at the Riverbank Acoustical Laboratory (IITRI) acoustical testing facility in Geneva, Illinois (see Figure I). To provide a worst case based upon our experience of airborne noise generation and structural excitation of the pipe wall, standard metal B-B's were circulated in the material handling system at a controlled rate monitored acoustically at the material handling fan discharge area.

The test results of Figure II made with the bare pipe control sample (four inch diameter, Schedule 5, Type 304 stainless steel welded pipe) confirm that this test configuration produced significant levels of all component sources of pipe noise; namely, dynamic excitation of the pipe wall by material impact on the inner surface, airborne noise in the pipe as generated by the fan blade passing frequency and associated turbulence generation and boundary layer noise producing effects along the pipe.

Each test specimen covered three feet of pipe in the reverberant test chamber. The remainder of the pipe and fittings in this room was covered with a heavy asbestos base insulation six inches thick which was, in turn, wrapped with six wrappings of one pound per square foot density vinyl-lead material. The transitions through the primary wall and between the heavy lagging and test samples were all sealed with a thick, dense mastic compound to ensure that all significant noise entering the test room was being transmitted through the coverings under test. The measurements were performed in accordance with the applicable portions of the American Society for Testing and Materials Standard E336-67T.

TEST SAMPLES

A brief description of the samples tested shows the variety of materials and configurations evaluated, with the selection based upon the present wide application to piping noise control problems and the potential of new materials for application to such problems with the need for a performance-cost optimum because of the large investment to be made in this aspect of noise control. The sample numbers are those assigned by the Riverbank Acoustical Laboratory.

1. NR70-23 - 1 inch thick molded fiber glass (4 lbs/cu ft) with standard flame retardant aluminum foil covering.
2. NR70-24 - 2 inch thick molded fiber glass (4 lbs/cu ft) with standard flame retardant aluminum foil covering.
3. NR70-25 - 2 inch thick molded fiber glass (4 lbs/cu ft) covered with a 3/4 inch thick urethane foam bonded to lead sheet (1 lb/sq ft) with outer covering of 1/4 inch of urethane foam (the 3/4 inch thick foam in contact with fiber glass).
4. NR70-26 - 1 inch thick molded fiber glass covered with the same foam-lead-foam sandwich described for sample NR70-25.
5. NR70-27 - A single layer of the 3/4 inch urethane foam-lead septum (1 lb/sq ft) - 1/4 inch urethane foam with heavy vinyl cloth covering - 3/4 inch foam to pipe.
6. NR70-28 - 3 inch thick molded fiber glass (4 lbs/cu ft) covered with a single layer of lead impregnated vinyl (0.87 lbs/sq ft).
7. NR70-29 - Same as NR70-28 with 2 inch fiber glass.
8. NR70-30 - Same as NR70-28 with 1 inch fiber glass.
9. NR70-31 - 3.09 inches thick heavy thermal insulation comprising amosite asbestos, sodium silicate binder and diatomaceous silicon filler covered with one layer of heavy asphalt roofing felt.
10. NR70-32 - Same as NR70-31 except 2.03 inches thick.
11. NR70-33 - Same as NR70-31 except 1.03 inches thick.
12. NR70-34 - Special Du Pont Co. laminated pipe.
13. NR70-35 - 1 inch molded fiber glass covered with special Du Pont Co. laminated pipe as per NR70-34.
14. NR70-36 - 2 inch molded fiber glass covered with special Du Pont Co. laminated pipe as per NR70-34.
15. NR70-37 - 3 inch molded fiber glass covered with special Du Pont Co. laminated pipe as per NR70-34.
16. NR70-38 - Sample NR70-36 was installed on sample NR70-34.
17. NR70-39 - Six layers of Du Pont Typar® spunbonded polypropylene sheet wrapped continuously around bare pipe covered with aluminum foil tape - 7-1/2 mil pressure sensitive with an overlap of 1/2 inch per turn.
18. NR70-40 - A single spiral wrapping of the 7-1/2 mil pressure sensitive aluminum foil tape with 1/2 inch overlap directly on the pipe.
19. NR70-41 - Same as NR70-31 except 5.22 inches thick.

A typical sample under test is shown in Figure III.

TEST RESULTS

The acoustical test data was used in conjunction with cost data for each sample to determine the optimum coverings in terms of maximum noise reduction performance for least installed cost.

There emerged from this study three clearly superior pipe covering configurations for broad-band frequency noise control categorized as low, medium and high noise reduction quality and four of the pipe covering samples tested were greatly superior at some particular one-third octave band center frequency or over a specific, limited range of frequency for the related cost.

The recommended pipe coverings are categorized with the related noise reduction data in Table I and the special limited frequency range optimums are shown in the footnote to this table.

The cost differential between the low, medium and high noise reduction pipe coverings is about $1.25/ft in each increment from low to high noise reduction. For the special limited frequency optimums the premium cost justifies the selection when the noise to be controlled is in the specified range of frequency.

TABLE I - OPTIMUM PIPE COVERINGS

1/3 Octave Band Center Frequency in Hertz (Hz)	Low Noise Reduction in dB ref. $0.0002\left(\frac{Dynes}{CM^2}\right)$ Sample NR70-23	Medium Noise Reduction in dB ref. $0.0002\left(\frac{Dynes}{CM^2}\right)$ Sample NR70-30	High Noise Reduction in dB ref. $0.0002\left(\frac{Dynes}{CM^2}\right)$ Sample NR70-29
200	1.8 dB	2.5 dB	2.0 dB
250	2.5	3.0	2.5
315	0.5	1.5	1.0
400	(-1)	1.0	1.0
500	1.5	5.0	4.5
630	0	3.0	3.5
800	1.5	4.0	4.9
1,000	4.8	12.0	13.5
1,250	6.5	17.0	17.5
1,600	9.5	18.0	19.0
2,000	13.8	24.0	26.0
2,500	14.5	23.5	28.0
3,150	30.5	26.0	28.0
4,000	22.0	28.0	31.0
5,000	26.5	33.0	35.0
6,300	30.0	35.0	37.5
8,000	31.0	36.0	38.5
10,000	33.0	35.0	39.0

Footnote: Special limited frequency range optimum pipe coverings

Sample No.	Noise Reduction*	Band Width of Superiority
NR70-23	30.5 dB	3,150 Hz center frequency
NR70-33	27 to 38 dB	5,000 to 10,000 Hz
NR70-37	38.4 dB	8,000 Hz center frequency
NR70-38	4 to 5 dB	200 to 630 Hz

* dB Ref. $0.0002\left(\frac{Dynes}{CM^2}\right)$

CONCLUSIONS

Some important facts have been learned as a result of this experimental program and the fact that clear-cut optimized noise reduction pipe coverings did emerge is a welcome result given the large potential expenditure for such materials and costs of installation as a significant part of our system - oriented plant noise control programs.

Many of the materials that we have tested present other major detractions not specifically included in the study such as asbestos material matrices, lack of structural integrity and poor appearance. Fortunately, those that performed well acoustically do not have these severe liabilities and this further distinguishes the optimized pipe coverings from the other samples tested.

Some interesting observations can be made with the data analysis on hand.

1. The lead impregnated vinyl distinguishes itself as a covering and the function of the fiber glass is to space the covering at an optimum distance from the pipe surface in a double-wall configuration and further to suppress double-wall resonance by its absorption characteristics.

2. The three- and five-inch thick amosite asbestos samples, NR70-31 and NR70-41 exhibited poorer performance than the one-and two-inch thicknesses on an overall basis.

3. The two-inch fiber glass spacer was consistently superior to the three-inch fiber glass spacer in all cases of comparison based upon the four-inch pipe.

4. None of the pipe coverings tested exhibited good noise reduction below 500 Hz which is indicative of the real problem in controlling pipe radiation noise at these low frequencies.

5. Proper sealing of all joints in the pipe covering envelope is a critical part of achieving the available noise reduction inherent in the covering configuration.

6. Only the constrained layer laminated pipe of Du Pont design and manufacture represented a true case of structural damping. The "damping" often attributed to low modulus, wrapped insulation by manufacturers and their agents is a misnomer in that very little energy is removed from the pipe wall structure.

It is hoped that the presentation of the results of this testing program will result in substantial savings in noise control programs involving the insulation of piping. We further hope that it will provoke an interest in manufacturers of insulation products to make those optimum configurations available to industry at a reasonable cost.

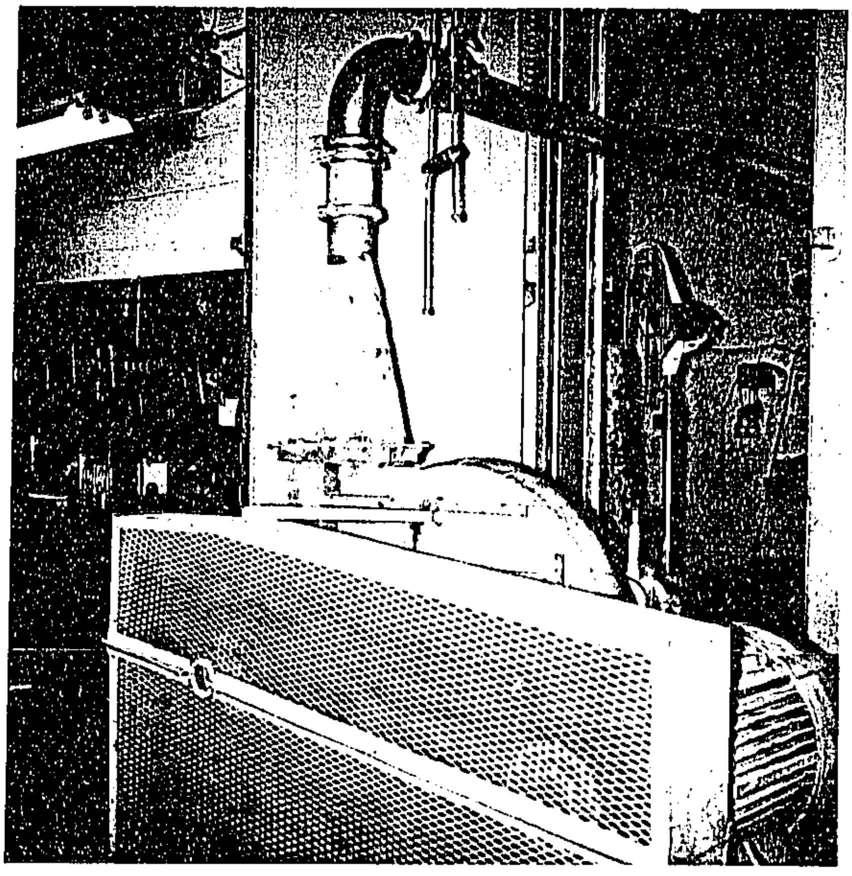
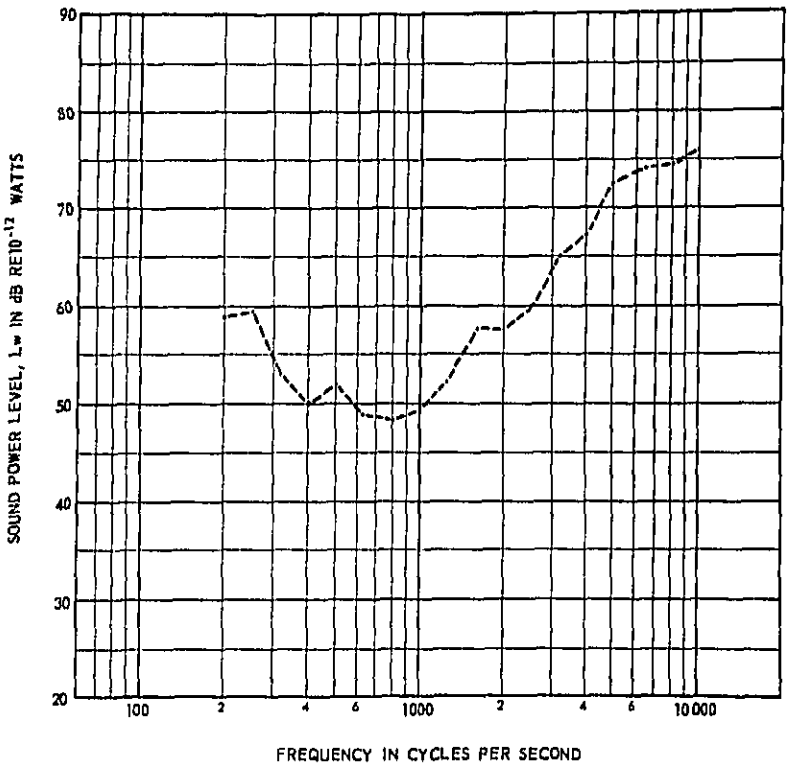


FIGURE I - MATERIAL HANDLING EQUIPMENT INSTALLATION - RIVERBANK LAB

NOISE REDUCTION OF PIPE COVERINGS
TEST RESULTS - IITRI RIVERBANK ACOUSTICAL LABORATORY
AUGUST 1970



TEST SAMPLE NO.* NR70- 42A (A designates reference level for first sample group)

DESCRIPTION:

Four inch diameter, schedule 5, Type 304, stainless steel welded pipe (0.083 inch wall, 4½ inch outside diameter) - bare pipe. See Appendix I for complete description.

*RIVERBANK ACOUSTICAL LABORATORY ASSIGNED TEST NUMBER

FIGURE II

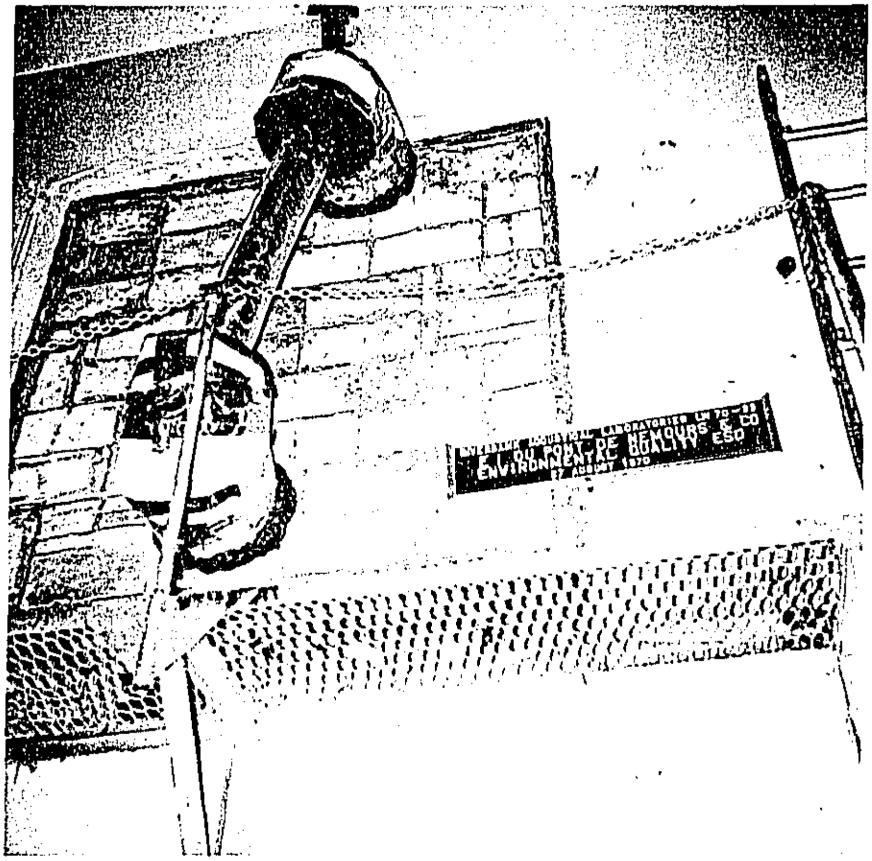


FIGURE III - TYPICAL PIPE COVERING SAMPLE UNDER TEST (NR70-23)

MACHINERY NOISE (I)

CONTROL VALVE NOISE AND ITS REDUCTION
STATE OF THE ART

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INTRODUCTION

The cost to the nation resulting from excessive exposure to industrial noise is enormous. As far back as 1961, Time Magazine placed the cost of noise at over $2 million per day in Workmen's Compensation for noise related injuries, lost man hours and decreased work output. It has been estimated that since 1950, closed court cases involving industrial noise have resulted in paid out claims in excess of $15 million dollars.

The recently enacted (April, 1971) Federal Noise Control Legislation in the form of the Occupational Safety and Health Act (OSHA) has placed the responsibility for noise control clearly on the shoulders of industry with stiff economic penalties for failure to comply with the specific noise codes.

Control valves, pressure regulators and throttling devices are, and have been for many years, integral and irreplaceable devices in our industrial plant technology. They are the pressure and flow controlling elements in all gas and liquid handling installations, and serve five primary functions: (1) Starting or stopping flow, (2) Regulating or throttling flow, (3) Preventing back flow, (4) Regulating pressure, and (5) Relieving excess pressure. In the performance of any or all of these five functions, valves and regulators are known to radiate noise levels that in many cases are harmful to workers who are required to be in their vicinity. The extent to which valves can contribute to excessive industrial noise can be appreciated by citing a published statement from a major valve manufacturer that, ". . . about 60% of the industrial control valves sold today for compressible fluid service will generate noise in excess of the 90 dB(A) 8-hour exposure permitted under the Walsh-Healey Act . . ." [1]. This sound pressure level restriction is identical to the OSHA requirement.

AERODYNAMIC SOURCES OF NOISE IN VALVES WITH COMPRESSIBLE FLUID FLOW

Control valves, blow-off valves and pressure regulators are essentially throttling means or energy dissipation devices. As such, the energy contained in the compressible fluid is converted into sonic streams at the valve's orifice. These jets in turn are slowed to the mean stream velocity by the turbulent mixing mechanism between the jet and the cavity gas in the space aft of the orifice after many diameters of length distance downstream from the orifice. It is clearly understood that the noise of valves is caused by this confined jet mixing aft of the orifice and the subsequent disturbing effect on the surrounding fluid. Sound generated by compressible fluid motion, thus aerodynamically, is a complex area of acoustics. It was only in 1952, with the publication of M. J. Lighthill's [2] classical paper that a first complete amd successful theory of aerodynamically generated sound was presented. As is the case in many of the powerful and successful theories of physics the resultant formulation is well known, but the limitations of the theory are often overlooked.

However, before proceeding with a brief review of the relevant aspects of the theory of aerodynamic sound a brief discussion of the classical sound sources in acoustics is presented. The different mechanisms of sound generated by flow can effectively be described by recourse to the analogy of these 3 classical sources radiating into a free field; the monopole, the dipole, and the quadrupole.

The monopole source can be visualized as a pulsating sphere with the radius varying harmonically. Any pulsating body, regardless of its shape may be approximated by a small sphere if its characteristic diameter is less than about one-third of the radiated wavelength. The monopole source is omnidirectional with the sound pressure at specific radius being dependent solely on the source strength. Intuitively the unsteady mass flow through an orifice as a result of flow instabilities could well result in monopole type sound sources.

The dipole type source can be described either by an oscillating sphere or as two - 180° out of phase monopole sources, each separated by a small distance. This representation is also equivalent to a source-sink model in fluid mechanics. The concept of an oscillating sphere leads one to relate dipole sources to fluctuating forces or momenta acting on the surrounding medium which in turn relates to the motion of fixed surfaces, such as airfoil fluctuating lift forces, vibrating surfaces and fluctuating fluid momenta. The far field pressure of a dipole source has a directional quality which is dependent on, θ, the angle relative to the axis of the sphere with no sound pressure noticable in the direction normal to the axis. The much stronger frequency dependence of the dipole compared to the monopole source should also be noted.

The quadrupole source can be thought of as either 2 dipoles of opposite sign separated by a short distance small compared to radiated wavelength, or 4 monopoles (source-sink arrangements) of proper signs and symmetrical spacing. The physical significance of the quadrupole is that it allows one to model the sound radiation from a distribution of dipoles (or monopoles) whose instantaneous volume (monopole) and force (dipole) source strength is zero. Thus, the net radiation is due to finite separation of the individual dipole (or monopoles) and the resultant incomplete destructive interference. Depending on the source arrangement, either longitudinal or lateral quadrupoles can result. The longitudinal quadrupole is the result of an "in-line" arrangment of 2 dipoles and is equivalent to direct stress in a fluid element. The lateral quadrupole is the result of a square arrangment of monopoles or side by side for dipoles, and is representative of conditions of fluid shear on a fluid element. As was the case for the dipole, the pressure pattern of the quadrupole source exhibits a distinct directionality in the far field with the sound pressure being proportional to 2θ. Considerations of acoustic radiation efficiency indicate that for the same source strength and frequency, the monopole is the most efficient radiator with the quadrupole source being the least efficient radiator of sound. Intuitively this fact can also be visualized by recognizing that there is no destructive interference for the case of the monopole since there is no interfering source; for the dipole there is some destructive interference as evidenced by the quiet zone along the normal plane already described; for the quadrupole there are two normal planes of quiet zones as a result of far greater destructive interference between sources.

These important simple source considerations lead to the conclusion that mass fluctuation in a fluid can be described as monopole sources, force fluctuations as dipoles, and shear stress fluctuations as quadrupole sources.

The classical paper of Lighthill's deals with the intensity of acoustic radiation from a limited region of free turbulence (shear type disturbances) imbedded in a quiescent medium of infinite extent [2]. The mathematical treatment of the problem is quite complex and can be found in the cited references. The physical meanings of Lighthill's equations are, however, very important. The resultant sound radiation at large distance from the free turbulence is equivalent to that generated by a static distribution of acoustic lateral and longitudinal quadrupoles. Dimensional analysis applied to these results shows that the mean acoustic intensity (the acoustic power per unit area of the sphere at radius R) measured by the observer a distance R from the source is given by the relationship (1), the well known "U^8" power law of free jet acoustics.

$$I_q \propto \rho_o \frac{\ell^2}{R^2} \frac{u^8}{c^5} \qquad (1)$$

In the case of Lighthill's theory, it is permissible, because of dimensional similarity, to interchange u with U.

where u is a characteristic velocity of the fluctuating turbulent flow, and ℓ is a characteristic length usually called the turbulence scale. For flows characterized by a single velocity scale U, such as the mean velocity at the orifice of a reducing valve a characteristic velocity u of the turbulence is typically only several percent of U and thus, even for the conditions of choked flow, so typical of valves with $U/c \approx 1$, the ratio $\frac{u}{c}$ is smaller than unity. Two very important points should be made about equation (1) 1. The well known U^8 law of acoustic intensity dependence on free stream velocity of a free jet is only valid when the observer is located at a distance R from the source that is much greater than ℓ, the characteristic turbulent eddy size. The observer, thus experiences acoustic radiation which intensity decays at the rate of $1/R^2$. 2. The turbulence is imbedded in an infinite medium with any reflecting surfaces at sufficiently large distances so that reflected waves at the observer are of negligible value.

The effects of a rigid enclosure surrounding a finite region of turbulence has only recently been theoretically analyzed [3] with the result that the velocity exponent of the quadrupole source mechanism might tend to be reduced from 8 to 6. Thus, in the light of Lighthill's original assumptions in arriving at the U^8 law, it should be evident that valve noise theories which assume free jet type flow in the orifical part of the valve are in error on three counts: 1. The jet downstream of the reducing section of the valve is enclosed by a semi-rigid acoustically reflecting valve body or pipe, thus a free field does not exist. 2. The walls surrounding the jet may be at a distance R, which is not much greater than the turbulence scale ℓ and the pressure fluctuations that excite the wall may not be acoustic in nature (near field conditions exist). 3. The jet generates turbulent energy of decreasing frequency with increasing distance along the jet stream from the orifice. The interaction of incident and reflected waves, dependent on the pipe vibratory modes, may either amplify or attenuate the mechanisms responsible for sound radiation.

The theory of Lighthill's was extended by Curle [4] to include the presence of solid boundaries near regions of turbulence. His expression for the sound intensity consists of 2 terms, one is identical to the U^8 law of Lighthill's representing the contribution from quadrupole sources, the other term relates to fluctuating force terms and represents dipole sound sources. The sound intensity generated by this second term is of the same order of magnitude as the Lighthill term and from dimensional analysis considerations is given by equation (2).

$$I_d \propto \rho_o \quad \frac{\ell^2}{R^2} \quad \frac{U^6}{c^3} \tag{2}$$

From equation 3, it is evident that at sufficiently low Mach numbers the dipole term may dominate the generation mechanism.

$$\frac{I_q}{I_d} = \frac{U^2}{c^2} \tag{3}$$

It is important to recognize that Curle's results, like Lighthill's, are valid only if R is large compared to the turbulence scale ℓ.

The turbulent flow over a real solid body with flexibility generates not only fluctuating forces, but also fluctuating displacements of the surface. These displacements result in mass fluctuations, which in turn result in monopole type sources with their inherent high radiation efficiency as previously discussed. The far field intensity generated by monopole sources from dimensional considerations is given by the following proportionality relation.

$$I_m \propto \rho_o \quad \frac{b^2}{R^2} \quad \frac{U^4}{c} \tag{4}$$

where b is a characteristic dimension of the vibrating surface provided that the requirement $b \gg L \frac{c}{u}$ is satisfied [5].

PSEUDO-SOUND AND WALL PRESSURE FLUCTUATION

The previous section discusses some highlights of hydrodynamic noise theory in terms of the three basic sound sources generated as a result of turbulence in the flow with a solid (or vibrating elastic) boundary present. All these considerations deal with radiation to a distant observer located in a free (non reflecting) field. This approach has been successful based on experimental evidence, for the case of aircraft, and other free jet applications. However, for the case of valve noise these approaches may not provide significant results because the two most important implicit assumptions are violated: 1. The characteristic scale of the turbulence in the pipe aft of a valve is of the same order as the pipe diameter. 2. The pipe walls are physically located between the sources and the observer. These two circumstances imply that the walls will be excited by the pseudo sound from the sources (meaning that sound pressure and particle velocity from the sources are not in phase) as well as the boundary layer pressure fluctuation and separated regions of flow in the pipe. The pipe by virtue of its mass and elasticity acts as a selective transmitter attenuating some of the disturbances and amplifying others. Separated flows, with subsequent reattachment, have been recently studied [6] with the result that fluctuating wall pressures are five to seven times greater than unseparated flows, and thus cannot be ignored in studying valve noise.

Thus, the effects of the sound generated within the pipe by orifice type jets, separated regions of flow and the pseudo-sound at the walls must all be considered in determining the transmission loss between sources and far field observer outside of the valve-pipe system.

CURRENT PREDICTION METHODS

The Instrument Society of America in its recently published (1971) Handbook of Control Valves [7], contains the two valve noise prediction methods that have achieved qualified success. The first method, based on the work of E. E. Allen of Fisher Controls Company, is an empirical method requiring the user to obtain experimental data for each specific valve class and for each pipe size and thickness. The second method proposed by H. Baumann of Masoneilan International is quasi-empirical and independent of valve style requiring, however, empirically obtained flow characteristics. The method is restricted to choked flows. Each of the methods is a significant starting point in valve noise prediction and has been demonstrated to be accurate within 15 dB under controlled conditions. The limitations and shortcomings of the two methods are next discussed.

The Fisher Method

In January, 1968, the Fisher Controls Company began a two year program of testing their own valves and regulators in order to develop an empirical formula which could be used to predict the sound pressure levels radiated. Air was used as the primary working fluid and the tests were performed in an "acoustically isolated chamber." The sound pressure levels were measured at 48" downstream from the valve and 29" from the pipe surface for many different valve styles, sizes, pressure ratios and piping configurations. The 250,000 data points were correlated to produce the following formula:

$$SPL = SPL_{\Delta p} + \Delta SPL_{C_g} + \Delta SPL_{\frac{\Delta P}{P_1}} + \Delta SPL_K \qquad (5)$$

where SPL is the overall sound pressure level in dB (re 0.0002 dynes/cm^2), four feet downstream from the valve and 29 inches from the pipe surface. $SPL_{\Delta p}$ is the base SPL in dB determined as a function of ΔP given on a straight line semi-logarithmic plot. SPL_{C_g} is the correction for the variation in critical flow coefficient C_g as a function of ΔSPL_g on a semi-logarithmic plot. $SPL_{\Delta P/P_1}$ is the correction for valve style and pressure drop ratio for specific size valves usually a 2 slope straight line plot of $\Delta P/P_1$ versus $\Delta SPL_{\Delta P/P_1}$. SPL_K is the correction for pipe wall attenuation given in the form of table of pipe size and schedule number. The formula, equation (5), gives the "overall sound pressure level" meaning dB (linear). This result contains no spectral information and, therefore, cannot be used to compute the dB(A) value which is needed to establish compliance with the OSHA requirements. The conversion to dB(A) can only be made if such spectral information as octave or 1/3 octave data is available. The spectral content of noise for different sizes, styles and installations can be expected to vary significantly, thus raising questions on the utility of the method.

Furthermore, the method gives no information on the directivity of the radiated sound. Because of the complex nature of the sources, and the occasional installation in relatively free field environments, directivity information could be used to advantage. If directional characteristics are significant, sound power calculation from single point SPL data is of questionable accuracy. It should be noted that there is a current recommendation by international and national standards organization for the labeling of products as to their noisiness in terms of sound power levels.

The correction factors used in the Fisher Method are plotted as straight lines without any indication of the statistical nature of the information. Thus, the effect of data variability in arriving at the final SPL may well be a significant factor in proving compliance with noise laws.

The tests were performed in a "soft room" [8] and air was supplied through an upstream in-line silencer. The downstream section, however, was not anechoically terminated so that standing waves peculiar to the test set up might bias the data under certain conditions [9]. There is also some concern on the use of "Soft Rooms" as opposed to good anechoic practice in view of the distinct pure tones that are generated by control valves which may set up room modes that will result in significant errors in SPL readings at the microphone positions.

As with all empirically derived prediction methods, the Fisher formula and the required correction factor, is valid for Fisher products. Application to other manufacturers products would require extensive testing under carefully controlled, standarized conditions to assure that comparable results are obtained. Such tests are very expensive and time consuming. The Fisher method can, therefore, not be considered a "universal" prediction scheme. The reasons given by Fisher for choosing a completely empirical approach are that: (a) The application of the Lighthill theory to confined jets is questionable, and (b) little is known concerning the turbulence level of a bounded stream [7]. Considerable knowledge is currently available on the turbulence levels of bounded flows.[10] With some added and much needed research, this knowledge could be applied to the development of a prediction scheme, which is based on stronger fundamental considerations.

Masoneilan Method

This approach as proposed by H. Baumann [11] is an analytical approach based on the conversion of mechanical power in a valve to acoustic power. The method is primarily limited to sonic or choked flow conditions at the throat of the valve. The formula contains terms that can be obtained from common performance tests and the expensive acoustic tests required for the Fisher Method are not necessarily required. The Masoneilan formula is given in equation (6).

$$SPL = 10 \log_{10} (\eta \times 10^{11} \times C_v C_f P_1 P_2) - TL - S_g \qquad (6)$$

where SPL is the sound pressure level (re 0.0002 dynes/cm^2) referred to the "A weighted scale" measured 3 feet from the throttling control valve operating under choked flow conditions.

η is the acoustic efficiency which is given as a function of valve pressure ratios and critical flow factors C_f

C_v is the valve flow coefficient, g.p.m. / $\sqrt{psi}$

C_f is the critical flow coefficient (which implicitly contains the valve geometry)

P_1 is the static upstream pressure, psia

P_2 is the static downstream pressure, psia

TL is the transmission loss through the pipe, dB

S_g is the gas property correction factor, dB

The formula is based on the assumption that noise generated by turbulence in a free field is applicable to the case of confined jet mixing. The reservations discussed previously are here referred to: namely the lack of free field conditions, the pseudo-sound or acoustic near field conditions, and the flow separation and unstable boundary layers in the pipe downstream of the valve. For these reasons, it is doubtful that the application of classical similarity arguments in estimating the Lighthill formulation would be successful. The use of the acoustic efficiency term as a starting point in the development of a prediction equation is, therefore, open to valid criticism. An additional area of concern is the formation of shock cells downstream of the choked cross section for over-expanded conditions. These cells involve sharp pressure discontinuities which in turn amplify the turbulence generated pressure discontinuities. [12]

The second basic assumption in the Masoneilan Method is that sound power and sound pressure are simply related following the criteria for spherical radiation. It is well known that a free jet does not radiate spherically as has been discussed before and there is no reason to believe that a confined jet and associated piping will radiate strictly spherically. This area of concern has been referred to in reference [11] by Baumann, but requires further verification. Two other areas of concern are the use of a transmission loss formula based on flat panel transmission loss (T.L.) theory and the use of a single frequency which is based on the peak value of the free jet spectral distribution to calculate the T.L. Flat panel transmission theory is not applicable due to the curvature and stiffening effect of the pipe as recently reported in reference [13]. But even for flat plates the transmission loss must be computed for three distinct frequency regions; the mass law region at low frequencies, the critical frequency itself, and the region above critical frequency. The use of a single frequency implies that all the sound energy which is being transmitted is contained in that Strouhal frequency. Such is not the case as the statistical nature of the spectral distribution and the response of the pipe system to acoustic and hydrodynamic energy over the spectrum must be considered. The statement that the result will be a good estimate of the dB(A) sound pressure level is thus open to question.

The experimental verification of the Masoneilan formula is subject to the same comments that were presented in the discussion of the experimental method employed by the Fisher Control Company.

NOISE REDUCTION METHODS

It has already been established that control valves and regulators are significant contributors to industrial noise. Therefore, the manufacturers recognizing this serious problem have applied much ingenuity and insight into the development of quieter valves.

Two basic approaches can be used: 1. The noise can be reduced at the source, the valve itself or; 2. The noise produced by the valve is absorbed in its path away from the source by the application of mufflers or acoustic pipe wall treatment.

The source treatments now proposed are: (a) The multiple series expansion through several carefully designed throttling devices downstream of the valve as proposed by Hynes [14] of Masoneilan which gives reduction in excess of 10 dB. (b) The use of multiple parallel expansion through carefully designed nozzles as proposed by Fisher Controls Company under the trade name of "Whisper Trim." This latter approach is similar in concept to the multiple tube silencers of current jet engines and gives reductions in actual jet engines of approximately 20 dB. (c) The "torturous path disc stack" approach embodied in the Control Components Incorporated (CCI) "Drag Valve" consists of a combination of the parallel and series approaches in that the radial flow through the many labyrinths is controlled by the axial position of the plug of the valve. This approach to reducing valve noise has shown great promise, giving reductions of the order of 40 dB compared to similar sized conventional valves. Because of the added complexity, these valves are considerably more expensive. Because of the many and small passages the "Drag Valve" is more sensitive to plugging by solid particles in the fluid.

The path treatment can consist of the well known acoustic treatment of pipe walls or the construction of acoustically treated in line silencers and enclosures. Such silencers can provide in excess of 30 dB of noise reduction at the peak frequency. The mass effect of heavy piping can also be used effectively. It is the authors' contention, however, that a better understanding of valve noise sources and transmission could provide substantial further reductions at lower costs to the user. Further research in these areas is needed.

CONCLUSIONS

Control valves and regulators are significant contributors to the industrial noise problem. The valve noise prediction method proposed by the Masoneilan Company is based on a much oversimplified application of the turbulence caused aerodynamic noise theories of Lighthill, Curle and others. The analytical model does not effectively treat the fact that near field conditions exist and that spherical radiation cannot be assumed because of directivity considerations and the effect of the piping on the radiation. The use of a single Strouhal frequency to calculate transmission loss and then relate this value to a suggested dBA sound pressure level is particularly open to question.

The empirical approach of the Fisher Controls Company requires extensive and time consuming test procedures and is suited to particularly Fisher valves.

In both cases, there is some concern about the test methods used to arrive at the noise output. Specifically the valve outlet termination can have a significant effect on the source characteristics because of the potential of standing waves in the system. Also the effect of incoming turbulence needs to be explored.

In spite of these reservations the two methods appear to be the only ones available, and they do provide usable estimates. Another method of value is proposed by the Leslie Company, which uses pipe wall acceleration measurements to estimate [14] octave band SPL. This method appears to be potentially promising for those applications where anechoic acoustic facilities are not available.

The state of knowledge of sound generated by choked and over choked free jets still requires the development of a quantitative theory.

Results obtained with such free, unconfined, jets show that the acoustic power generated can vary as U^{12} to U^{15} for cold jets and U^{17} to U^{19} for hot jets. This result is of course a major departure from the oft quoted results of Lighthill, which are valid only for subsonic jets in a free field [15].

The effects of confinement in a reverberant medium and in the presence of flow separation and wall boundary layer also need further investigation as do questions related to directionality of the radiation and pipe transmission loss.

In summary, we conclude that neither prediction method is soundly based on fundamental noise source and transmission considerations. However, with the present incomplete state of analytical understanding of valve noise generation, transmission and radiation, a complete fomulation of a prediction equation on theoretical grounds alone is out of the question. We believe, however, from data available at present, a more soundly based formulation can be prepared similar in format to the Fisher Method which in addition permits calculation of octave band sound pressure levels. An improved method should, of course, avoid some of the objections raised in the discussions in this paper.

Several approaches to noise reduction are available or under development. Reduction of noise at the source, the valve, consist of either multiple expansions through either series or parallel or both types of paths. Reductions of noise by acoustic treatment of the path have also shown success and consist of mufflers to reduce air-borne sound transmission and/or flexible couplings to reduce solid-borne sound transmission. Acoustically absorbent pipe wall treatment and enclosures are well known approaches in the field.

The fields of valve noise prediction and reduction are, generally speaking, in their infancy, thus further applied research should provide significant results.

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STATIONARY AND PORTABLE AIR COMPRESSORS

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Introduction

Compressors have been identified as major noise sources. Large numbers of compressors, of all types, are used in refineries, chemical plants, generating stations, and other major industries. Certain types of compressors generate relatively high noise levels -- above those permitted by the Occupational Safety and Health Act -- and therefore need attention. Portable compressors produce some of the most objectionable noise on city construction projects, and for this reason, most city noise control codes set maximum permissible levels for compressors. Pending Federal legislation includes compressors in the list of products for which noise emission standards will be established. It is obvious that compressor sound control is needed, and this requires an understanding of the noise generating process. Various techniques have been found to be effective in reducing the noise of centrifugal, axial, axi, and reciprocating compressors.

Compressor Noise Sources

The noise radiated from a compressor is complex, and consists of components from many sources. In order to reduce the total noise, the various contributions must be identified and evaluated, and the largest ones worked on first. The ideal approach is to prevent the generation of noise by design, but this is not always the most economical solution. Noise reduction after a machine has been built is often the most practical procedure.

Turbulence

Consider turbulence for example. Turbulence is the most important source of noise in centrifugal compressors. It is really a combination of two effects, (a) Vortex shedding, and (b) up-stream turbulence. The boundary layer over each blade is turbulent by the time it reaches the trailing edge. The turbulent layers on the top and bottom surfaces produce a fluctuation in lift, and this fluctuation has a broad frequency spectrum. The application of a fluctuating force to a gas generates sound at the same frequency. Therefore, broad band noise is radiated. If the flow is turbulent when it enters a blade row, the turbulence is increased and the noise is greater.

Turbulence noise is radiated through the compressor casing, and it can be controlled by an acoustic enclosure -- after the compressor has been installed. It is almost impossible to eliminate turbulence by design.

Piping noise, produced by the same source can be reduced effectively by lagging the pipes with 2 to 3 inches of Fiberglass, Ultracoustic, Rockwool, or similar material, with a density of about 4 pounds per cubic foot, and covering this with a jacket weighing about 1 pound per square foot. The jacket can be #24-gage steel, or the equivalent weight of aluminum, lead or leaded-vinyl.

To be most effective, enclosures and pipe lagging must be tight. Leaks in an enclosure greatly reduce its effectiveness. It is unfortunate that a small leak in a high quality enclosure is more damaging than the same leak in a poor enclosure. For example, an opening with an area of 1 percent of a wall whose transmission loss is 50 dB reduces the overall transmission loss to only 20 dB. A 1 percent leak in a 25dB wall results in a final transmission loss of 19dB. That is, the effectiveness of the 50dB enclosure is reduced by 30dB while the effectiveness of the 25dB enclosure is reduced by 6dB. This shows that if you plan to have leaks in the enclosure there is no point in paying for a high quality one.

The installation of inlet and discharge silencers is another example of effective noise reduction after a compressor has been built. Silencers reduce the noise entering the inlet and discharge pipes and makes piping noise reduction easier. In some instances, pipe lagging is not necessary, depending, of course, on the final noise level required. When silencers are used they should be as close as possible to the compressor inlet and discharge flanges. It should be noted that it is almost impossible to make a significant reduction in the noise from centrifugal compressor installations unless the piping is treated.

Interaction of Rotating and Stationary Vanes

On the other hand, centrifugal compressors have other major noise sources which can be reduced by design. An example of this is the noise produced by interaction of rotating impeller blades with stationary vanes.

Every time a blade passes a given point, the air or fluid at that point receives an impulse. Therefore, that point will receive impulses at a frequency equal to the number of impeller blades times revolutions per second. In axial flow compressors the magnitude of this blade-passing frequency component is one of the largest in its generated sound spectrum. It is present also in centrifugal compressors with diffusers, but in most cases it is not as important as the blade-rate frequency. This is calculated as follows:

$$f = \frac{N_r \times N_s}{K} \times R.P.S.$$

where f = Frequency in hertz
N_r = Number of rotating (impeller) blades
N_s = Number of stationary (diffuser) vanes
K = Highest common factor of N_r and N_s

For example, let $N_r = 6$, $N_s = 8$, and the speed equal to 6000 RPM. Then

$$f = \frac{6 \times 8}{2} \times 100 = 2400 \text{ Hz}.$$

That is, there are 24 times in each revolution when impeller blades line up with diffuser vanes, and each time this happens, 2 impeller blades match 2 diffuser vanes. Therefore, the frequency will be 24 times RPS and the pulses will be of double strength.

When there are 6 impeller blades and 9 diffuser vanes, there are 18 times when impeller blades are in line with diffuser vanes. Each pulse is 3 times as strong as it would be with a single coincidence, because 3 rotating blades match 3 stationary vanes. The frequency is 18 times RPS.

It is obvious that combinations like 12 and 12 are not recommended because of the many points of coincidences, and the strength of the pulses.

It can be seen that it is better to use unequal numbers of rotating and stationary vanes. Prime numbers are the best of all because they have no common factor. This produces high frequency pulses, which are easier to control than low frequency ones. Furthermore, when prime numbers are used, each pulse is only of single strength.

Impeller-Diffuser Distance

Increasing the radial distance between impeller blades and diffusers reduces noise. It is particularly effective in reducing the blade-passing frequency and blade-rate components. Unfortunately this procedure also decreases performance, but for close initial spacing, the decrease in noise is greater than the decrease in performance. That is, noise increases rapidly as the spacing becomes smaller and smaller.

Effect of Horsepower

Centrifugal compressor noise is affected by many operating parameters. There is a direct relation between horsepower and noise, but the relation is not the same for all types. On one particular class, the overall noise can be predicted quite accurately by

$$\text{Increase in dB} = 17 \log \text{H.P. Ratio}$$

In most instances, doubling the horsepower results in an increase of about 4 to 5 dB in the overall noise.

Effect of Speed

Rotational speed has a definite effect on noise. For any particular design, the sound level will increase anywhere from 20 to 50 times the logarithm of the speed ratio. At lower speeds, centrifugal compressor noise will increase about 20 log RPM ratio. At high speeds, the increase will more nearly equal 50 log RPM ratio. The increase in sound with speed applies to the overall noise and to the component of highest level -- usually blade-passing frequency or blade-rate frequency. The increase at other frequencies is not as great, and may be of the order of 10 to 15 times the log of the speed ratio.

These same relations apply to impeller tip speeds, but in general less noise will be produced with large diameter, slow speed units, than with small diameter, high speed machines, even though the impeller tip speeds are the same in both cases. There are several reasons for this:

(a) Not all the turbulence is produced by the impeller. Even though the tip speeds are the same, the slow speed machine will have larger areas in internal passages, with lower velocities and less restriction.

Extra care in producing fine interior finish in centrifugal compressor casings to reduce noise is not justified. There is no detectable difference when passages are hand finished.

(b) Mechanical forces due to unbalance are proportional to the square of the speed. Low speed will produce less structural vibration.

Number of Stages

The noise generated by centrifugal compressors can be reduced by decreasing the work per stage -- that is, by increasing the number of stages.

Effect of Gas Molecular Weight

The molecular weight of the gas in a compressor system has a pronounced effect on the generated noise. Very little test data is available on this and it is difficult to predict mathematically. It is certain though that more noise is produced with high molecular weight than with low molecular weight gas.

Head-Capacity Operating Point

Mass flow and discharge pressure both have a profound effect on the noise produced by a compressor. As the mass flow is reduced the noise decreases until a point near surge is reached. Beyond this point the noise increases rapidly.

Inlet and Discharge

The inlet to a centrifugal compressor plays an extremely important part in the generation of noise and pressure pulsations.

It is sometimes thought that if the inlet and discharge areas are increased, the gas velocities will be reduced and, therefore, less noise will be generated. On the contrary, if theinlet opening is increased, it may actually create more noise rather than reduce it. In order to obtain proper entrance, the gas should be as near as possible to the impeller center line. This, of course, indicates a small inlet. When the gas enters near the center line on its way to the impeller vane there will be relatively low shock and turbulence. If the suction opening is enlarged, the gas is admitted farther up on the impeller vane where the linear speed is higher. The sudden change from low velocity in the large inlet to high velocity part way up on the vane causes shock, turbulence and increased noise.

Reducing the flow velocity in discharge piping is an effective way to reduce piping noise. This should not be accomplished by enlarging the opening at the cutwater, but by using a properly designed pipe increaser between the correct discharge opening and the piping.

Axi Compressors

Axi compressors are helical rotor, positive displacement compressors. Gas enters the intake ports and flows into a pocket formed between the rotors and the wall of the casing.

The pocket rotates away from the intake ports, and as the lobes and grooves roll into each other the pockets shorten and compress the gas. It moves axially and is carried toward the discharge end.

The major noise source in compressors of this type is the discharge. Next in importance is the inlet, and then the noise radiated from the compressor casing. The sound pressure level measured in the discharge of the larger, higher horsepower units may be in the range of 140 to 145 dB. Without silencers or an enclosure, the sound level 3 feet from the compressor may be about 132 dBA, and it consists of a fundamental frequency equal to the number of lobes times revolutions per second, plus higher order harmonics of this.

The amount of necessary sound control depends on the final acoustic design criterion. For maximum noise reduction, both discharge and inlet silencers should be installed, as close to the compressor as possible. Inlet and discharge piping and inlet and discharge silencers should be lagged, and an acoustic enclosure should be installed on the compressor itself to reduce casing radiation.

Reciprocating Compressors

In general, the sound level of reciprocating compressors is not very high. It usually consists of multiples of the piston movement, and is generated by both aerodynamic and mechanical forces.

Inlet and discharge noises are major components, but the pulsating flow can be reduced effectively by a chamber-type silencer or snubber.

Inertia forces are major exciting factors causing vibration and noise in reciprocating compressors. These forces are due to the motion of the pistons and related parts, and the imbalance of the connecting rod and crank mechanism. The forces produced by unbalanced masses also appear in the rotating parts of the machine, as both static and dynamic unbalance.

Impacts in the crank-connecting rod system, and the knocking of pistons against cylinder-liners during crossover, are major sources of noise in reciprocating compressors. During each revolution of the crankshaft, the piston shifts from one side to the other, several times, moving in the plane of the connecting rod motion. The gap between the piston and the cylinder liner permits the piston to move with a certain velocity in the transverse direction, impacting against the wall of the cylinder. These knocks produce an intense vibration of the cylinder walls at their resonant frequency.

This discussion is confined to some of the major noise sources in compressor systems, and does not include those attributed to motors, turbines, gears, or valves. It does not include foundation noise either, even though improper mounting on an inadequate foundation is often the cause of very high vibration and noise.

A REVIEW OF NOISE AND VIBRATION CONTROL FOR IMPACT MACHINES

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The need for noise and vibration control of impact machines has been evidenced for many years; some owners of these machines have been concerned about worker comfort/annoyance and community complaints. More recently, the reports of the CHABA working group and the American Council of Government Hygienists and the legal requirements of the Walsh-Healey Act and the Occupational Safety and Health Act have increased the awareness of owners/operators and manufacturers of these machines to where all are concerned to some extent about noise and vibration control.

Typical impact machines include punch presses, drop hammers, tumblers, some conveyers, chipping hammers, jack hammers. Since the punch press is representative of these machines; it will be considered in some detail in this paper. Noise control treatments that have been used to control the noise of impact devices include absorption within the space, vibration isolation of the machine from the floor and parts of the machine from the machine structure, damping material applied to the machine structure and other parts of the machine, enclosures (both partial and total) of the machine, absorptive barriers between the machine and the operator and mufflers. Other noise control techniques for punch presses include having the punch operate in shear as well as various other modifications of the tooling. In addition to presenting a review of these techniques, this paper presents a survey of the manufacturers of punch presses to determine their success at solving the noise problems.

IMPACT MACHINE NOISE LEVELS

The range of noise levels in dBA at the operator's position with these particular machines operating is:

Impact Machine	Noise Level in dBA
Punch presses	88-112
Drop hammers	90-110+
Tumblers	92-115
Chipping Hammers	98-110

+peak levels of 130 to 150 dB for each impact have been measured

Since the employee is often continuously exposed to the noise generated by these machines, the OSHA 90 dBA noise level limit applies; thus, the objective of the industry, both owners/operators and manufacturers, should be to reduce the noise levels generated by these machines (while operating under load) to less than 90 dBA. In some situations, this could require 20 to 25 dBA noise reduction.

NOISE CONTROL TREATMENTS

Absorption

When the impact machine and the individual to be protected are sufficiently separated within a plant space, it is possible to achieve significant noise reduction through the application of absorptive materials to the space. Figure 1 presents octave-band noise measurements measured before and after a sound-absorptive treatment was installed in a room with several large punch presses in operation. The measurement positions are about 30' from the punch presses. The sound-absorptive treatment consisted of hanging 2'x4' absorptive baffles on 3' center spacings over the punch press area and over the area where the measurements were made. In addition, 800 square feet of the inside wall (behind the punch presses) were covered with glass fibre board. Noise reduction on the order of 9 dBA was accomplished for this position some 30' from the noise sources [1]. At closer positions less noise reduction is accomplished with the use of absorptive treatments. In a typical punch press operation, where the presses are spaced perhaps 10 to 15 feet apart, an absorptive ceiling treatment can significantly reduce the contribution of the surrounding presses to the total noise level at a particular press. The maximum noise reduction of the total noise level that can be achieved at a close-in operator's position through the use of an absorptive ceiling treatment is on the order of 2 to 3 dBA. Although noise reductions of this magnitude do not often solve noise problems, it should be noted that the decrease in reverberation time with the absorptive treatment will enable an operator to more easily identify the processes associated with his particular machine and thereby may enable him to better control its operation.

Enclosures

Partial and total enclosures for impact machines have been designed, most often in the retrofit stage. For punch presses, the partial enclosures usually consist of add-on panels, containing operable windows, that close-off the opening into the press -- including the front and/or back openings for tooling adjustments and openings for scrap, material entrance and product exit. Depending upon the owner-operator's production requirements some of the openings may not receive full treatment (with lined duct, etc.). Figure 2 presents a comparison of the close-in noise measurements before and after a partial enclosure consisting of a metal framework and plexiglass windows was applied to a 22 ton punch press operating at approximately 300 strokes per minute. Approximately 10 dBA noise reduction was accomplished using the partial enclosure concept.[1]

A total enclosure, erected close to the punch press with appropriate opening for maintenance of the press can be used to reduce the noise. Such an enclosure was designed by personnel at IBM for a multi-slide punch press.[2] The enclosure consisted of steel panels mounted on a steel frame. Resilient seals were provided at the stock entrance to the enclosure and for access panels. Ventilation for the enclosure was provided using a lined duct and small blower. Figure 3 presents the noise reduction accomplished using this enclosure -- about 20 dBA. It is often easier to achieve greater noise reduction using a close-fitted total enclosure with access panels than with a partial enclosure.

When there is sufficient floor space available, a larger enclosure around the machine with sufficient room for maintenance within the enclosure has certain advantages. With such an enclosure, it is often possible to significantly reduce the size of openings for stock, product and scrap. In addition, such an enclosure has the advantage that the walls can be constructed of heavier materials. On the order of 30 dBA noise reduction can be accomplished using this approach.

Other Noise Control Measures

Other forms of noise control treatments that have been utilized include vibration isolation, damping material, absorptive barriers and mufflers. Depending upon the specific details of the construction of the machine, it is sometimes possible to achieve 5 to 15 dB noise reduction through these techniques, for example: by providing a damping material as a liner for tumbling barrels or conveyors, vibration isolation for resonant sheet metal panels, absorptive barriers greater than head high (with an absorptive ceiling treatment), mufflers for the air exhaust on punch presses, jack hammers, chipping hammers, etc.

Recent investigations by personnel at General Electric have confirmed that the impact noise of a punch press can be significantly reduced (about 15 dB) by operation of the punch in shear, thereby reducing the forces exciting the structure [3]. It should be noted that hydraulic presses usually operate in shear and are generally considered to be quieter by the industry. Another way to reduce the noise of the impact may be to conserve energy in the manner utilized in a transfer press where a coining (forming) operation follows a blanking operation.

Low speed punch presses often utilize an air knockout -- high pressure air impinging on the edge of the punched part to remove it from the die. Other types of knockouts that have been suggested (but not evaluated) to reduce the noise include:

1) magnetic ejector [4]
2) air knock-out through base of die [5]
3) utilization of more jets at significantly lower pressure drops across each nozzle
4) mechanical knockouts

RESPONSIBILITY FOR NOISE CONTROL

The basic question of responsibility for noise control of impact machines continues to be debated. This is especially true in the metal stamping industry where the responsibility seems to be divided between the owner and the manufacturer since the punch press manufacturer very seldom supplies the tooling (die) for press operation and since the impact is the major noise source.

In a recent survey of ten punch press manufacturers, several questions were asked:

1) Is your firm required to comply with any noise level specifications when bidding on jobs?
2) Does your firm guarantee noise levels of your machines with tooling?
3) Do you have any add-on noise control treatments (for example, a partial enclosure or a total enclosure)?
4) Are you investigating noise control by way of press redesign?
5) Are you investigating noise control by way of die design?

Table 1 presents the results of this survey.

Several manufacturers have accepted the responsibility to provide noise control treatments that meet the specifications (varying between 80 and 90 dBA) when the machine is operating with load. The added cost for noise control is on the order of 3 to 10% of the base price. One of these manufacturers has indicated that they were also able to reduce the noise level by 8 dBA by altering the design of the stripper plate from a moving action to a stationary action; this involves redesign of the tooling. Clearly the manufacturers who supply tooling are in a better position to investigate the total problem. If the noise of impact devices under load is to be controlled to within acceptable limits, it will be essential for owners/operators and manufacturers to share in the responsibility jointly and in a coordinated manner.

CONCLUSION

In summary, we can say that

1) absorptive treatments can be effective in certain applications
2) partial enclosures of the machines can yield as much as 10 dBA noise reduction
3) total close-fitting enclosure can achieve about 20 dBA
4) well-designed total enclosures with lots of space can achieve as much as 30 dBA

5) redesigned stripper plate actions can reduce noise levels by about 8 dBA
6) operating the punch in shear can reduce the noise levels by about 15 dBA.

Although there are good practical solutions to the noise problems of some impact machines, there is still much work to be done before a practical solution for the impact itself is achieved.

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[3] Shinaishin, O.A., Punch Press Noise -- Outline for Analysis and Reduction, General Electric Technical Information Series Report No. 72CRD103

[4] Voorhees, J., personal communication

[5] Speicher, W., personal communication

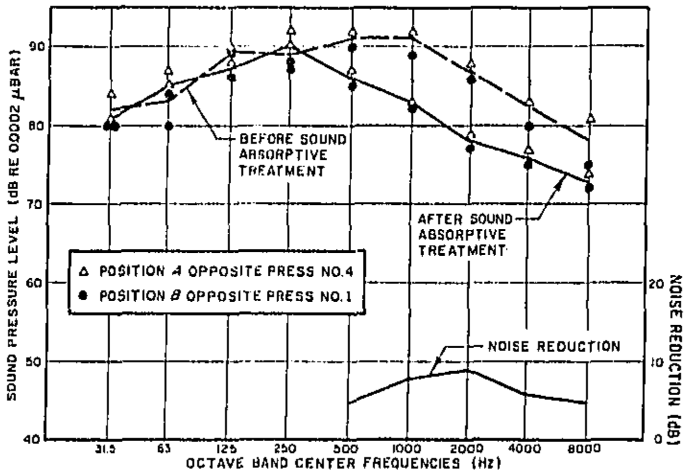

FIG. 1 OCTAVE BAND NOISE LEVELS AT BENCH PRESS AREA BEFORE AND AFTER SOUND ABSORPTIVE TREATMENT

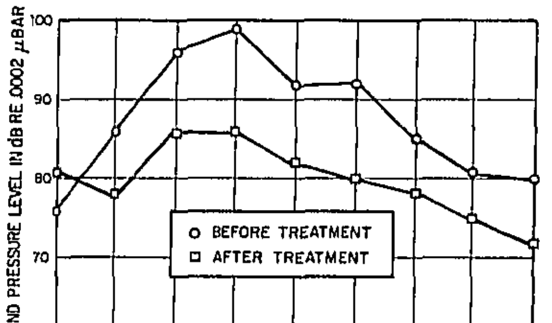

FIG. 2 COMPARISON OF CLOSE-IN NOISE MEASUREMENTS BEFORE AND AFTER PARTIAL NOISE CONTROL TREATMENT ENCLOSURE

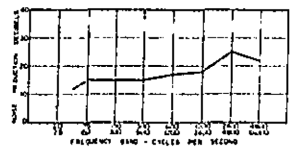

FIG. 3 NOISE REDUCTION PROVIDED BY ACOUSTIC COVER FOR MULTI-SLIDE PUNCH PRESS AT I.B.M.

TABLE 1 - SUMMARY OF SURVEY OF PUNCH PRESS MANUFACTURERS

Manufacturer	Customer specifies noise levels*	Guarantee noise levels with tooling	Supply tooling	Supply noise control	cost	R&D via press design	R&D via die design	comments
A	Yes	No	No	Yes	5-10%	No	Yes	Have concluded that total enclosure is best solution. Get about 20 dBA with enclosure. Got about 8 dBA experimenting with die design.
B	Yes	No	No	No	-	No	No	Have allocated funds to consider noise problem, will probably use it for "press design"
C	Yes	No	No	No	-	Yes	Yes	Considering noise at no load condition, also considering use of heavier plates and I beams rather than box beams, also use of pneumatic die cushion.
D	Yes	No	No	No	-	Yes	No	Doing complete dynamic analysis of a straight-sided press and developing computer model to use in analysis of noise.
E	No	No	No	No	-	No	No	No research for noise
F	No	No	No	No	-	No	No	Some concern about noise at no load condition, have used gears immersed in oil to reduce gear noise.
G	Yes	Yes	Yes	Yes	10%	No	No	Will supply enclosure, will guarantee noise levels with tooling
H	Yes	No	No	No	-	Yes	No	Considering use of castings instead of welded structures, also fiberglass guardings
I	No	No	No	No	-	No	No	-----
J	Yes	No	No	No	-	Yes	No	Motor, blower and gear noise of concern to them
K	Yes	No	No	No	-	Yes	No	Investigating dynamic balance
L	Yes	No	No	Yes	3-6%	Yes	No	Noise control keyed to reducing noise at no load
M	Yes	Yes	Yes	Yes	-	Yes	Yes	Noise control directed toward quieting power units and operating in shear

*the noise levels specified vary from 80 to 90 dBA with the press operating at no load.

+$3,500-$8,000

NOISE CONTROL FOR INDUSTRIAL AIR MOVING DEVICES

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Some of the common sources of noise in air handling systems are fans for air conditioning systems, cooling towers and forced draft installations as well as gas turbines, electric generators, electric motors, jet engines, internal combustion engines, and air compressors. Each has two air handling openings and a case. Each of the three is a source of noise. These devices are often found in pairs. While the jet engines, gas turbine, electric motor and generator, and internal combustion engine use air while producing power, the fans and air compressors use power to deliver air. Noise is typically created by both the power source and power user device in air handling systems.

The similarity between the systems allows us to consider noise reduction required for one and transfer the technique to any of the other systems. The gas turbine driven generator set will be used as the example system.

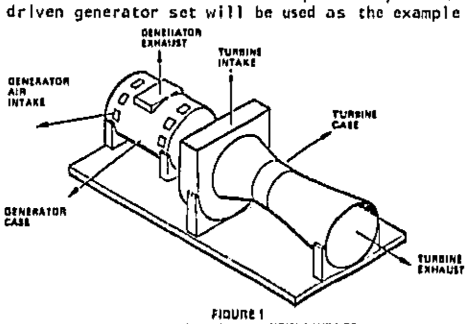
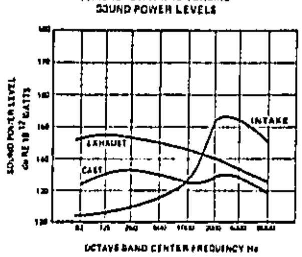


FIGURE 1
GAS TURBINE GENERATOR NOISE SOURCES

FIGURE 2
TYPICAL 40 MW GAS TURBINE
SOUND POWER LEVELS



OCTAVE BAND CENTER FREQUENCY Hz

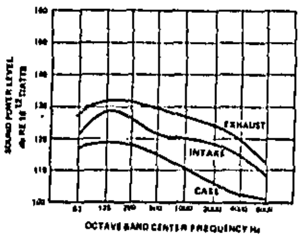
The six primary noise sources in the system are shown in Figure 1. They include the inlet and exhaust openings of the gas turbine as well as the inlet and exhaust openings for generator cooling air. The case of each machine is also a noise source. The turbine intake produces a high pitched whine with most of its sound energy in the frequency range between 2,000 and 8,000 Hz. Levels for a typical 40 megawatt gas turbine drive unit, along with typical exhaust and case noise spectra, are shown in Figure 2. It will be noted that the turbine intake produces very little noise at lower frequencies. The turbine exhaust is almost as loud as the intake, however, the noise produced is in the low frequency ranges and differs substantially in character.

While the low frequency exhaust rumble may rattle windows, it is normally not nearly as disturbing to the ear as the high pitched whine of the intake. This is because the ear is significantly more sensitive in the frequency range where the intake noise is loudest than it is at low frequencies where the exhaust noise is loudest.

The turbine case radiates a combination of intake and exhaust noise, however, the level is down because the sound must first travel through the walls of the case before radiating to the observer.

The generator produces noise at the fundamental and harmonic frequencies of the 60 cycle line frequency by the vibration of the generator core laminations. The machine also produces noise by it's cooling air fans which circulate air through the machine to keep it cool during operation. Both types of noise radiate from the generator case, the cooling air intake and exhaust openings. Typical sound power spectra and levels for a 40 megawatt generator are shown in Figure 3. It will be noted that the highest noise levels are in the 125 and 250 Hz band in each case.

FIGURE 3
TYPICAL 40 MW GENERATOR
SOUND POWER LEVELS



Once the sound power output of each machine is known, it is possible to design a system to reduce the sound power generated by each noise source to sound pressure levels which meet the specified criteria at the distance under consideration.

The first requirement is to determine the unsilenced sound pressure level (SPL) at the point of concern. This can be done by using the SPL/distance relationship of Figure 4 to convert the sound power levels of Figures 2 and 3 to sound pressure levels at the distance of interest. This chart simply accounts for the reduction in sound intensity as the energy is spread over a sphere of ever increasing radius and surface area. It also accounts for the reduction in sound level due to acoustic adsorption in the atmosphere at great distances.

FIGURE 4. RELATIVE SPL VS. DISTANCE FROM SOURCE BY OCTAVE BANDS

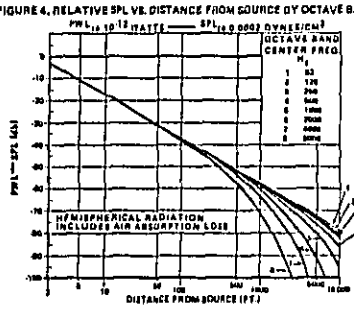
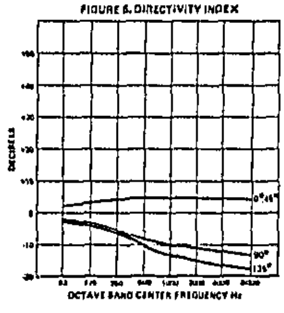


FIGURE 5. DIRECTIVITY INDEX



In addition to using Figure 4 to account for the distance from the source to the observer, Figure 5 must be used to correct for the sound propagation direction of the source compared with the direction to the observer. Air intake and exhaust stacks have directional characteristics much like a horn. In each case, the noise is loudest in front of the opening and at a minimum 180 degrees from the axis of the opening.

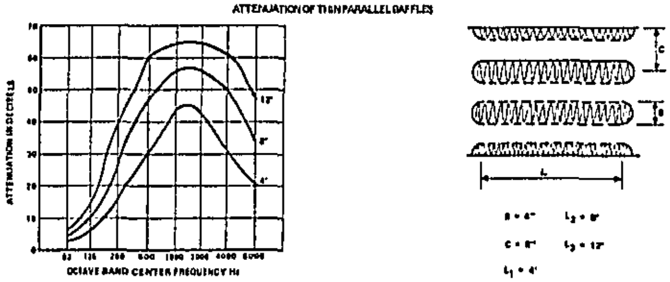
The SPL, at a given distance, is then calculated by subtracting the distance and directivity corrections of Figure 4 and Figure 5, respectively, from the sound power levels determined from Figures 2 and 3. We now arrive at the noise reduction required for each source by subtracting the criteria under consideration from the previously derived SPL.

It should be kept in mind that the total sound produced by the whole system must meet the criteria, therefore, some sources will have to be reduced below the criteria level in order to take cognizance of the fact that the sound powers add together. It is satisfactory for the first rough cut at system design to neglect this fact since the normal attenuation characteristics of silencers rarely match the exact requirements.

Once the silenced levels are determined for each source using the most appropriate silencer design, it will be found that in many octave bands each source is reduced below the required level in spite of all that is done in silencer design to match the device to the requirement. It will normally be found that the turbine intake requires almost exclusively high frequency noise reduction while the turbine exhaust requires almost exclusively low frequency treatment. Turbine case will normally require silencing which is effective at both high and low frequencies. Generator noise is normally not as loud as turbine noise with each of the three noise components requiring somewhat more low frequency noise reduction than high.

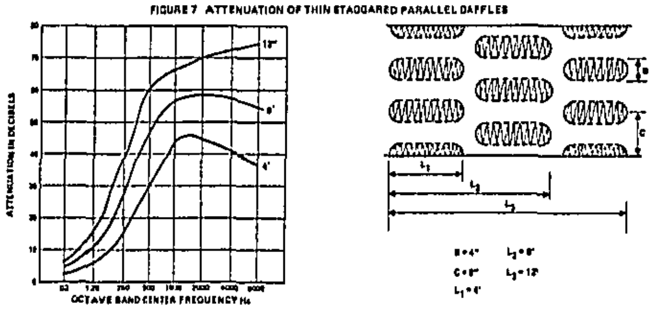
In order to match the attenuation requirements of each noise source, it is necessary to have a battery of noise control systems to work with. Several are presented here as examples. An example of thin parallel baffles having fifty percent free area through the silencer is shown in Figure 6.

FIGURE 6
ATTENUATION OF THIN PARALLEL BAFFLES



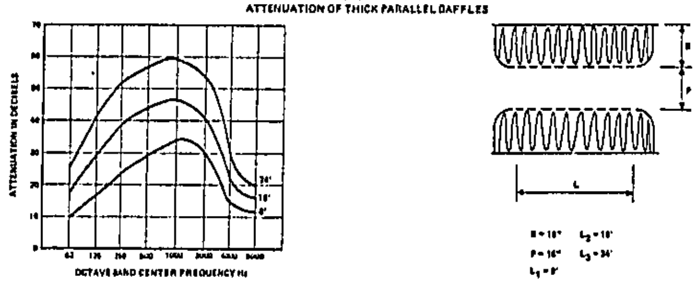
Designs like this are often used in turbine intakes. It will be noted that there is little attenuation at frequencies below 250 Hz. The figure shows performance for lengths of 4, 8, and 12 feet. The high frequency performance of this system can be further enhanced if the panels are staggered as shown in Figure 7.

FIGURE 7 ATTENUATION OF THIN STAGGARED PARALLEL BAFFLES



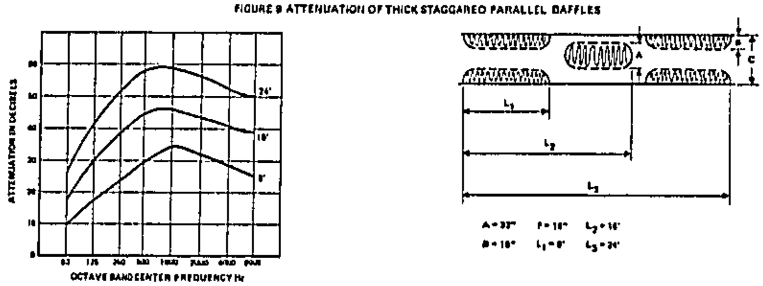
The low frequency characteristics are about the same while the high frequency performance is significantly better, particularly at 4,000 and 8,000 Hz.

FIGURE 8
ATTENUATION OF THICK PARALLEL BAFFLES



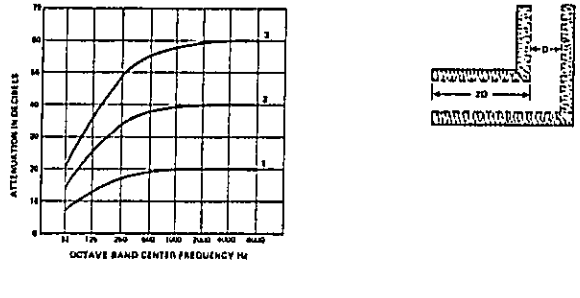
A configuration with significant low frequency performance and little high frequency performance is shown in Figure 8. In this case, the silencer has 33 percent open area and thick panels. This type of system is effective for turbine exhaust and generator noise control. In some cases, it is necessary to have a silencer with good performance at both low and high frequencies. This can be achieved by taking the configuration of Figure 8 and staggering the panels to produce the configuration shown in Figure 9.

FIGURE 9 ATTENUATION OF THICK STAGGARED PARALLEL BAFFLES



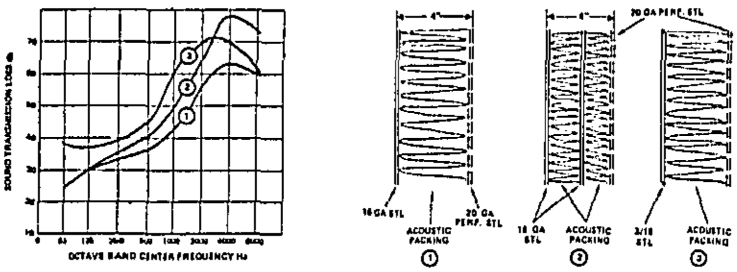
Many noise control systems involve a change in direction of the air or gas passages and, therefore, are ideally suited for the use of lined bends. The performance of 1, 2, and 3 typical bends are shown in Figure 10. Bends are very effective at high frequencies and can be made reasonably effective at low frequencies if the lining thickness is increased to give a smaller percent open area. The size of the bend also influences it's performance, therefore, detailed design information is needed for proper evaluation.

FIGURE 10
ATTENUATION OF ONE, TWO, OR THREE LINED BENDS



Typical wall constructions useful in noise control enclosures are shown in Figure 11 along with their performances. The construction to be used depends upon the noise reduction required. The noise control engineer, of course, has many more systems in his bag of tricks. These six systems represent only an example of available designs.

FIGURE 11
SOUND TRANSMISSION LOSS OF TYPICAL NOISE CONTROL PANELS



At this point, it is simply a matter of matching a proper design to the noise control requirements for each source, and adding the SPL from each source together to determine the total SPL produced by a system. If the total comes out too high, it is necessary to systematically reduce the particular sources which are the loudest in the octave bands presenting the problem, keeping in mind the cost of noise reduction in each system in order to bring the total system in line with the criteria at minimum costs.

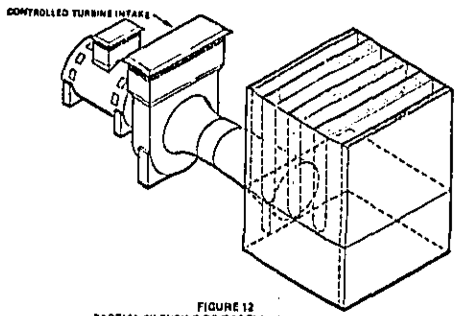


FIGURE 12
PARTIAL SILENCING OF GAS TURBINE GENERATOR

An example of what the solution might look like is shown in Figures 12 and 13. In Figure 12, an exhaust silencer has been added to the generator cooling air opening. The silencer, of course, has a hood to prevent the entry of rain into the generator. A silencer with a hood is also shown on the turbine intake. In each case, the length and configuration of the silencer would match the attenuation requirements.

The exhaust system is silenced with an elbow and thick parallel baffles similar to those shown in Figure 8. The exhaust elbow creates a problem of it's own. The walls of the exhaust plenum become a potential noise source and their transmission loss must be evaluated to determine whether or not this source is of significant concern. It may be necessary to add to the walls to increase their transmission loss to the point where this source is not a problem.

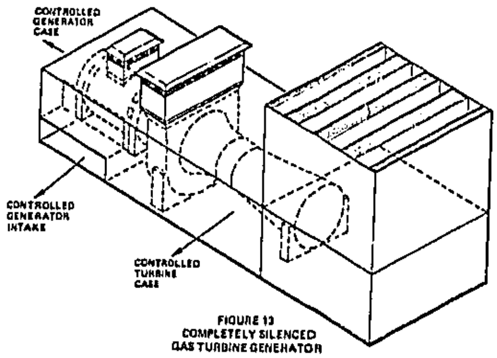


FIGURE 13
COMPLETELY SILENCED
GAS TURBINE GENERATOR

In Figure 13, an enclosure has been added around the turbine and generator. It will be noted that an opening is left in the generator enclosure to provide access for cooling air to enter. This opening, of course, will have a silencer built into it to control noise which might escape through the opening.

The great propounderance of air handling system noise control problems are similar to the one just described. It is typically necessary to determine the sources involved, the sound power output of each source, and then work to control each source by an appropriate silencer, keeping in mind that the total noise output from all sources when silenced must be below the criteria being considered.

NOISE OF FANS AND BLOWERS

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INTRODUCTION

This paper is intended to review the current state of the art of fan noise and to provide both an explanation of the data presently available to the design engineer and to provide reference sources for further information.

A review of the current state of the art of fan noise must start with an explanation of the data currently available to air handling system design engineers. The fan industry reports fan noise in terms of sound power levels re 10^{-12} watt in 8 octave bands (63, 125, 250, 500, 1000, 2000, 4000, 8000 Hz). These sound power levels are determined according to a method outlined in the Air Moving and Conditioning Association (AMCA) Bulletin 300-67 - Test Code for Sound Rating.[1] This test code calls for the determination of sound power levels using a semi-reverberant room and the substitution method using a calibrated reference sound source.

SCOPE

At the present time AMCA Bulletin 300-67 is intended to apply to the following types of fan equipment (1) Central station air conditioning and heating and ventilating units, (2) Centrifugal fans, (3) Industrial, axial and propeller fans, (4) Power roof and wall ventilators and (5) Steam and hot water unit heaters.

Reported noise data must be based on a product sample or a production model which must be of the same design and materials that will be sold. There is no recognized way to use the noise data of one type of fan to predict the noise levels of other types of fans.

The test code measures only the noise radiated from the inlet or the discharge. This is the noise radiated into an attached duct work system or into the surrounding space in the case of an open inlet or outlet fan. The test code does not attempt to evaluate the amount of noise radiated from the fan casing.

The test code does not include any information or directivity of the noise radiated from the fan.

TEST PROCEDURE

The test procedure involves a comparison between the sound power level produced by the fan and that produced by the reference sound source in the same room and under the same acoustic conditions in the room. Sound pressure level readings are taken on an octave band basis for both the fan and the reference sound source and the sound power levels of the fan are calculated from these data. In order to get a valid statistical sample of the sound pressure level in the test space, the microphone is moved through a specified path during the test procedure.

The AMCA test must be conducted in a diffuse sound field to get an accurate evaluation of the sound pressure levels.

The test room must be qualified by recording the sound pressure level of the reference sound source at two different microphone paths and comparing the readings obtained. If the difference between the two readings is less than 3 dB for a doubling of the distance from the reference sound source, the room is qualified for AMCA tests.

The method assumes that equal sound power is radiated from the discharge opening and the inlet opening of the fan. The data reported in most publications is the total sound power level of the fan and the sound radiated from either the inlet or the outlet is obtained by subtracting 3 dB from the total sound power level.

POINT OF RATING

All sound power level data apply only to the point of rating on the fan performance curve that was used for the test since there is no accepted way to calculate the sound power level at other points on the fan performance curve.

END REFLECTION CORRECTION

When noise is radiated from the open end of a tube the amount of low frequency noise radiated from this open end is a function of the diameter of the tube. The smaller the diameter, the smaller the fraction of low frequency noise radiated into the surrounding space. The low frequencies are reflected back into the tube just as if the open end of the tube were closed with a solid plate. This end reflection phenomena is important to system design engineers because low frequency noise will be reflected back in the same manner from duct and stack terminations.

Fans are usually tested with test ducts terminating in a nozzle or an orifice plate and the open inlet and/or outlet of the fan is similar to a duct termination. Both conditions form an open-ended tube and the low frequencies are not radiated efficiently from such terminations but are reflected back into the test unit.

Under the conditions of AMCA Bulletin 300-67, it is assumed that the sound energy reflected back into the fan-duct combination is completely dissipated internally and does not appear as a part of the direct sound from the fan. To account for the loss of these low frequency components a procedure is given in the code which calls for correction factors to be added to the low frequency part of the spectrum. Thus the end reflection correction is <u>added</u> to the fan noise data to indicate the total noise generated inside the fan casing by the fan wheel.

Whether or not this total noise, especially the low frequencies, is actually radiated to the surrounding space depends on the dimensions of the duct work attached to the fan in the final installation. As mentioned above, these low frequencies cannot be radiated through small openings to this surrounding space. Therefore, in the design of the system, these low frequency corrections should be <u>subtracted</u> from the sound power level of the fan to give the correct values for noise radiation from the outlets.

RATINGS PROGRAM

Obviously the fan manufacturer cannot possibly test all sizes of fans at all possible speeds so a Ratings Program has been devised to limit the number of tests and yet give good data on the noise generated by the fan. The conditions of this program are defined in AMCA Bulletin 311-67 - Certified Sound Ratings Program.[2]

<u>Size and Speed Change</u>. Under the terms of the Certified Ratings Program it is permissible to test one size of a particular fan design and use that sound power level data for other sizes of the <u>same</u> design. Also, within limits it is permissible to use test data taken at one speed to calculate the resulting sound power levels throughout a range of speeds.

The sound power levels as determined under the conditions of AMCA Bulletin 300-67, are reduced to standard levels in each octave band by using the following equation:

$$L_g = L_w - 40 \log_{10} \frac{RPM_t}{1000} - 70 \log_{10} \frac{D_t}{20} - 10 \log_{10} \text{(bandwidth, Hz)}$$

where:

L_g = standard sound power level in each octave band

L_w = sound power level of the fan calculated from tests

RPM_t = test speed, RPM

D_t = test wheel diameter, inches

(This calculation obviously reduces all fan noise data to a common denominator of 1000 RPM and 20" wheel diameter.)

The results of this calculation are plotted on an octave band basis and a smooth curve is drawn through these points. This becomes the "generalized sound power level" curve.

The generalized sound power spectrum is plotted on a chart which is a part of AMCA Bulletin 311-67 and, by use of a speed scale on this chart, the spectrum shape is changed as a function of speed. The new spectrum values are read from the chart and are corrected to the power level values at actual size and speed by:

$$L_s = L_g + 40 \log_{10} \frac{RPM_s}{1000} + 70 \log_{10} \frac{D_s}{20} + 10 \log_{10} \text{(bandwidth, Hz)}$$

where:

L_s = sound power level of fan selection

RPM_s = speed of fan selection, RPM

D_s = diameter of fan wheel selection, inches

AMCA BULLETIN 300-67 - LIMITATIONS

Probably the greatest limitation of AMCA Bulletin 300-67 as presently written is the lack of data on pure tone components. The present method of measuring the octave bands tends to conceal the pure tone frequencies present in the fan noise spectrum. Most fan noise spectra contain these pure tone components and the design procedures for attenuation should take them into account. AMCA Bulletin 300-67 is now being revised by an Engineering Committee of AMCA and some improvement will probably be made in this area.

Another limitation of AMCA Bulletin 300-67 is the qualification procedure for the semi-reverberant test space. At the present time the qualification for the acceptable test room is not very stringent and it is quite possible that in some rooms a true statistical evaluation of the radiated sound pressure field is lacking. The revision will probably include a change in room qualification for future noise testing.

AMCA BULLETIN 300-67 - FUTURE PLANS

The Engineering Committee which prepared AMCA Bulletin 300-67 was fully aware of these shortcomings but for valid reasons (especially in 1967 - the year the test code was released) made the decision that the test code, as written, fully satisfied the needs of the times and continues to do so. However, AMCA plans to revise this test code as further knowledge becomes available and the demand for more precise information becomes a reality.

A research project is being conducted at Purdue University at the present time on the feasibility of "in-duct" sound testing. This work appears to be very promising and it is quite possible that an in-duct test procedure will be developed in the near future.

FAN NOISE ATTENUATION

In principle, the task of attenuating fan noise to acceptable levels is very simple. The sound power level of the fan is obtained from the fan manufacturer and the acoustical environment in the final installation is specified by the user. The design engineer must provide sufficient attenuation to reduce the power level of the fan to a point where the specified sound pressure levels will be met. This simple task sometimes involves special problems.

Sources of Sound Power Levels for Fans. Typical sound power levels for fans are published in a number of publications.[3,4,5] These values should be used for estimating purposes only and actual sound power levels, especially for critical designs, should be obtained from fan manufacturers.

Selection of the Type of Fans. Obviously some types of fans make more noise than others and the design engineer may have the option of utilizing the quietest fan for that service. In actual practice the choice of fans may be somewhat limited because the requirements for a particular type of fan are set by other specifications than the acoustical requirements. However, the designer should investigate all possible types.[3]

To demonstrate the advantages of fan selection, an example of "after the fact" remedy can be mentioned. At an existing industrial plant the exhaust fumes were being handled by a straight radial bladed fan which generated a strong blade frequency tone and had a high overall noise level. These fans were located on the roof of the building and were causing some complaints in the surrounding community. Since some changes were being made to the existing air handling system, the plant engineer decided to change fan types to meet the new requirements and to eliminate the need for additional noise attenuation. The new fans were of a backwardly curved blade design which have a lower blade frequency component and a generally lower noise level. The "before and after" one-third octave band analysis taken on the roof of the building were as shown in Fig. 1. In this particular case this reduction in noise level was sufficient to stop the community complaints.

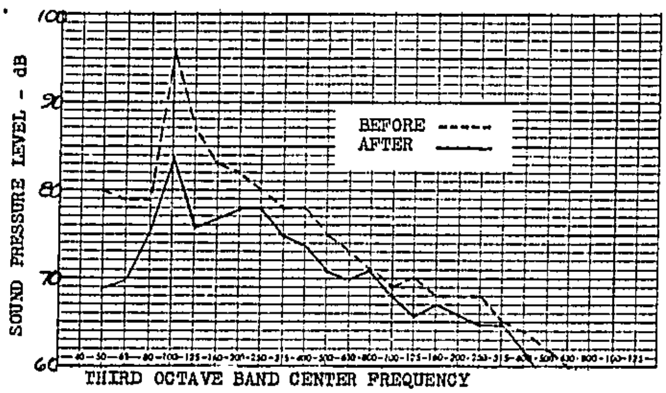


FIG. 1 - FAN TYPE CHANGE

This emphasizes the importance of proper fan selection during the design stages so that changes do not have to be made after the equipment is installed.

In the case of fan type changes, there is a limit to the reduction that can be achieved by this method. The reduction in radiated fan noise will be a definite function of the two fan types and if the specified level is not achieved in this fashion, the additional reduction must be achieved by attenuation.

Selection of Size and Speed. In order to minimize the noise generated by a fan, it is necessary to operate the fan near peak efficiency. This means selecting the proper fan size and speed.

Examples of the differences in noise level that can occur at different points of rating on a fan curve are shown in Figs. 2 and 3. Fig. 2 shows the difference in sound power levels at several different points on a performance curve of a centrifugal fan as indicated in the figure. It may be seen that throughout most of the operating points of the curve, the sound power level does not change significantly. At the extreme position on the curve there is a change but a properly selected fan would not normally be operating in this range.

On the other hand, certain axial flow fans have quite different operating characteristics and will change more significantly as a result of change in the point of rating. An example of these changes is shown in Fig. 3. This should not be interpreted as a disadvantage of all axial flow fans since some axial flow fan types do not change as much as the example used in Fig. 3 and axial fans have other advantages. However, this is a commonly used type of axial and it is important to understand that the point of rating on the fan curve can be significant from a noise standpoint.

This emphasizes the need for good fan selection.

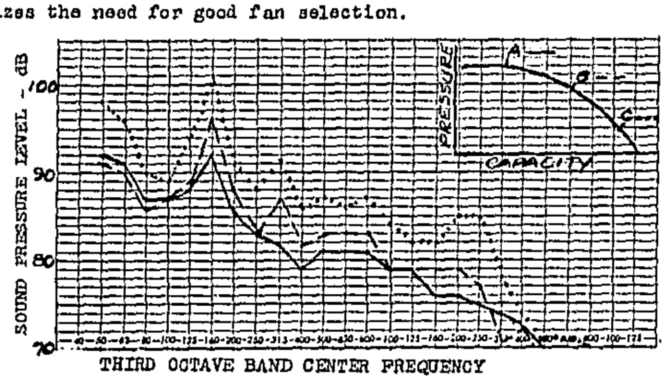
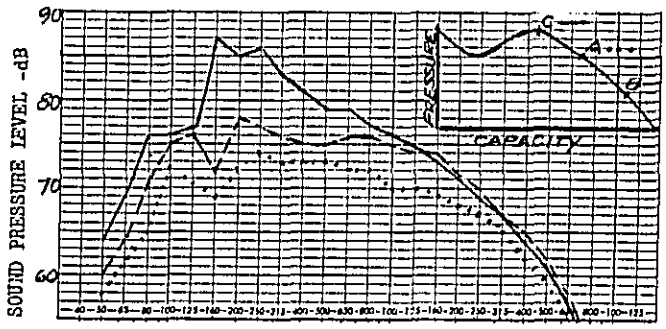


FIG. 2 - CENTRIFUGAL FAN NOISE

Duct and Stack Attenuators. Attenuation of fan noise radiated to a duct system is done by duct attenuators and these attenuators can be used in exhaust stacks as well as duct systems.[6]

In one installation exhaust fans were creating a community noise complaint but it was not feasible to replace the fans with another type. These fans were equipped with exhaust stacks and it was decided to insert noise attenuators into the existing stacks. The "before and after" results of this approach are shown in Fig. 4. This improvement was sufficient to eliminate the community objections.

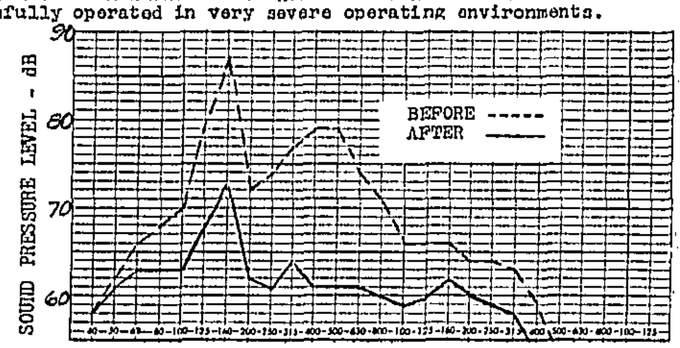
In the case of noise attenuators there is no practical limit to the amount



THIRD OCTAVE BAND CENTER FREQUENCY
FIG. 3 - AXIAL FAN NOISE

of attenuation that can be accomplished. Excess attenuation is uneconomical but if space is provided the noise level can be reduced to any specified level.

Duct and stack attenuators are commercially available and normally do not require special design procedures. Attenuators of this type are cataloged items and can be selected by the design engineer. In those cases where special attenuation characteristics are required, severe operating conditions are encountered or special material must be used, the designer is advised to contact the manufacturer. Attenuators have been constructed and have successfully operated in very severe operating environments.



THIRD OCTAVE BAND CENTER FREQUENCY
FIG. 4 - ADDITION OF ATTENUATOR

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NOISE CONTROL IN THE TEXTILE INDUSTRY

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INTRODUCTION

The textile industry may be considered to include a variety of chemical processing operations having noise problems similar to those discussed in other papers at this symposium. Accordingly, the noise problems associated with plastic production will not be included in this talk.

This paper will concern itself with noise associated with the basic process of fiber processing including carding, picking, blending, followed by examples of noise control in filament processing including twisting, drawing, spinning, etc. The most severe noise problem occurs in cloth production and this discussion will include noise control in weaving, braiding and knitting. Following the actual preparation of the cloth there are usually some noise problems associated with finishing operations, but these are noise problems not directly associated with the handling of the cloth as much as they are the result of ventilation and other fluid flow sources. The actual fabrication of garments can involve noise problems but discussion of this will be omitted.

OPENING,PICKING, AND CARDING

Noise levels found in typical carding operations are shown in Table 1 and can be seen to be fairly minimal in most cases. There are occasional noise problems associated with material handling equipment or ventilation systems, but the application of mufflers on air discharge and ventilation system openings generally will handle the problems in these areas of the textile mill.

FILAMENT PROCESSING

In most cases the noise associated with fiber and filament modification has very little to do with the actual processing of the fibers or the filaments, but rather is associated with gearing and dynamic unbalance in the equipment itself. The increase in noise is usually directly proportional to the speed at which the machine is run and so the slower, older machines have minimal problems while modern, high speed processing equipment quite often have rather significant noise hazards. Noise levels for a variety of different filament processing operations are shown in Table 2. Detailed analysis of the source of noise in such equipment has been performed by Crawford (1) and Bruce (2) and results in some knowledge as to the sources and treatment potential. As one might expect, the obvious noise sources are the gearing and power transmission linkages along with the spindle bearing assemblies.

The higher speed machines now used for texturizing and false twisting tend to also have aerodynamic noise associated with movement of the filament and cavities or perforations in the rotating parts. In some cases speeds in excess of 300,000 rpm mean pure tone sources at frequencies in the upper end of the audible spectrum.

The use of resilient or damped gears and treated enclosures for the head stock results in appreciable noise reduction. In his analysis of the unbalanced forces associated with the rotating spindles Higgs (3) shows that 90% reduction in rail vibrations can be achieved through resilient mounting of the spindle bearing. The reduction of sound in an actual installation is shown in Table 3.

The noise associated with the drive belt mechanism commonly used to drive the spindles has been examined in great detail by Bruce. His analysis showed that changes in the surface of the driven pulley, the width of the belt, and the driving speeds all have pronounced effects on the noise generated by this mechanism. Because of the difficulty in modifying the drive system without detrimental side effects, the approach taken by one machine manufacturer is that of enclosing the drive belt and driven spindles with multiple small enclosures. Other barriers have been designed that enclose the entire bobbin assembly. Reductions of 8-10 dBA with the result of this type have been achieved this way.

In the case of many of the fiber processing machines there is additional noise associated with the speed changers or twisters used as part of the process. For example, Crawford found that drawtwist machine noise included a large contribution by the gearing and bearing assemblies associated with the drawing operation. The reduction afforded by modification of this portion of this machine is 4-14 dB in the high frequency bands.

Aerodynamic noise sources in high speed machines may be the result of siren action associated with cavities of balance holes in the rotating parts. It is not uncommon to balance the spools or bobbins by drilling fairly large holes in the hubs. This type of noise can be eliminated by covering the holes with a smooth surface, such as a thin layer of pressure sensitive tape capable of withstanding the abuse of handling. In much of this type of machinery there are high speed motors which tend to cause considerable noise in the area. The noise is almost always associated with the cooling fans mounted on the motor shaft, and this noise can be eliminated or reduced by adequate muffling as described in other sections of this paper.

WEAVING MACHINERY

The highest noise levels in the textile industry are almost always associated with the actual production of cloth via shuttle looms.

There are a wide variety of weaving machines now available although the bulk of the cloth is still manufactured using one of two basic looms. Namely, the Draper loom which is used for most cotton materials and some synthetics, and the Crompton and Knowles loom which is used for woolens and some synthetics. Both of these looms are fly shuttle looms and use very similar mechanisms for accelerating the shuttle. In recent years a larger number of weaving machines have appeared on the market that do not utilize a filler containing shuttle, but rather push or pull the thread across in another manner. Such looms as the Sulzer, the Draper shuttleless loom, the water jet loom, all have different characteristics and produce varying amounts of noise. They are also all restricted in the kinds of materials that can be woven. Spectra of these various types of looms are shown in Table 4. These spectra are for multiple installations on the looms with an inter machine spacing that would commonly be found in industry.

It can be seen from Table 4 that the weaving room noise problem would be considerably less if most cloth was woven on the water pick type of loom. At the present time this is not feasible since many materials would have water damage associated with the inherently wet process of the water jet loom.

In all these weaving machines there is discontinuous motion due to the fact that each thread must be pushed into place after it has been threaded through the warp. Because of this discontinuous motion there are impulse sounds associated with most weaving machines. These impulse sounds are of a very brief nature and in most cases lasting less than 20 milliseconds each. Because such brief pulses in an individual machine cannot properly be measured on a conventional sound level meter, noise control studies are facilitated by a special type of analysis. What one would really like to do is monitor the maximum instantaneous value of the sound over a time period of 5 to 10 milliseconds and be able to average this over ten or more cycles of the loom in such a way as to allow examination of each impulse that occurs during the loom cycle. To accomplish this in an analog fashion one simply uses a timed gate (this must obviously be a stepless gate) and a variable time delay synchronized with some point on the loom cycle. By slowly varying the time delay one can in effect slowly scan the window across the entire loom cycle and graphically record that which represents the instantaneous sound averaged over a number of loom cycles for each portion of the loom cycle. On this basis, one can then examine the amount of energy from each of the loom pulses and determine the effectiveness of each control measure that is installed.

Using this scanning gate technique one finds that the Draper X2 loom has a distribution of impulse energies, as shown in Table 5. From this table it can be seen that many of the individual pulses must be reduced or eliminated in order to achieve more than a few decibels of a noise reduction in the total loom signal. Correlation of accelerometer data and high speed photographs makes it clear that the loom noise is largely associated with accelerating and de-accelerating the shuttle. There are a number of impact points subjected to high loadings during these times of the loom cycle, and it is the resulting vibration of the surfaces and adjacent parts that creates the noise in the loom. Replacement of parts with resilient materials instead of metal produces a pronounced reduction in the impulse noise in the loom, and allows some noise reduction to be achieved without adverse effects on the loom operation. Although changes as great as 15 decibels have been achieved it is probably unlikely that a reduction in excess of 10 decibels can be expected under production conditions, since the amount of noise is very dependent on the various adjustments of the loom. At this time no large scale installations have been attempted in the weave room, and one can only speculate on the basis of noise changes measured in the laboratory.

Other approaches to the control of loom noise have involved various partial enclosures over the parts of the loom known to be associated with noise generation, and some of the changes brought about by this type of approach are shown in Table 5. Unfortunately, the major noise source is associated with the boxing surfaces and so an enclosure over that part of the mechanism would be required if more than a few decibels of noise reduction are desired. Of course the loom operator must have access to the adjustment points around the box and the shuttle must be re-supplied with filling fairly often, so that such an enclosure around the box would be almost impossible to live with. There have been attempts to produce entire enclosures around the loom, and this is certainly one solution to the loom noise problem. Unfortunately, again, access must be provided to all parts of the loom and in most weave rooms there is insufficient space around the loom to be able to open such an enclosure.

KNITTING MACHINES

Typical spectra for knitting machines are shown in Table 6. In most cases the high noise levels are the result of material handling equipment or vacuum systems rather than the knitting process itself, but there are some high speed knitting machines that produce levels in excess of 90 decibels on the A scale. At the time of this writing there appears to be no concerted effort to reduce knitting machine noise so little additional comments seem to be warranted.

TABLE 1

FIBER PREPARATION

Example	Decibels	A Scale	C Scale
Opening		87	90
Picking		90-92	98
Carding		90-92	99
Combing		88-93	

TABLE 2

FILAMENT PROCESSING

Example	Octave Band Center Frequency							
	63.5	125	250	500	1000	2000	4000	8000
Spinning	88	84	86	86	84	84	82	74
Spinning	82	85	88	89	87	85	81	78
Drawtwist	83	86	87	89	90	88	88	90
Texturizing	90	78	78	81	85	88	87	90
False Twist	86	80	80	80	83	90	90	95

TABLE 3

DRAWTWIST NOISE CONTROL IN PRODUCTION MACHINE

Example	63	125	250	500	1000	2000	4000	8000
Normal Machine	83	94	91	93	96	93	87	84
Modified	78	86	91	82	84	83	82	76

TABLE 4

WEAVE ROOM NOISE

Example	Octave Band Center Frequency							
	63	125	250	500	1000	2000	4000	8000
Cotton-Draper X3	91	92	95	102	102	103	100	91
Cotton X2	90	89	92	95	97	97	92	83
Draper Shuttleless	98	92	92	90	89	87	82	77
Sulzer	87	90	90	89	87	88	83	78
Water Jet	78	76	76	78	78	79	79	72

TABLE 5

LOOM NOISE ENERGY DISTRIBUTION

NOISE ENERGY IN MILLIWATT-SECONDS FOR IDENTIFIABLE PULSES DURING ONE HALF OF THE LOOM CYCLE X2 LOOM

Motion or part involved	A		B	C	D
Boxing					
Binder A_{1L}	1006	*(5.5%)	1030	412	860
Picker Stick (A_{2L})	4274	(23.6%)	3588	808	615
Lay End Stop (A_{3L})	1740	(9.6%)	1584	280	429
Protector Rod (A_{4L})	99	(0.5%)	51	54	48
Pickout					
B_{1L}	723	(4.0%)	574	342	266
B_{2L}	901	(4.9%)	679	409	425
C_{1L}	303	(1.65%)	226	234	78

*Percent of total noise for the complete loom cycle.

A. Unmodified loom
B. Box surfaces treated and nylon pick ball.
C. Box surfaces treated, enclosure over box, enclosure over link parallel, and nylon pick ball.
D. Box surfaces treated, enclosure over link parallel, overthrow cushion, and nylon pick ball.

TABLE 6

KNITTING MACHINE NOISE

Example	Octave Band Center Frequency							
	63	125	250	500	1000	2000	4000	8000
Tricot Knitting	90	92	92	90	88	86	82	73
Knitting	78	77	81	78	78	76	74	73
Knitting with Waste Vacuum oper.	79	77	81	78	80	86	84	84

BRAIDING MACHINE NOISE CONTROL

Introduction

With the advent of the Occupational Health and Safety Act tremendous pressure has been applied upon manufacturing plants to eliminate job hazards and improve working conditions in their plants. One major hazard in many plants, which is also considered to be one of the top three pollutions, is noise. It has been shown that noise is not only detremental to a worker's health in that it causes loss of hearing over a period of time, but it will also effect productivity since it lowers the efficiency of the worker if he is subjected to excessive noise during his work day. One industry which is having a great deal of difficulty in conforming to the noise levels established by OHSA is the textile industry. This paper will deal with one particular part of textiles which is the braiding operation. In many braiding plants noise levels as high as 101 dB(A) have been recorded which is 11 db above the maximum permissible exposure level as prescribed by OHSA.

In order to lower the noise level in braiding plants it was necessary to reduce the noise at its source namely the braiding machine. The braiding machine is constructed with two plates between which a train of gears operates. These gears have flanges at the top which engage the bottom end of the braider carrier on which the yarn is held. The upper of the two plates is called the top plate and is formed with a serpentine path to guide the carriers. The braider carrier is made to such shape that it fits into the grove in the top plate by which it is guided and the lower end of the carrier is in the form of a lug which engages in the flanges of the gears by means of which it is carried around the machine. It is through the action of the carriers criss-crossing each other, as they are guided around the top plate, that the yarn is made into braid.

It was impractical to consider the use of enclosures or acoustical panels due to the physical setup of braiding machines in the plants. In most instances the braiding machines are set up on benches in rows which usually run the length of the plant. The rows of machines are just wide enough for maintenance thus it would be very inconvenient if the operators had to work around enclosures. The outcome would probably be reduced production and little noise reduction.

Laboratory Testing

The analysis of the noise problem of the braiding machine was conducted in a test laboratory with a typical braiding machine set up for production running. The machine used for the noise measurements was a 13-carrier braider which uses the mosspeed type carrier. This type of carrier is used to increase production on the braiding machines as compared to the older type of carriers which must be run at lower speeds. In order to measure the noise level a type 1558-BP Octave-Band Noise Analyzer with type 1560-PG Microphone Assembly was used with a type 1562 Sound Level Calibrator. A braiding machine emits a broad-band noise and thus in analyzing the noise problem an octave band sound level meter of the 1558-BP type was ideal for the noise measurement.

Table 7 shows the octave band noise levels which were recorded with the 13-carrier braiding machine running at a handle speed of 340 RPM, which is typical of the speed that the machine would run at in production. The microphone was located about 18" from the front of the machine and about 10" above the top plate of the braider. (46" from the floor). From the graph of the octave bands we can see that the highest noise level occurs at 4,000 Hertz, the overall noise level of the machine was 94 dB(A). A study of the mechanical function of the braiding machine showed that the high noise level was due to the cast iron carriers changing tracks as they pass from one cam to the next in the top plate. Since the cams and plates are made from steel, there exist a high resonant condition in the system which aide in generating and conducting noise.

Table 7

Thirteen carrier braiding machine running at 340 RPM handle speed (Steel carriers).

Octave Band Center Frequency

	63	125	250	500	1000	2000	4000	8000
Readings	67	71	72	72	82	89	89	87

Polyurethane Braider Carriers

In order to reduce the noise we investigated the possibility of replacing the cast iron carrier bases with an injection moldable plastic material with good sound damping properties. Many types of plastics were tried before one was found that would satisfy the requirements necessary to run properly in the braiding machine. All of the plastic carrier bases that were made would somewhat satisfy the requirement of all over noise level but most of them were not structurally sound enough to function properly in production running conditions. The material that was finally chosen was an injection moldable polyurethane which was stiff enough to work properly in the braiding machine but still retain enough inherent sound damping properties to reduce the noise level of the braider. Table 8 shows the octave band readings which were taken from the 13 carrier braiding machine running at a handle speed of 340 RPM using the new polyurethane carrier bases. The overall noise level of the machine now was 83 dB(A). If we compare Table 7 with Table 8 we can see the large drop in noise level at the higher frequencies. The overall drop in noise level between the cast iron carrier bases and the polyurethane bases was 11 dB(A) which is quite significant.

Table 8

Thirteen carrier braiding machine running at 340 RPM handle speed (Polyurethane Carriers)

Octave Band Center Frequency

	63	125	250	500	1000	2000	4000	8000
Readings	53	56	61	67	72	78	78	77

Plant Testing

Since we had proven that we could lower the noise level of the braiding machine in the laboratory, the next step was to install the new carrier bases into a braid plant and measure the noise reduction of the braiding machines in production.

It was agreed that the J. P. Stevens North Carolina Braid Plant would set up a complete row of braiding machines equipped with the plastic carrier bases, and adjacent to it a row of machines with the original cast iron carrier bases. The braiding machines were of the 13 carrier type which run at 340 RPM handle speed. There were 84 machines in each row which meant we were testing a total of 1094 plastic carriers. The readings were taken using a General Radio 1565B sound level meter with the microphone located 3 feet from the center line of the row of machines, and about 3 feet above the floor. In addition, the output of the sound level meter was recorded on a Norelco model EL 3302 cassette tape recorder for octave band analysis in the laboratory. The following results were obtained from the sound level meter with the microphone pointed at the braider row center line.

Condition	Reading in dB(A)	Reading in dB(C)
1. Both lines operating	97	96
2. Metal carriers only	97	96
3. Plastic carriers only	90	89
4. Both lines shut down (background noise)	85	84

The background noise was associated with the other braiding machines in the vicinity of the test rows. In addition, there was noise associated with the ventilation system and the drive motors of the braiding machines. We should note that the background noise is sufficiently high so that a true measure of noise reduction of the carrier modification cannot be accomplished. The actual reduction achieved is 1-2 dB more than the 7 decibel reduction measured. Thus the actual reading of the row of eighty-four machines with the polyurethane carriers would be 88-89 dB. This indicated that the modification of all the braiders in the plant with plastic carriers will result in operator exposure noise levels which will be below the 90 dB(A) level established by the Occupational Health and Safety Act.

Aside from the important fact that the polyurethane carriers lower the noise level of the braiding machine there are also several other benefits which may be gained through the use of the new carriers. A) Less weight (the polyurethane carriers weigh half as much as the cast iron carriers), B) Less vibration in the braiding machine, C) Less power loss since the mass inertia of the system is lower, and D) Higher production output.

The braiding machine is one case in which the attempt to solve a noise problem has brought with it several other factors which make it more appealing to the manufacture. The new polyurethane carrier is now being manufactured by Stevens Molded Products and marketed through Mossberg Hubbard, Division of Wanskuck Company, who are the sole suppliers of braiding machines.

Summary

In summary we have been able to prove both in the laboratory on a single test braider and in the plant that through the use of the polyurethane carriers the noise level of the braiding machine can be reduced so as to meet the maximum noise exposure level as prescribed by the Occupational Health and Safety Act.

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PRACTICAL DESIGN OF MACHINERY FOUNDATIONS FOR VIBRATION AND NOISE CONTROL

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INTRODUCTION

Optimum noise and vibration control requires that a machinery foundation be considered as one element of a vibratory system. Other elements include the structure supporting the foundation, the isolation system, and the machinery itself. In this paper emphasis will be placed on the foundation and its isolation system. Foundation configurations and design parameters will be discussed in terms of isolating the machinery from the foundation. Finally, a realistic review of isolation systems will be presented.

FOUNDATION CONFIGURATIONS

The more important parameters controlling the noise and vibration performance of a machinery foundation are:

1. Physical dimensions
2. Stiffness
3. Mass
4. Local characteristics at the points of equipment attachment

The first three factors determine the resonance frequencies of the foundation and, of course, it is desirable that these frequencies do not coincide with high energy vibration frequencies of the supported equipment. In reality, it is impossible to design a foundation having no resonant frequencies, so a designer must take care to insure that the lowest resonance frequencies of the foundation are removed from the excitation frequencies of the mounted equipment.

The local dynamic characteristics of a foundation are important factors in determining its noise and vibration performance. For example, if the mounted equipment is supported at the modal points of a particular resonance frequency of the foundation, it is difficult for the equipment to excite this mode. This case is shown in Figure 1 for the first bending resonance of a foundation.

In addition, it is particularly helpful at higher frequencies to make the local attachment points of the supported equipment as stiff and massive as possible. These localized areas represent points where structureborne energy traveling away from the equipment can be reflected back toward it.

FOUNDATION DESIGN PARAMETERS

The problem of specifying and attaining the noise and vibration performance of a foundation is especially difficult because of the inability to separate the foundation performance from the performance of the elements on both its input and output sides.

For example, specification of performance in terms of the supported equipment can lead to the erroneous conclusion that the foundation itself is poorly designed—when in reality it is the supported equipment at fault. Moreover, specification of vibration levels on the structure below the foundation can also lead to incorrect conclusions.

The common error, then, lies in the specification of performance in terms of inappropriate parameters to be measured. Suppose it is desired to reduce the noise transmitted through isolation mounts to a foundation. If the performance criteria are specified in terms of the force transmitted to the foundation by the mounts, two typical foundations, (A) and (B), might produce performance curves as shown in Figure 2.

(A) differs from (B) in that its mechanical impedance is lower, and, clearly, (A) is better than system (B) according to the "force transmitted" criteria previously mentioned.

Let us change the performance criteria, now to acceleration or velocity levels measured on the foundation. The new performance curves appear as shown in Figure 3. The performance of foundations (A) and (B) has reversed, with configuration (B) now the better of the two. These seemingly contradictory results are not inconsistent, however. Foundation (A) does receive less force than (B), but because of its lower mechanical impedance, (A) will experience greater motion than (B). Hence, in this case, acceleration or velocity·level specification would be the correct criteria on which to base performance.

All of this points to the fact that one must exercise caution in defining the performance criteria for a machinery foundation. Ideally, one would like performance criteria independent of both the supported machinery and also the structure supporting the foundation. This criteria must, of course, still relate to the required functions of the foundation. In the likely event that both of the above cannot be achieved, all characteristics of the total system should be specified. In the previous example, the performance criteria should explicitely describe the machinery that is mounted, its operating conditions, the characteristics of the mounts used, and the characteristics of the structure which supports the foundation. In this way, the correct parameters to be measured on the foundation can be determined.

An alternate approach would be to specify the performance criteria in terms of the foundation properties as a function of frequency. These properties would include such factors as the desired natural frequencies of the foundation, its local and overall impedance, the damping present, and other pertinent factors. Now that we have a set of performance criteria, we can begin the job of isolating the machinery from the foundation.

REAL ISOLATOR PERFORMANCE

The performance of real vibration isolators often differs from that predicted by classical theory because both the isolator materials and geometry are, at best, crudely characterized in this simple theory. In fact, simple theory sometimes ignores effects which can control the total isolator performance.

Steel springs, often used to isolate vibrations, are highly resistant to most industrial contaminants and can operate over rather broad temperature ranges. Their use is generally limited to low frequency isolation problems where they can provide adequate and effective isolation. Steel springs have high frequency limitations, however, because of inter-coil resonance or spring surge. This occurs when the mass of the coils interacts with the stiffness of the spring to produce a resonance within the spring itself. Clearly, such a resonance magnifies the vibration input, thus increasing the vibrational output to the detriment of the isolation.

In certain cases, a steel spring isolation system may occasionally operate near resonance so that snubbers and/or auxiliary dampers must be used to limit the system's motion. Severe damage or failure would result without these external limiting devices.

Another common type of isolator is constructed from polymeric (rubber-like) materials which can resist a great number of industrial environments. In other cases, these isolators can be protected from the environment by using protective coatings and housings.

Polymeric isolators can be designed for low frequency applications, although this is rather more difficult to achieve than with steel springs. Like steel springs, polymeric isolators are also subject to internal resonances called "standing waves". These resonances result from wave propagation through the body of the isolator. Vibrational waves propagate within the body of the isolator because the stiffness and mass of the polymer are distributed within the volume of the isolator. As the vibrational waves propagate through the body of the isolator, standing waves, similar to those generated in organ pipes, are set up. These stanidng waves cause some degradation of the isolation, but it is not as severe as with steel springs. The lack of severity is a result of the relatively high internal damping that polymers have in comparison with steel springs which have nearly no damping. The distributed internal damping of polymers extracts energy from the waves as they travel in the polymer. Since part of the energy of the traveling wave is dissipated, less energy is left to be magnified by the standing wave resonance, and the isolation is not severely degraded.

Figure 4 compares the vibration isolation of a steel spring and polymeric isolation system. Note that the steel springs provide good low-frequency isolation; however, at high frequencies, the performance is rather seriously degraded by resonances within the spring. In contrast, the polymeric isolator provides effective isolation, even at high frequencies. Standing waves are well-controlled and do not seriously compromise the vibration isolation.

Air mountings are used in certain special applications. These mountings consist of volumes of air contained within rubber bags or metal cylinders. Air mountings can provide extremely good isolation. They combine the low natural frequencies that can be obtained with steel springs with high-frequency attenuation that is free from standing waves. It is, however, necessary to design the air spring so that solid-borne vibrations propagating in the housing containing the entrapped air do not short-circuit the isolation system. It should also be noted that the increased complexity of the air spring often results in greater cost and maintenance. Thus, air mountings are used only where critical demands justify the increased cost.

The isolation systems already described consist of discrete isolators applied at specific "points" of the machinery foundation. In addition to this, "distributed" systems composed of machinery isolation pads may also be used effectively. Isolation pads may be made of rubber, cork, felt, fiberglass, and of various layered combinations. These pads may be inserted directly under all or a portion of the machinery foundation. The user should make certain, however, that the pads operate within their safe-loading ranges and that the area loading is chosen such that the proper natural frequency can be obtained. In addition, vibration pads should be chosen for the proper service life and environmental resistance.

STRUCTURAL DAMPING — ITS EFFECTS & KINDS OF TREATMENT

We have already seen that damping controls the resonant motion in isolation systems. Damping can also be applied in different forms to help control structural resonances within the machinery foundation and its supporting structure.

Clearly, the best procedure is to design the machinery foundation and supporting structure so that none of their resonances fall within the frequency range of concern. This is not always possible nor economically feasible. Many machines emit vibrations that cover a broad frequency spectrum. In such cases, it is impossible to design foundation structures that will not have resonances within the range of emitted frequencies. Similarly, it may be economically unreasonable to design complicated machinery foundation structures to avoid resonances. This is particularly true when the supported machinery is quite large and massive. In such cases, structural damping treatments may be used to considerable advantage.

In limiting the resonant motion of structures, damping makes the structure appear more massive. This is illustrated in Figure 5. Here the mechanical impedance of a beam is shown with and without an applied damping treatment. Note the increase in the apparent mass of the damped beam at the resonant frequencies. This increased massiveness or decrease in resonant motion inhibits the flow of vibrational energy through the machinery foundation and into the supporting structure.

Damping treatments exist in tremendous variety. A discussion of all of them is clearly beyond the scope of this paper. However, damping treatments generally fall into two basic categories: Unconstrained and constrained.

An unconstrained treatment, in its simplest form, consists of a layer of rigid but highly damped polymer adhered to the surface of a vibrating structure. Energy is extracted from the bending vibration of the structure due to extension and compression strain in the polymeric layer.

A constrained layer treatment consists of a soft, highly damped polymer sandwiched between two stiff structural elements. When the structural elements are deformed by bending vibrations, the soft polymer is strained in shear. Energy is extracted by this shear deformation.

Unconstrained damping treatments are relatively inexpensive and simple to install. Solid tiles of damping material may be cemented to vibrating structures or liquid dispersions may be sprayed on. The damping performance of unconstrained treatments is not frequency dependent. Their temperature bandwidth, while somewhat restricted, is adequate for many industrial applications.

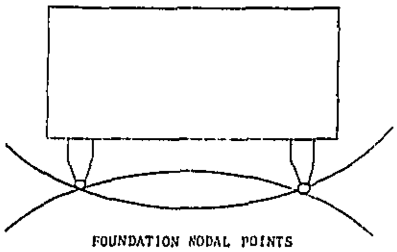
Constrained layer treatments offer greater damping and broader temperature range. However, these treatments are often more expensive. Their damping performance is frequency sensitive, and each application should be checked to be sure that the proposed design is compatible with the important frequency range.

CONCLUSION

In looking at machinery foundations, we found that there are certain factors that must be considered in any realistic design:

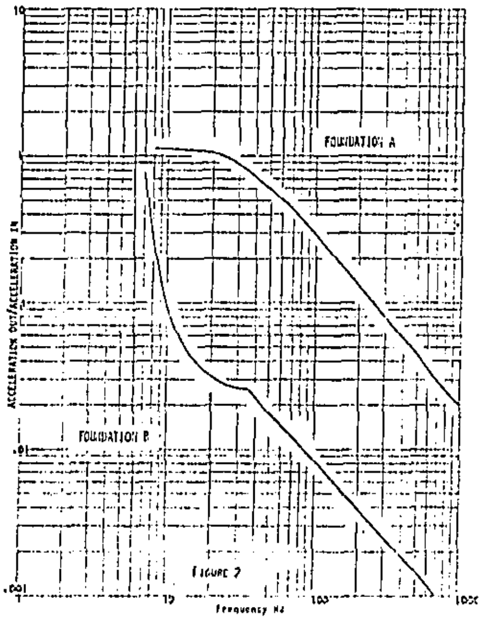
1. The frequency spectrum of the vibrations generated by the machinery,
2. The dynamic characteristics of the foundation, including its resonance frequencies, modal patterns, and internal damping,
3. The dynamic characteristics of the structure supporting the foundation,
4. The elastic and dissipative characteristics of the isolators, whether steel springs, polymeric isolators, or distributed vibration isolation pads, and
5. The geometric arrangement of the isolator attachment to the foundation.

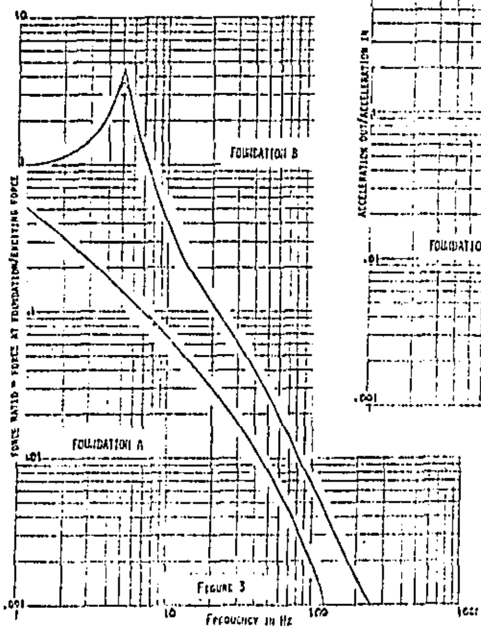
Each of the above items should be considered both separately and in terms of its effect on total system performance. To neglect any one of these items could result in poor foundation performance.



FOUNDATION NODAL POINTS

FIGURE I





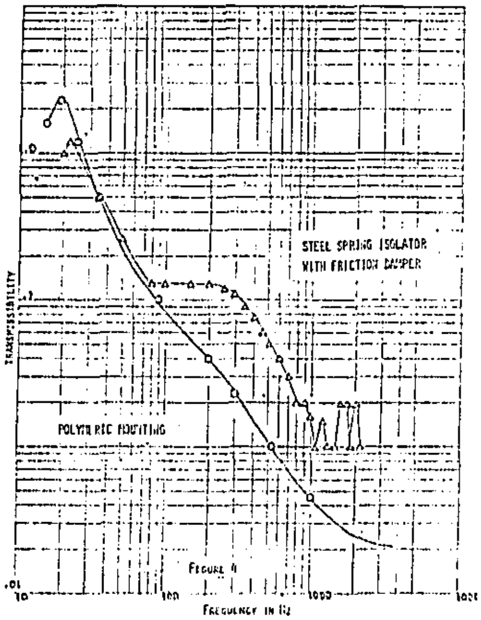


FIGURE 4

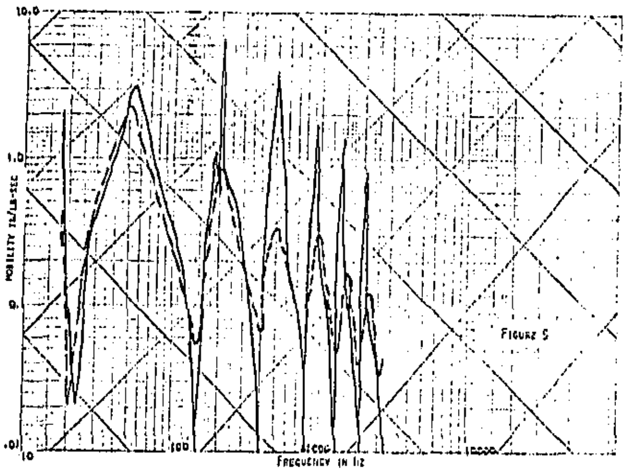


FIGURE 5

THE SYSTEMS CONCEPT FOR GEAR NOISE

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Noise reduction has become a challenge for many of us. Gear noise, because of its discrete frequency characteristic, presents a particular problem.

Gear design and manufacture are specialized techniques and from years of specialization, gears have evolved as mechanisms unique to themselves. Let's review some of the basics of gear teeth to determine the factors affecting noise.

Contrary to general opinion, gear teeth are not conjugate, that is, they do not transmit uniform motion, except perhaps at one load condition. Gear teeth are generally modified or crowned both in the profile and lengthwise directions. This crowning permits the contact between the teeth to spread over the full tooth surface, without concentrations at the edges, under the full prescribed load. The crowning is necessary to compensate for the deflections of the teeth, the blanks, and the mountings under this full load. At any other load, the deflections are different and the modifications to the tooth surfaces cause slight departures from conjugacy resulting in a non-uniform transfer of motion.

The error which results from the crowning or mismatch of the tooth surfaces is very small. For bevel gears, we have defined this variation as an angular displacement error because of the rotational nature of gearing. Experiments leading to the measurement of this gear parameter, and the development of a computer program to enable evaluation of it, indicate that the order of magnitude of once per tooth mesh displacement error is approximately 1.2 seconds of arc for a quality bevel gear set under light load.

Gearing not only produces noise at tooth mesh frequency, but also at multiples or harmonics of it. Considering the tooth mesh frequency as the Fundamental or first harmonic, Fundamental, Second, and Third Harmonics are usually most predominant and therefore of most interest.

To understand gear noise, we must first consider the system in which the gear set is operating.

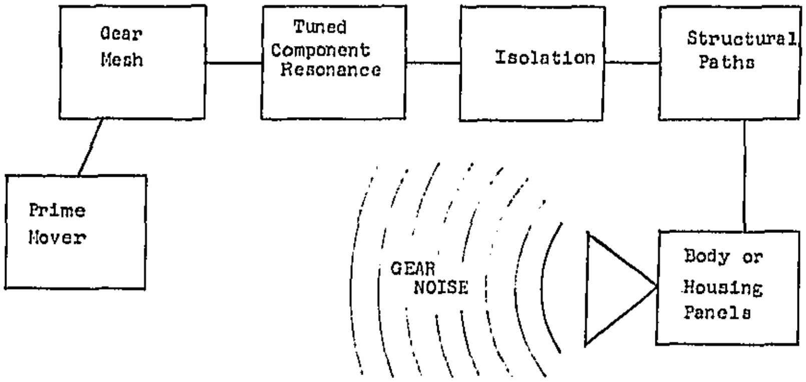

Fig. 1 – TYPICAL GEAR NOISE SYSTEM

Although some gearing is open, most applications have totally enclosed gear sets. The noise commonly called "Gear Noise" is not emitted directly from the gear set but rather as a response of the entire assembly to gear mesh vibration. The gears, as part of a power train system, act as a vibration generator with the output frequency dependent on the numbers of teeth and the rotational speeds of the corresponding shafts.

The shafts, housing members, and other power train components usually exhibit tuned resonances, which amplify the minute gear mesh vibrations when their frequencies are equal to those of the component resonances. The amplified vibrations are then structurally transferred to radiating panels which convert the structural vibration into the airborn sound called "Gear Noise" much in the same manner that a loudspeaker converts electrical impulses into sound.

The necessity of the system approach arises from the finding that the vibration created by the gear set maintains a constant displacement error for a constant load, even under varying speed conditions. The resultant sound output would be constant at all frequencies if it were not for the tuned resonances within the power train system.

When measuring gear noise, it is necessary to use equipment capable of narrow band filtering, particularly with complex systems, having several potential noise sources. A measure of overall noise does not adequately define the source. Very often, the gear noise amplitude will be lower than, and therefore buried, in the overall sound level.

Even though the gear noise may be lower in amplitude than the overall sound level, its discrete frequency is often very annoying. The annoyance is a result of the human ear's ability to hear a pure tone sound buried in the ambient noise, provided the two sounds have enough frequency separation.

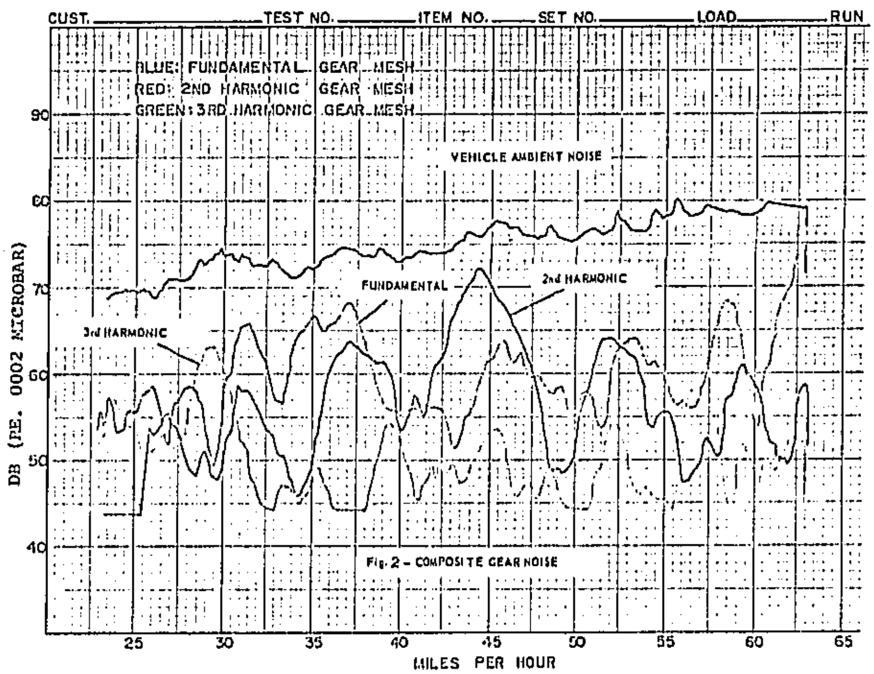

Fig. 2 - COMPOSITE GEAR NOISE

Since our goal is to reduce gear noise to an acceptable level, the following points may give an insight into some of the practical limitations on the reduction of gear noise.

The amount of vibration generated by a gear mesh is governed by the amount of localization required to accept the mounting deflections under load. When reducing gear noise at the source, the amount of localization or mismatch must be revised so that in order to effect a 6 dB change in sound pressure level, the excitation vibration at the source must be reduced by 1/2 of its already minute value. Making such a change at the source would require significant strengthening of the mountings so that the gear set will operate satisfactorily at the prescribed maximum load.

The angular motion error affects the noise output and relative position of the gear members affects the angular motion error. Close assembly tolerances will assure the correct position for minimum angular motion error and noise output.

Having reached the point where gear development and assembly are optimized, further reduction must be accomplished by revision to the system components by eliminating or changing resonant conditions.

SURFACE TRANSPORTATION NOISE

NOISE RESEARCH AT NPL WITH PARTICULAR REFERENCE TO ROAD TRAFFIC

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1. INTRODUCTION

One aim of current NPL noise research is to establish methods for rating the environmental impact of noise by establishing correlations between objective and subjective measures. The use of A-weighted sound level as an instantaneous measure of noise level is now almost universal but as yet there is no general agreement on what constitutes a satisfactory single measure of noise exposure, although various units have been proposed. L_{eq} continues to be widely used in continental Europe for a range of applications, NNI has been used specifically for rating overall exposure to aircraft noise, and TNI has been used for rating traffic noise exposure. The index known as Noise Pollution Level, which involves algebraic processing of the statistical distribution of noise levels, has been put forward by NPL (1) as a means of embracing various common types of environmental noise. Experimental work confirms that fluctuations in level increase the adverse reaction to noise (2,3) and further developments of the index are in progress to take into account other significant features of the temporal pattern. However, the more closely such formulae mirror the complex reaction of people to non-constant noises, the more complicated they tend to become. The question of standardizing a unified scale of measurement having general predictive validity must therefore involve a compromise between fidelity to the observed subjective response and reasonable simplicity for practical use by engineers and planners.

Meantime, and as an interim measure only, the Noise Advisory Council in the United Kingdom recently recommended the adoption of L_{10} as an index for rating the disturbance caused by traffic noise. They have recommended that the average value of L_{10} (18-hour average over the period 06.00 to 24.00 hrs) at the exposed facade of houses adjacent to new traffic routes should not, as a conscious act of public policy, exceed 70 dB(A); this cannot be implemented until an acceptable design guide become available. In many situations a significantly higher noise standard will be demanded and noise level predictions covering a much wider range of road/housing configurations are thus required.

Thus there is an immediate demand for predictive methods for L_{10} but in the very near future information on the complete time-level distribution for traffic noise will be needed.

The NPL Research Programme currently includes studies of traffic noise from several different aspects; field studies of noise propagation in well-defined urban situations (4) and the effects of mean speed, flow and composition on noise level (5); scale-model studies of propagation in complex urban situations and the effect of road configuration (6); computer-simulation techniques (7) and the computation of diffraction and shielding by simple barriers interposed between the traffic stream and the observation point (8).

Physically the problem can be resolved into

(a) predicting the noise level at a reference distance in terms of the estimated traffic parameters (flow-rate, mean speed and percentage of heavy goods vehicles)

(b) predicting the sound field in terms of the road configuration, the intervening ground cover, and the complex urban fabric

and these are considered in the following sections.

2. PREDICTION OF NOISE LEVEL AT A REFERENCE DISTANCE

There is more than one approach to establishing prediction methods. Computer simulation of traffic noise using random "snapshot" techniques to estimate the time-level distribution has been employed (9) and the author has extended this to produce sequential samples and thus a true time-level history (7). Such simulation techniques can yield valuable data but they do require detailed input information and so far they have not fully taken into account the varied driving conditions adopted by drivers of motor vehicles on the highway.

As an alternative approach empirical relations can be derived from analyses of field data and recently a simple scheme for predicting noise levels for freely-flowing traffic has been established, based on an analysis of field data obtained from measurements on open sites near substantially straight and level portions of motorways and major roads (5).

These NPL field studies comprised nearly 100 samples of noise measured at a height of 1.2m above ground at various sites adjacent to the road, with a time-level distribution curve at a reference distance. The traffic volume and compositions for both carriageways had been recorded together with a histogram of the speed of the individual vehicles on the nearside carriageway during each 15-minute sample period. From the distribution curves the values of L_{10}, L_{50} and L_{90} were derived and separately used, together with the arithmetic mean speed of the vehicles, v, the total traffic flow, Q, and the percentage of heavy goods vehicles, p, in multiple-regression analyses. The reference distance used in the original work was 7.5m which in some cases proved inconveniently near the traffic stream so that adjustment to a distance of 10m has been effected assuming propagation over hard reflecting ground. In addition available evidence suggests that the intrinsic noise level of motor traffic has produced slight increases in noise level since the original data were obtained, and appropriate corrections for this have also been made. This results in the following regression equations for the unobstructed values of noise level at a distance of 10m from the traffic stream:

$$L_{10} = 18.1 + 16.2 \log v + 8.9 \log Q + 0.117 p$$

$$L_{50} = -3.0 + 13.0 \log v + 15.1 \log Q + 0.096 p$$

$$L_{90} = -24.2 + 9.8 \log v + 21.3 \log Q + 0.075 p$$

where v is the mean traffic speed in km/h
Q is the total rate of traffic flow in vehicles/h
p is the percentage of heavy vehicles (over 1500 kg).

Using these equations, Tables permitting noise levels at the reference distance to be predicted without the need for calculation are readily prepared (8).

It should be noted that the parametric variation of noise level with traffic parameters is substantially different from that implicit in several prediction schemes which have been widely used in the United Kingdom (10,11). Previously the effect of mean speed on overall noise level has been greatly overestimated whilst the contribution of heavy vehicles has been underestimated.

3. VARIATION OF NOISE LEVEL WITH DISTANCE OVER OPEN GROUND

In addition to the measurements at the reference distance, well over 100 measurements have been made at distances d of up to 160m from the various roads for a microphone height of 1.2m above ground. At five sites the ground cover was short grass, at two others it was growing wheat approximately 0.6m high, whilst the remaining three sites could broadly be classed as cultivated ground. Limited measurements (out to only 25m) were also available for propagation which, apart from a small grass verge, was over a hard concrete surface.

Inspection showed that the data for propagation over grassland were highly consistent. A linear relation between noise level and log d represented a good approximation to the measured attenuation data and the best-fit line was obtained by least-squares fit. In the case of L_{10} the slope was given by -14.8 log d, the correlation coefficient amounting to -0.96; values for L_{50} and L_{90} are given Table 1 (see also Fig. 1).

Data relating to propagation over growing wheat were less numerous but were treated similarly and yielded attenuation rates approximately 50% greater than those found for grassland. Data for propagation over cultivated ground appeared to be less homogeneous and a significant difference between sites was found. As was to be expected, attenuation rates for propagation over a hard reflecting surface were significantly less than those for grassland.

L dB(A)	Concrete	Grass	Cultivated	Wheat
L_{10}	10.5	14.8	15.9 - 19.8	21.9
L_{50}	8.4	11.1	11.2 - 17.0	16.6
L_{90}	6.1	8.2	7.4 - 14.1	11.4

Table 1. Attenuation rates for propagation over open ground. (coefficient of log d where d is distance from the traffic stream)

There are grounds for expecting the relation between L and log d to be other than linear for propagation over typical absorbing ground and there is also evidence from the computer simulation that the attenuation rate depends to some extent on p and Q. However, data covering a wider range of variables, would be required before a more complex relationship could be established empirically.

The above data relate to a height of 1.2m above ground but for practical application predictions for other heights are required, particularly for the commonly-met case of open grassland. Now excess attenuation associated with near-grazing propagation of sound over the ground has been widely reported and a theoretical explanation of the phenomenon has been established (12) Basically the sound-wave reflected at the ground surface interferes with the wave propagating directly from the source to the receiver and causes a frequency-dependent attenuation which depends on the height of the source and receiver and on their horizontal separation. Direct methods of calculating the magnitude of the effect and nearly all previous studies have used point sound sources. However, a computer simulation of road traffic on up to six lanes has been developed at NPL which takes into account the effect of having randomly-spaced moving sources and includes the effects of typical ground absorption for an A-weighted vehicle noise spectrum. This programme has been used to predict the levels of L_{10} at various distances up to 120m from the nearside kerb and at height of up to 10m above ground level, and forms the basis of a set of contours relating to propagation over completely flat ground under isothermal conditions with no wind gradient (see Fig. 2).

At a height of 10m above ground the contours merge into the corresponding contours for propagation over hard reflecting ground, for at this height the effects of ground absorption are negligible out to distances of order 100m from the road. On the other hand, at a height of 1.2m above ground the contours agree closely with the field data on traffic noise over typical grassland, reported above, which indicate a logarithmic decrease of L_{10} with distance corresponding to approximately 4.4 dB(A)/doubling distance.

4. EFFECTS OF NOISE BARRIERS

Much of the available information on shielding by a barrier relates to stationary point sources of sound. A theoretical solution for the semi-infinite barrier has long been available but restrictions arising from the necessary approximations limit its range of validity. An entirely empirical approach to the semi-infinite barrier has been adopted by Maekawa (13) which has led to a normalized curve giving shielding in excess of spherical spreading (6 dB decrease per doubling distance) as a function of $2d/\lambda$ where d is the difference in path length between the diffracted and the direct sound path, and where λ is the wavelength of the sound. This has been extended to the multiple-source situation required for road traffic and to typical vehicle-noise spectra. Basically the point-source shielding was applied to an array of spaced sources (the limiting case of a dense traffic stream) each having a typical octave-band vehicle-noise spectrum and results obtained for a range of barrier heights and source/receiver distances; using these data a second normalized curve could be derived which gives the shielding in excess of cylindrical spreading (3 dB decrease per doubling distance) in terms of the difference in path length between the diffracted and the direct sound wave in the plane perpendicular to the line source.

From this curve it is a relatively simple task to evaluate the shielding for any simple barrier configuration and a systematic series of prediction contours (isobars) for immediate application has been derived (8). These give the total effective reduction in noise level relative to that at the 10m reference distance. Each contour set can be used to predict for a range of road/barrier configurations and they cover many of the simpler situations encountered in practice (an example is shown in Fig. 3).

5. EFFECT OF DIFFERENT ROAD/HOUSING CONFIGURATIONS

Many complex urban situations arise which defy currently-available calculation procedures and a number of these have been investigated using a 30:1 scale model technique (6) to investigate the propagation of noise out to distances of 100m from the road. A single small noise source has been used, consisting of an air jet impinging on thin crossed vanes, producing a noise spectrum extending to 100 kHz which is subsequently shaped electronically to simulate a typical A-weighted vehicle noise spectrum. Road and other reflecting surfaces are simulated by sheet aluminium and typical grassland is simulated using Insulite fibreboard covered with a nylon sheet; houses and other buildings are made of plywood surfaced with hardboard. The source, representing a single motor vehicle, is traversed in a slot along the road whilst at some appropriate point the noise history of the single drive-by is recorded using a 1/4-inch condenser microphone with digital record to punched paper tape.

The technique has been validated by comparing model results with field data for ten different field sites. For the first phase of the work the summated energy for the model drive-by was used and it was found that this correlated well with the corresponding values of L_{10}; the rms prediction error is in the range 1 - 2.5 dB(A), depending on the site complexity. Recent developments include an on-line system in which the computer simulation and the model drive-by data are combined to give the time/level noise distribution at any point.

A report giving results for a wide-range of different configurations with a motorway in cut will be available shortly (6) and further site configurations are now being studied with practical road schemes very much in mind.

6. PREDICTION ACCURACY

The quantity of reliable field data available for evaluating prediction accuracy is surprisingly meagre and none of the previously published prediction techniques seem to give firm estimates of what accuracy can be expected from their use. It may be claimed for the prediction methods outlined in this paper that a number of cases are based upon, or have been confirmed by, direct measurement and that extension to other cases is largely based on established physical principles.

From comparisons of predicted values with levels measured adjacent to a number of different major-roads and motorways it has been established that the rms prediction error for the noise level at the reference distance is some 1.4 dB(A) so that, assuming a Gaussian error distribution, some 85% of measured values should fall within ±2 dB(A) of the corresponding predictions. A similar prediction accuracy has been found for open-ground propagation up to 100m from the kerb and for the noise level behind various noise barriers. With some of the more complex urban situations investigated using the model technique the rms prediction error increases to 2.5 dB(A) but even here 77% of results should fall within ±3 dB(A) of the predicted values and this accuracy should be adequate for most planning purposes. Moreover, this represents an estimate of the error which can be expected when comparing predictions with measured field data. It must be remembered, however, that a proportion of the error variance arises from the field measurements themselves and the mean prediction error when used for planning purposes should be better than the quoted values of overall prediction error. Furthermore, the relative accuracy when comparing different configurations for any given road should certainly be higher, for the error in predicting the basic noise level at the reference distance is not then incurred.

7. FURTHER WORK

It is not claimed that the prediction procedures outlined here give exhaustive coverage. Results are currently being obtained for finite-length barriers and barriers with apertures and it is planned to publish these as soon as they become available.

There are some situations such as roundabouts, intersections and gradients which have so far defied currently available calculation and simulation procedures and the modelling technique; the necessary data for these will have to be obtained by systematic field measurements.

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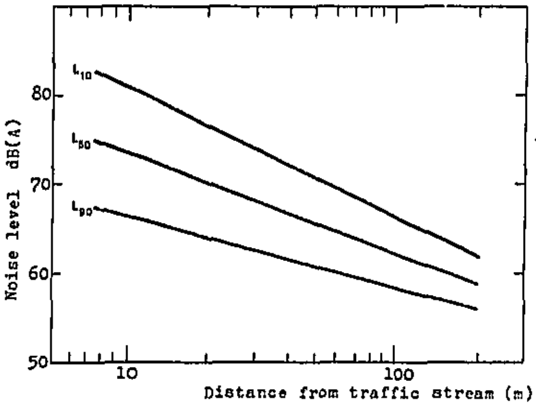

Figure 1
Noise level over typical open grassland.
2000 vehicles/h
80 km/h
20% heavy vehicles

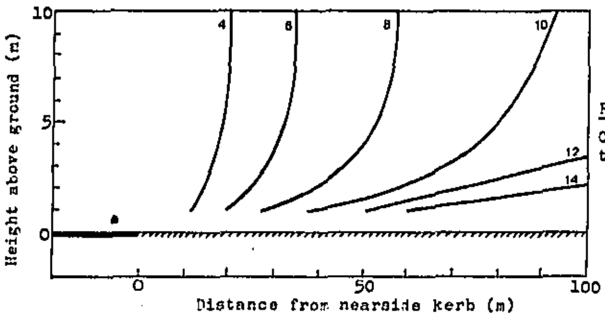

Figure 2
Contours for L_{10} over typical level grassland.

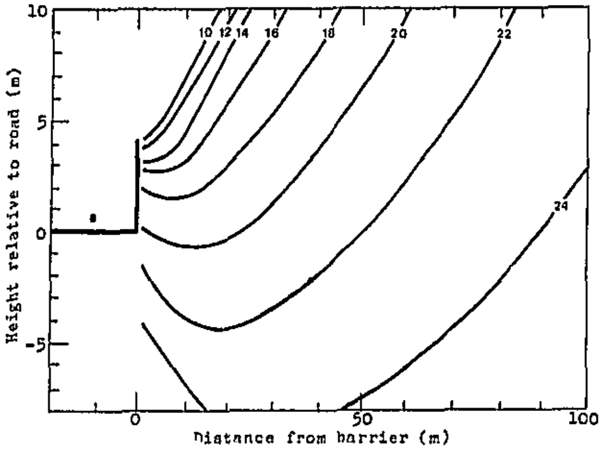

Figure 3
Contours for L_{10};
4m high barrier, 10m from traffic stream.

SELF DEFENSE AGAINST SURFACE TRANSPORTATION NOISE

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The noise from all types of motor vehicles on our highways throughout the country is undoubtedly the greatest noise nuisance in existence, because it is so widespread and affects so many people, distributed along the hundreds of thousands of miles of highways and turnpikes throughout the country. Every new highway or turnpike which is created or extended through a residential area creates a new corridor of noise nuisance, elevating the noise above that community's previous experience for a distance from 1/4 to 1 mile, depending upon the nature and experience of the community.

It is well and good to perform a noise impact study in each new situation and evaluate the expected noise exposure for the community. But what will be done about this new exposure, both by the Highway Commissions which choose the routes, design the highways, and contract for their construction, and what must the community do to protect itself beyond the degree to which the highway design can furnish desirable protection? Again we are faced with the inherent conflict between the desirable expansion of transportation, commerce and travel for the benefit of large segments of population versus the rights of another large segment of population living along these routes and being entitled by the constitution to live in peace and tranquility. The next few years will test this country's determination and ingenuity for resolving this conflict.

In contrast with the more sophisticated, if less pervasive noise nuisance from modern airports, in the case of the highway noise situation all the technical factors are known and the methods for resolution are ready and waiting. The major sources of noise are identified, and the techniques for substantial reduction are at hand, requiring only legal enforcement. Still greater reduction of the sources to even more manageable levels requires research which is well under way, with excellent prospects. A great deal can be done in the design of the highways to reduce the spreading of noise to the nearby neighboring community under most circumstances. The design of buildings for exclusion of the remaining noise is an acoustical technique which is thoroughly known. But how shall the burden be divided among these three potential means for controlling the excessive noise exposure? Since most of the vehicles which create the noise pass freely along their business from one state to another, it is clear that the guidelines for noise control must come from a federal agency, and we look hopefully to the Environmental Protection Agency for this function.

To view the noise exposure, we may look briefly at two extremes among the vehicles. A graph will show the noise spectra created at a distance of 50 feet by heavy Diesel driven trucks, which constitute from the order of 2% to 6% of the traffic which may be counted along our major interstate highways. The maximum spectrum is generated by a rather small majority of these trucks which are still permitted to operate with little or no mufflers. The lower spectrum is for a truck of the same capacity, but equipped with a high quality muffler, as indeed are many of those which leave the plants of major manufacturers. Intermediate are spectra for trucks which will meet the current California State noise limit of 88 dBA. We note that the noise reduction occurs mostly in the low frequency range, which is dominated by engine noise, while reduction at the high frequency range is limited by the noise of the tires on the highway surface. The range of noise on a statistical basis for trucks over 6000 pounds is shown on another graph, and the location of the current California limit is shown along with the proposed limit for 1975 and 1985. It is to be hoped that these planned reductions will in fact be enforced and become federal law as well. Another graph shows the range of spectra for passenger vehicles, showing again the current California State limit. The corresponding statistical graph is shown for the distribution of noise level among this class of vehicles, along with the proposed limits, requiring progressive reduction of the noise from passenger vehicles.

Our concern for the noise exposure of a given community or a particular building along a highway must focus, not on the individual passenger vehicle, but rather on the aggregate of vehicles over a period of a day or a week. A graph will show the rate of flow of vehicles past a particular point along a major California highway. There are morning and evening peak periods, and there are long minimum periods, particularly during the nighttime hours. Another graph will show the noise level emitted by this highway as a function of distance from the highway. Naturally, the farther away we move from the highway, the greater is the number of vehicles spread along the highway whose noise must be integrated as it floods into a particular distant location. Another graph will show the noise at a distant location as a function of the vehicle travel rate. It is interesting indeed that the noise level does not vary over a range which is commensurate with the variation in traffic rate. The reason is that, as the flow rate increases, the entire flow slows down, so that at high rates we are dealing with a large number of vehicles moving very slowly and therefore generating less noise, whereas at low flow rates we are dealing with fewer vehicles traveling at high speed and creating maximum noise.

In selecting criteria for tolerable noise from vehicular traffic, we have many conditions to consider. During the day we have noise flooding into school grounds, for example, and penetrating the buildings to create masking noise above which the verbal communications of teaching must be able to prevail. Similarly, in commercial buildings speech communication is again essential. In hospitals, for example, we must be concerned not only with necessary functions, but also with the well being of patients whose tolerance is already highly strained. In all these buildings, daytime conditions are of primary concern. However, in residences and, again, in hospitals, we must be concerned about nighttime conditions. In this case, the flow rate may have decreased greatly. We may be concerned with noise from several passenger vehicles, all in close enough range to contribute jointly to the exposure. However, when it comes to trucks, and these will probably always be the most vociferous sources, the passage rate will be reduced in most cases to the point where each vehicle stands out as a source isolated in time. Therefore, for all buildings where the sleeping environment is perhaps our greatest concern, the reasonable tolerance for time isolated noise exposures will be our guiding criterion, and the individual truck passage will be our sound source of greatest concern.

A particular and perhaps most extreme example illustrating the problem will be discussed.

Considering now more generally, what shall we take for residual noise criteria in various buildings, and how will this degree of noise control be achieved? The next graph shows proposed values for maximum noise exposure within various types of buildings for daytime and nighttime occupancy. In residences and hospitals, of course, the nighttime requirement dominates, whereas in schools, churches and commercial buildings the daytime requirement will dictate the required construction.

There are two broadly different configurations which dictate the feasible responsibility for noise control and the extent to which it must be achieved. First, in the case where both the noise source and the noise receiver are on the surface or at the same elevation, a great deal can be accomplished within the highway design by the construction of noise barriers along the perimeter of the highway. The height of the required barriers will be dictated by the noise at the source, the proximity and type of building occupancy along the highway, and therefore the interior residual noise criterion. Others on these programs will have discussed the effectiveness of perimeter noise barriers. There is, of course, in addition to cost, reasonable concern about the architectural or landscaping character of such noise barriers. A brief survey and estimate, however, would indicate that the cost of continuous noise barriers may in general be the order of 5% of the cost of the highway.

Under such circumstances, i.e. when the source and the receiver are on the same level, it is obvious that the distance of the building from the highway will be a dominant factor. An obvious approach to controlling the noise nuisance is for municipalities or perhaps even for state governments to determine that zoning regulations will be formulated to prevent noise sensitive buildings or functions from occupying a buffer zone for a prescribed distance from the highway. This is obviously a method of easing the problem which is available to public organization, i.e. community government. But a fascinating legal problem develops, which will be discussed.

A second configuration, however, is much more serious in terms of the implications for an adjoining building, namely when the receiver of the sound is at a much higher elevation than the surface of the highway. This will obviously always be the case for tall buildings, where either multiple residences or commercial buildings of multi-story construction may either exist or be projected along a highway. In such cases, barrier walls are not practical because they would have to be too high. For limited lengths enclosure of at least the truck lanes should be considered. In many cases, the task must be recognized and accepted by the owners of adjoining buildings rather than by the highway. Generally speaking, the greatest additional cost will be for proper soundproofing of windows which, of course, generally implies forced ventilation for the buildings. It is clear that the extent and cost of this sound isolation will be governed first and foremost by the maximum noise sources along the highway, stressing again the importance of reducing the noise of large trucks. It is especially significant that when one looks at the required noise reduction spectrum, he finds that the most serious burden of noise reduction falls in the lower frequency range, rarely higher than 1000 cps. We note, therefore, that as we stressed before, it is in the reduction of the lower frequency portion of the truck spectrum, controlled by engine exhaust and mufflers where the improvement is most urgently needed. On the other hand, the higher frequency noise from tire whine is generally adequately reduced by most shell construction, including windows. Therefore, the tire noise is more of concern for outdoor functions.

Let us now return to the hypothetical situation which is most common, and therefore most important, i.e. major highway passing through the residential community. Before the zoning group and the land manipulators go too far with their machinations, let us consider what architectural and acoustical ingenuity might provide to find a solution to this problem. A drawing will illustrate the approach that we recommend to the land developer when he wishes to create a large multiple-occupancy residential community adjacent to the highway. We maintain that it is possible for the development in effect to turn its back on the noise problem. The method will be described in detail.

The plan of the development is a C-shaped form. Along the length of the highway, the multi-story building must be continuous for the length of the development, and the ends must swing around in order to create a shielded open area or park within the enfolding arms. If the building may not be continuous, or the extent is uneconomically expensive, then portions which are not needed for occupancy may be filled in with an integrated architectural noise barrier. The important feature of the development, acoustically of course, is for the noise barrier to shield the entire enclosed area from the highway noise. The accompanying graph shows the degree of noise reduction which may be attained by a five story, i.e. 50 feet high, development in C-shape plan. The entire length of the face of the building toward the highway must, of course, be of soundproof construction. Any windows facing this direction should, of course, be sealed and of adequate acoustical design, which for current noise sources, means that they must be good double windows. The walls need not be masonry, because multi-layer lighter weight acoustical construction can furnish adequate noise reduction. Within the interior park area, the shell construction may be completely normal, with openable windows, patios, porches, playgrounds, swimming pools, and all modes of recreation proceeding without awareness of the highway beyond. If the demand for land is such that these developments may be interlinked along extensive lengths of highway, the same principle is valid. It must be recognized, however, that large gaps between constructions will permit passage of the sound to the area beyond. This in itself would not be of concern if it were not for lower and interspersed perhaps individual family dwellings in the area beyond. If the taller multiple occupancy development could extend continuously, then the entire community beyond would be greatly protected. If, however, there are gaps, then the validity of the C-shaped form, even for the protection of the particular development, is threatened, because the noise exposure of the walls beyond, which face the highway, also serve to reflect that noise back into the open unshielded portion of the C-shaped development. This is obviously the reason why the spaces which are visually shielded between homes may not be acoustically shielded, because the sound which diffracts over one building is reflected from the building beyond. Therefore, although the sound level drops off more rapidly as one progresses away from a highway through a residential community than it does in open space, it is nevertheless the backward reflection from hard surfaces which reduces the potential shielding benefits. If, however, in a planned community these reflecting walls can be constructed with substantial sound absorbing material on their surfaces, this backward reflecting property would be greatly controlled. Results of model experients on this subject are shown on the graph.

It is apparent, then, that while it may not be necessary to exclude residential construction or, for that matter, other noise sensitive buildings such as schools, churches and hospitals, from the large areas immediately adjacent to highways, it is imperative that these buildings be designed with thorough recognition of the acoustical problem. It is also clear that there is a great deal by way of community planning, as well as the efforts of the individual land developer or individual home owner, that can be done to assure adequate living conditions and other intended functions for the buildings which are threatened by an existing or planned highway development. It seems clear, however, that, through ignorance alone, such planning is not likely to occur without guiding information and legislative regulation from government. Since these problems are universal throughout the country and the highway network is, of course, interstate as well as the traffic upon it, it seems only reasonable that such regulation and construction should come from the federal government. We all, of course, look toward the youthful Environmental Protection Agency for this function. We must wish for them not only the greatest competence, but also superhuman courage, particularly when we realize that opposing this unique Environmental Protection Agency there are also many agencies both inside and outside the government which also have these letters, EPA, after their pedigree. Only for those agencies, the initials stand for Environmental Pollution Agency. We, therefore, look with realistic apprehension, but the greatest hope, to the Environmental Protection Agency for peaceful sound in our time.

EUROPEAN EFFORTS TO REDUCE THE IMPACT OF TRAFFIC NOISE

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Back in 1963, the Wilson Committee, set up at the request of the British Government to propose solutions to all noise problems, reported that in large towns "Road traffic is the predominant source of annoyance and no other single noise is of comparable importance". All the noise measurements made in Europe confirm this: road traffic is almost invariably the main offender in built-up areas. It is true that aircraft noise may be worse than traffic noise, but unlike in the United States, only a relatively limited number of areas are affected. What is more, traffic noise is continuing to increase both in time and in space. The period of quiet at night is becoming shorter and shorter, as the measurements taken by the Greater London Council have shown. As regards space, the residential suburbs with their inadequate public transport services are suffering increasingly dense local traffic. This increase in road traffic noise in Europe reflects the growth of car and motor-cycle ownership and very fast urban development. Public opinion has reacted sharply to this nuisance. In all the surveys made in France, Norway, Sweden, Switzerland and Great Britain, town-dwellers cite traffic noise as the principal nuisance.

In London and Paris there have been demonstrations against new urban motorways, or freeways. In the face of this pressure from the public, noise abatement officials have been appointed, legislation has been enacted, noise abatement bodies have been set up, research has been initiated with the object of reducing noise at the source, and local measures have been taken to combat noise.

Some countries and some city authorities are more active than others in the matter of traffic noise abatement, and some schemes have made more progress than others. I shall therefore try to give a quick assessment of the various kinds of action being taken, dealing firstly with the problem of enforcing existing noise regulations and secondly with local measures taken to reduce the impact of traffic noise on the public.

THE ENFORCEMENT OF NOISE RULES

As the Common Market countries have now adopted common maximum permitted noise levels for motor vehicles and most of the other European countries have introduced their own regulations, it is interesting to see how these rules are applied. For new vehicles leaving the factory, there is an official acceptance and certification scheme; in other words, before vehicles of a given model are launched on the market, a typical vehicle is measured for noise emission by the ISO method and a certificate of conformity is issued. The noise limits set so far, however, are not sufficiently low since 95 to 100 per cent of vehicles sail through the acceptance tests.

What is more, with the ISO procedure which has been followed for fixing the permitted levels, it is impossible to carry out checks on the road. Only one country, Switzerland, has laid down maximum noise levels for stationary vehicles which allow easy police checks.

A number of countries, namely France, Switzerland, Norway, Germany and Spain, already have or are building vehicle-testing centres originally intended for compulsory periodical inspections from the safety standpoint. These inspections already include, or will include, a check for excessive noise and examination of the silencing system. In France and in Switzerland, for example, there are also special stations for measuring noise from vehicles which have been found "by ear" alone to be making too much noise on the road. On a country-wide basis, however, there are too few noise measuring stations and too few spot checks, due to shortage of staff and equipment.

Hence, the most interesting action is very often that taken at local level by certain particularly active and well-run cities. An example is the Noise Brigade formed 13 years ago by the police at Lausanne in Switzerland. This squad consists of five police officers and, while its task is to reduce and control noise from all sources such as traffic, building sites, industrial plants, bars, dance halls and so on, it has mainly been concerned with traffic noise. One of its priorities is to ensure quiet at night-time and it therefore operates at a number of points in the city from 10 p.m. to 6 a.m. The entire police force is of course required to assist the noise squad both night and day. If a squad member considers subjectively that a vehicle is creating too much noise he signals the driver to stop. No instruments are used at this stage and experience has shown that with practice a police officer rapidly acquires a sensitive ear and can decide with remarkable accuracy whether the noise of a vehicle exceeds the accepted limit. The vehicle is rapidly inspected for road-worthiness and should any of its components be defective or fail to conform to the laid-down standards, the vehicle is immediately towed to a garage for repair. The owner then has to bring it to the squad's testing station for approval before he is allowed to put it on the road again. If the rapid inspection shows only minor defects, the registration papers are withdrawn and the driver is given a temporary permit until he has had the defects put right. He can then recover his registration papers. If he disputes the summary diagnosis he is invited to take his vehicle to the testing station for thorough inspection.

In doubtful cases where the squad cannot find any obvious fault that could account for the excessive noise, the driver has to leave his vehicle where it is or, exceptionally, he is allowed to continue his journey at reduced speed without making too much noise. The registration papers are withdrawn and the driver is later called to the testing station for his vehicle to be checked.

The worst offenders picked up by the noise squad are teenage moped riders who tamper with the air intake, remove or punch holes in the silencer system, or modify the carburettor to make their machines noisier.

In France this year, on the initiative of the official appointed by the Government in 1971 to be in charge of noise abatement - they call him M. Silence - 37 noise squads have been formed throughout the country. Staff and equipment costs are borne by the Ministry of the Environment and the Ministry of the Interior.

I had my doubts at first as to whether they really existed but their existence was recently brought home to me in an amusing way. A friend of mine is Secretary of the Noise Committee at the Ministry for the Environment. Coming to see me one day, when he started up his car he found that his muffler had blown. But he started out nonetheless, only to be stopped after a mile or so in Paris by two policement whose job was purely and simply to stop vehicles making too much noise. The fact that he was in the Ministry of the Environment saved him from paying a fine but he had to leave his car there and then at the nearest garage. It is clear from this incident that a policeman does not need a noise level meter to stop a noisy vehicle any more than he needs a camera to prove that a car has passed a traffic light at red. But what is desirable is that police officers assigned to noise control should not at the same time be on traffic control duty; if the two duties are combined, noise control is inevitably neglected. Special squads must be formed specifically for noise abatement work.

Since Europe is so highly urbanised and vehicles move more and more freely from one country to another, there is a need for uniformity as regards not only the noise abatement regulations but also the enforcement measures. Switzerland provides a striking example. There are no more than one and a half million vehicles registered there and yet more than 30 million run over the country's roads each year, at some time or another. How can vehicle noise be controlled effectively if only Swiss-registered vehicles are subjected to roadside checks? The control measures cannot be applied to all vehicles since this would affect the country's tourist trade and conflict with the principle of free circulation, quite apart from the absence of sufficient personnel. This example shows the need for all countries in Europe to carry out sufficiently effective controls to prevent what might be called "trans-frontier noise pollution", which is particularly noticeable in the countries that take steps to restrict the noise made by their own vehicles but are unable to take effective steps with regard to foreign-registered vehicles in transit.

After this brief look at how maximum permitted noise levels are applied in Europe, let us turn to other measures that have been taken to reduce the impact of traffic noise on the population, that is, city planning and traffic control measures.

MEASURES DESIGNED TO PROVIDE PROTECTION AGAINST TRAFFIC NOISE

Noise abatement measures associated with city planning and traffic control are many and varied, ranging from acoustic barriers to vehicle-free zones. They include night-time ban on certain vehicles, provision of quieter public transport, designation of no-building areas, soundproofing of housing, and so on.

Acoustic screens have been constructed in Berlin and near London and Paris. Outside London, a plastics barrier 300 metres long and 2.60 metres high has been built along the motorway from the city to London Airport as a protection for private houses near the highway. The mean reduction in noise achieved is around 10 dBA. The cost per linear metre was about $200, so that the whole project cost $60,000.

Near Paris, an experimental concrete screen 300 metres long and 3.50 metres high has been built, again along the motorway carrying traffic to the airport and at the same cost as the one built near London. In the Paris area, unlike London, the housing in question is in the form of multi-storey apartment blocks and the motorway has 12 lanes (compared with London's six). This means that only the apartments at ground level and one storey up are protected. What is more, the barrier is not long enough; this causes reverberation and some apartments are affected by an extremely unpleasant low-frequency noise.

The Paris ring motorway which is nearing completion also has some interesting features. A number of sections consist of tunnels or cuttings, particularly to the west of the city. There is even, one two hundred metre section in a cutting where half the carriageway has been covered over to prevent noise reaching a new embassy to be built nearby. Unfortunately, no cost figures are available.

In some new French towns - Cergy-Pontoise about fifteen miles outside Paris is a case in point - a system of zoning is applied. Alongside motorways no housing is allowed on a belt 30 metres wide from the edge of the carriageway and apartments between thirty and eighty metres from the edge of the carriageway have to be soundproofed. Along other roads housing construction is not permitted within thirteen metres from the edge of the carriageway.

But the most noteworthy examples of protection against traffic noise in new towns are in Great Britain. In Stevenage, for example - a new town of some 100,000 inhabitants - road traffic, cyclists and pedestrians are entirely segregated. Transit traffic has to take a circular route and traffic to the town itself ends up in parking lots outside the housing estates. Cycle and footpaths are the only routes passing through the residential areas. Thus, apart from increased safety and a better environment for play and for walking, traffic noise is considerably reduced. Motor traffic is being prohibited on an increasing scale in some of the older quarters of towns in Europe. In Rouen in France, two streets in the centre have been closed to vehicles and a broad pedestrian precinct is planned. Vehicle-free zones have been set up in Norwich in Great Britain, Copenhagen in Denmark, Essen in Germany, Berne in Switzerland, Vienna in Austria and elsewhere. Other towns, including London, will be introducing the same restrictions. The reduction of noise is not the only objective or even the main one. The principal reason for closing entire streets to traffic is usually to restore the busy, older streets to strollers and shoppers. Nonetheless, a marginal but far from insignificant side-effect is the lowering of ambient noise in these areas, where human sounds are heard again. There are even cases where such restrictions have been introduced purely for reasons of noise, particularly as regards night-time traffic. In Switzerland, a dozen streets in Zurich are closed to mopeds and motor cycles between ten at night and seven in the morning in order to protect the local hospitals and residential areas from noise, and since 1959 vehicles passing through Lausanne have had to go round the city during night-time. While on the subject of Switzerland, two other points are worth mentioning: firstly, heavy vehicles are not allowed to run anywhere in the country during the night; the only exemptions are buses, fire engines and trucks carrying certain perishable goods. The second point is that at the time of a referendum in Berne, the Federal capital of Switzerland, the public decided not to spend money on buses but to extend the trolley-bus routes which cause less pollution and noise. The outcome of this referendum has had a salutary effect on other Swiss cities where transport undertakings are now thinking twice before buying buses and prefer to increase the number of trolley-buses when they have the necessary infrastructure, despite the fact that trolley-buses are more costly than motor buses. Berne thus provides a perfect example of a combination of traffic noise abatement - a ban on trucks, vehicle-free zones and near-silent public transport.

At a last halt in this whistle-stop tour of measures taken in Europe to reduce the impact of traffic noise, the recent scheme introduced by the Ministry of Equipment and Housing in France is worth mentioning. It came into effect this year and features an "acoustic approval mark" for housing in the public sector. The approval mark is to be granted only for housing that meets specific standards of internal and external soundproofing and it entitles the builders to an additional loan of up to 6 per cent of the main sum advanced. It is hoped that this will lead to a general improvement in the acoustic insulation of new buildings, an area which leaves a lot to be desired at the moment, since the private sector will certainly find itself obliged to provide the same standards as those of government-sponsored housing.

As can be seen, the efforts to reduce the impact of traffic noise in Europe are in various directions and it is too early to give any final judgment on a number of schemes which are still in their infancy. Acoustic insulation for housing, quieter public transport, and traffic bans and diversions will no doubt come to be used more and more to protect the public from noise. But large-scale achievements are still a long way off. The growing number of vehicles and the extensive housing development add to the problem whereas those solutions that have been found are limited in scope.

Undoubtedly the efforts being made in certain countries which do not have any influence on the reduction of vehicle noise at the source - Switzerland being a case in point - merit special attention since they could well serve as an example to others. But there is no doubt that many of the measures I have referred to are no more than temporary palliatives.

Steps need to be taken to reduce noise at the source, since this is the only real way to achieve any general improvement. In this connection it should be noted that research is under way in the United States, in Great Britain, Germany and France on ways of reducing the noise emitted by vehicles, but considerable time will elapse before it produces significant results - and the lead time from the laboratories to the mass production factories is even longer.

Meanwhile, all we can do is to improve the implementation of the existing norms, and seize every opportunity offered by city planning and traffic control measures.

TRAFFIC NOISE ABATEMENT RESPONSIBILITIES OF STATE HIGHWAY DEPARTMENTS

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Traffic noise pollution is not a new phenomenom; it has been around for quite a long while. Excessive traffic noise is known to have existed for more than two thousand years, when, during the reign of Julius Caesar, chariot traffic produced offensive noise levels in the streets of Rome. Annoyed at the racket, the Emperor decided to institute his own traffic noise abatement program by banishing chariots from the streets during the night hours. Thus, Caesar probably became the first expert in traffic noise abatement.

Unfortunately, state highway departments do not have the expertise in the noise control business that Caesar had. Kentucky just started in September of 1971 with one full-time engineer. This is about average for most states, although some, like California, have anticipated federal regulation and have worked in the area for a number of years.

What actually is the federal regulation that has caused state highway departments to add engineers whose responsibilities are the solutions to noise problems? And have these expenditures in personnel and also those in equipment, support and training proven to have been successful investments of the tax dollars? The answer to the second question is an unqualified yes, and this will become even more evident as time passes. The answer to the first question is given by Section 3 of Policy and Procedure Memorandum 90-2, Noise Standards, as issued by the Federal Highway Administration, which states in part: "In order to be eligible for Federal-Aid participation, all applicable projects shall include noise abatement measures to obtain the design noise levels in the standards unless exceptions have been approved as provided herein." In other words, if a state traffic noise abatement engineer does an unsatisfactory job, his state may lose a large amount of federal money.

The Federal-Aid Highway Act of 1970 amended Section 109 of Title 23 of the United States Code to include the following:

> (h) Not later than July 1, 1972 the Secretary (of Transportation), after consultation with appropriate Federal and State officials shall submit to Congress, and not later than 90 days after such submission, promulgate guidelines designed to assure that possible adverse economic, social, and environmental effects relating to any proposed project on any Federal-Aid system have been fully considered in developing such project, and that the final decisions on the project

are made in the best overall interest, taking into consideration the need for fast, safe, and efficient transportation, public services, and the costs of eliminating or minimizing such adverse effects and the following:

(1) air, noise, and water pollution (among other things)

Section 109 (i) of this same title was also added to state that the standards for highway noise levels must be compatible with different land uses, and that no federal-aid project which did not receive location phase approval before July 1 of this year must comply with the standards in order to be eligible for approval.

In the development of PPM 90-2, which is the response to the Federal-Aid Highway Act of 1970, the Federal Highway Administration did its best to follow the letter of the law. For example, the FHWA circulated a draft of the PPM, and requested that all states send in comments. In addition, the American Association of State Highway Officials' Task Force on Environmental Design made comments. This Task Force represents a key group of men, since it is made up largely of the directors of the highway design branches of nine state highway departments. After compiling all the comments received, the FHWA made a sincere effort to incorporate needed changes into PPM 90-2. The result is that PPM 90-2 represents a workable yet effective method for reducing traffic noise resulting from improper highway design.

Precisely how are the state highway departments required to respond to the Noise Standards? Previous federal regulations have required the submittal of the following: The location or corridor study report, and the design study report, as required by PPM 20-8; and the pre-draft, draft, and final environmental impact statements, as required by PPM 90-1. PPM 90-2 requires that sections on noise be included in these other reports. At some future date, similar sections will be required for other environmental factors, including air pollution. If a state highway department feels that it is necessary to apply for an exception to the Noise Standards, it does so in the environmental impact statement. Approval of the E.I.S. signifies approval of the exception, thus eliminating the need for submitting a separate request.

The Federal Highway Administration is also requiring that each state highway department develop an interdisciplinary team, of staff, to handle environmental problems. This process, which is called the Environmental Action Plan, must be fully implemented and operational by October 1, 1973. This interdisciplinary team will have at least one full-time noise abatement engineer who will work with all phases and divisions of the highway program. In addition, most state highway departments are attempting to train their planning and design engineers in the basics of traffic noise prediction and abatement procedures. By having each engineer familiar with the techniques used in traffic noise control, the interdisciplinary team member whose speciality is noise abatement engineering will be free to handle the more difficult problems.

One problem that the state highway department has been faced with is where to find a noise abatement engineer. Very few engineering schools in the United States offer much course work in environmental noise control, and many of those who do offer such programs do so only at the graduate level. Therefore, most state highway departments have resorted to the retraining of certain promising young civil or highway engineers.

While PPM 90-2 does not list any requirements in regard to type and amount of noise monitoring equipment needed, it is necessary for the state highway department to have something. The PPM states that a comparison must be made between the predicted traffic noise levels and the existing ambient noise levels. The only accurate way to determine existing ambient noise levels is to actually go into the field and measure them. Thus the need for equipment exists.

The minimum equipment required is a sound level meter and a calibrator. The Kentucky Department of Highways has also purchased a graphic level recorder, which allows for better interpretation of the data gathered. This set of equipment costs approximately three thousand dollars, including support equipment. The Kentucky Department of Highways believes that the purchase of two more sets of equipment is all that will be required to monitor all projects in the state. Therefore, the total initial outlay for Kentucky in terms of noise measuring and recording equipment will be less than ten thousand dollars.

As mentioned earlier, each state must also make predictions on future traffic noise levels that will exist in the design year of the project. The Federal Highway Administration has approved several methods for making these predictions. One of these is National Cooperative Highway Research Program Report 117, "Highway Noise - A Design Guide for Highway Engineers", a project funded by the Highway Research Board and performed by Bolt, Beranek and Newman, Inc. The Kentucky Department of Highways has adapted this design guide to a computer program, and has also used it as the basis for a time-sharing computer program that generates noise level contours. Kentucky is also in the process of comparing results obtained from Report 117 with results from actual field measurements in order to add validation to the design guide.

The next question to consider is, what will be the short and long range results of the implementation of PPM 90-2. This is very important when one considers the devastating effects that traffic noise has on the modern American. For example, the people of Seattle, Washington nearly rejected a major urban expressway, mainly because of excessive traffic noise. Only after the Washington Department of Highways redesigned the expressway to include revolutionary noise abatement procedures did the public allow it to be built.[1]

The one section of PPM 90-2 that more than any other will determine success or failure is the appendix containing the actual numerical limits themselves. All levels given in the PPM are in terms of L10, which is the noise level exceeded ten percent of the time. The most commonly used limit is seventy dBA, which applies to residences, schools, hospitals, hotels, and the like. The first draft of PPM 90-2 had this limit as sixty dBA. It did not take long for the states to realize that this was entirely too low, since most traffic noise predictions seemed to fall in sixty to seventy dBA range. As a result of comments received from the state highway departments and the ASSHO Task Force on Environmental Design, the Federal Highway Administration raised this limit up to seventy dBA. This provides a workable level that state highway departments are able to incorporate into their projects without causing any unnecessary delays. On the other hand, an L_{10} of seventy dBA is strict enough to eliminate many obvious problem areas.

It is, of course, too early to tell if PPM 90-2 will make significant contributions in traffic noise abatement. One can only hope it will, and he can also be encouraged by the positive response received thus far by the state highway departments. Contrary to what may be popular belief, most highway administrators realize their responsibility to the protection of the environment, and have cooperated with the Federal Highway Administration in the development of the noise standards.

[1]Bulley, W. A. (Washington Department of Highways), "Design for Noise Control", presented at the national meeting of A.A.S.H.O., Miami, Florida, December 7, 1971.

NOISE CONTROL FOR INTERSTATE 205 IN PORTLAND, OREGON

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The introduction of a major highway into an urban residential area may cause severe environmental impact unless careful analysis and planning is undertaken early in the design process. Recently, Bolt Beranek and Newman Inc. performed an evaluation for the Oregon State Highway Division of the proposed configuration of I-205 in East Portland. The analysis was designed to pinpoint areas of impact along the freeway route, and to recommend noise control measures for reducing these impact areas. In this paper I will describe the techniques developed for defining noise impact areas and the considerations involved in determining effective noise control measures.

Of primary importance in the task of defining noise impact is the capability of predicting noise levels that will result from future highway traffic. Towards this end we developed a computer program which calculates L50 noise levels for freely flowing traffic. This computer program follows the basic principals, and incorporates the basic data of the Highway Noise Design Guide[1] published by the Highway Research Board. The program takes a section of highway and divides it up into point sources, with a power level defined in each source for car and truck noise. The noise level heard at a location near the highway is determined by summation of the contributions from all the point sources. If a barrier is located between the observer and a particular point source, the contribution of that particular point source is diminished by an amount based on the barrier attenuation curve of Maekawa[2], as modified in the Design Guide. A barrier may result from the natural terrain of the area, structures located between the highway and observer, or may be caused by shielding due to the freeway itself.

Output of the computer program may be in a graphical format. The first figure illustrates a sample plot produced by the computer. In this figure, the representation of the highway as a set of point sources, indicated by stars, can readily be seen. Barriers are indicated with a solid line, while rows of houses are indicated with dashed lines. Noise levels resulting from traffic on the highway are depicted by various symbols.

From a computer plot such as this, it is easy to recognize areas along the freeway route which will be exposed to high levels of noise.

The determination of noise impact is based on two considerations: whether the expected noise levels exceed criteria levels, or whether the noise levels exceed ambient levels already existing in the community. Of course, for different land uses, different criteria would apply. In this study we defined noise criteria in terms of L50 and L10 noise levels for day and nighttime periods, based primarily on speech communication and sleep requirements, as follows:

		Day L50	Night L50	Day L10	Night L10
Residences	Inside	45	40	51	46
	Outside	50	45	56	51
Schools	Inside	40	40	46	46
	Outside	55		61	

If expected noise levels from highway traffic exceeded these design criteria by up to 5 dB, a condition of "minor impact" would exist. There is "no impact" when the criterion level is not exceeded, and "significant impact" if the criterion is exceeded by more than 5 dB.

It has been found that people generally tend to react to changes in their environment; the greater the change the more severe the reaction. Thus, one must also consider the increase in ambient noise levels that will result with the new highway, even if the increased levels do not exceed the design criterion. Generally, exceeding the ambient level by 5 to 15 dB will result in "minor impact". "Significant impact" will result if the ambient level is exceeded by more than 15 dB.

As an aid in evaluating the noise impact of new highway noise on an area, the various impact categories can be related to judgments of the acceptability of the noise environment, and to the annoyance or dissatisfaction it may cause:

NO IMPACT - The highway noise environment generally is not expected to cause annoyance or dissatisfaction.

MINOR IMPACT - The highway noise environment is expected to be acceptable to most individuals, but there may be some annoyance or dissatisfaction.

SIGNIFICANT IMPACT - The highway noise environment may be judged as unacceptable by many with considerable annoyance and dissatisfaction.

When analysis indicates that a noise impact will result from the freeway traffic, there are a variety of noise control measures which may be used to alleviate the potential noise problem. Depression or elevation of the highway, relocation of a ramp, or even realignment of main lanes of the highway are all measures which may be considered if the noise control analysis takes place early enough in the design process. All too often, however, the only effective means of noise control remaining to a highway designer, because the roadway alignment and elevation has been permanently set, is the design and construction of noise barriers along portions of the highway.

In some situations the cost of noise control is totally out of proportion to its benefits. An equally important consideration is that of visual impact; there is a point where there may be more objection by a community to a high wall for noise control for example, than there would be to the noise exposure that would result if the wall were not there. It should be recognized, therefore, that it is not always feasible to decrease traffic noise to levels which would be satisfactory to all residents and compatible with all land use.

To illustrate the noise analysis procedures used in this study, let's consider a portion of I-205 that runs through the community of Lents. Just north of this area, I-205 intersects with another major highway. In order to minimize the amount of weaving on the freeway, while at the same time providing access to I-205 for the Lents community, a braided ramp configuration in conjunction with collector-distributor lanes was designed for both sides of the freeway (see Fig. 2). Although the freeway itself is slightly depressed in this immediate area, the braided ramp structure utilizes two lanes elevated above grade, with one 12 feet above grade at the highest point. This ramp configuration is located immediately adjacent to a school. Further complicating the situation is the terrain, which drops off considerably to the south of the Lents area, while the highway remains at basically the same elevation.

Two alternative ramp configurations were developed by the Highway Division in an attempt to reduce the noise levels from this structure. One alternative called for the relocation of the ramp structure to an area south of the school; the basic ramp configuration would be unchanged. The second alternative involved placing all lanes at or below grade in the vicinity of the school with the ramps that were originally elevated, now depressed about 28 feet below grade at the lowest point.

Our analysis indicated that regardless of the braided ramp design adopted, noise control barriers would still be required in order to minimize the impact of the highway on the Lents area, and to achieve an acceptable noise environment at the school. Because of the high elevated ramp in the initial design, the needed sound barriers would have to be impracticably high to be effective. The first alternative, involving relocation of the ramp structure, was ruled out because the cost would be prohibitively high. We, therefore, recommended that the final design, in which none of the braided ramps rose above grade, be adopted.

With this design consideration as a basis, we developed noise contours for this area using the computer program described above. The next figure shows these contours, which depict L50 noise levels for peak morning traffic in 1990.

In order to compare these predicted noise levels with current ambient levels, field noise measurements were obtained at three locations in this community. At each location, 10 minute noise samples were recorded on magnetic tape once an hour for 24 hours. The tapes were analyzed by playback through an A-weighting network into a statistical distribution analyzer. From the distribution, it was possible to obtain the various percentile levels such as the L10, L50 and the L90 levels exceeded within each hour, and over the course of the day. Indicated on the figure are the L50 noise levels at three locations, computed for day and night time periods.

Note that the noise levels do not vary greatly from position to position. We have found this to be true in many such communities, where measurement locations are away from the influence of heavily traveled streets.

Comparison of the predicted noise level contours and the current ambient levels, and comparison of the predicted noise levels with the outside daytime criterion level for residences of 50 dBA and for schools of 55 dBA, indicate the need for noise control measures along this section of the freeway.

Our recommendations involved a series of barriers along the freeway edge, indicated in the next figure. The center barrier is 15 feet high, and may be a wall, earth berm, or combination thereof. It may well be built as a vertical wall along the boundary of the school, providing a backboard for school sports activities. The remaining three barriers are each designed to be 10 feet high.

L50 noise contours, reflecting the use of the noise control barriers, are indicated in the figure. The overall reduction in noise level as compared with the contours of the previous figure is 3 to 8 dB, depending upon the location.

From knowledge of the variation in traffic flow over the course of the day, the change in projected noise levels with time of day was determined. Comparison of daytime and nighttime L50 and L10 noise levels for peak and nonpeak traffic flows with day and nighttime design criteria for indoor and outside activities, and with existing ambient noise levels during the day and night periods, define the various noise impact areas for this community. The next figure depicts the various impact areas during the day, categorized into inside and outside activities. Nighttime impact areas can be depicted in a similar manner. However, since the difference between daytime and nighttime highway noise levels is generally greater than the difference between the noise criteria for the corresponding time periods, the nighttime impact areas are usually smaller than the daytime periods.

Depiction of noise impact areas in a graphical manner such as this is a valuable tool to the highway designer, and can also be an important part of a highway environmental impact statement.

Based on the design criterion listed above, there would be "no impact" from the freeway noise outside of Lents school. In order to determine the impact inside the building, noise measurements were made of the outside-to-inside noise reduction of the structure. It was found that with windows open, because of the large window area facing the highway, one could expect only a 10 dB reduction of highway noise from outside to inside. On the basis of this, the "minor impact" classification applies inside the classroom with the windows open. With the windows closed, there would be "no impact" from the freeway.

While the noise control barriers do not completely eliminate the noise impact in this community, they do reduce the "minor" and "significant impact" areas to a minimum without degradation of the appearance of the neighborhood. Although higher or larger barriers might further reduce the noise impact, the visual impact of these barriers would be severe.

One final note is in order. Several state and city governments are setting limits to control the noise produced by motor vehicles. Noise limits specifying the maximum allowable noise levels of individual vehicles already exist in a few areas such as California and Hawaii, and many other states and cities are considering adoption of similar ordinances. Several of the present codes call for more stringent noise requirements from year to year. These trends in noise legislation, caused by increasing concern about motor vehicle noise, imply that it would be reasonable to expect significant decreases in the noise output of motor vehicles over the next several years. Although the degree of noise reduction, as one may expect, is extremely difficult to predict, it would not be unreasonable to expect that by 1990, trucks may be 10 dBA quieter than they are today and cars 6 dBA quieter. This would result in a highway noise level 6 to 9 dBA lower than the levels predicted based on 1972 vehicle noise characteristics. On this basis, projected noise impact areas would be significantly reduced in size.

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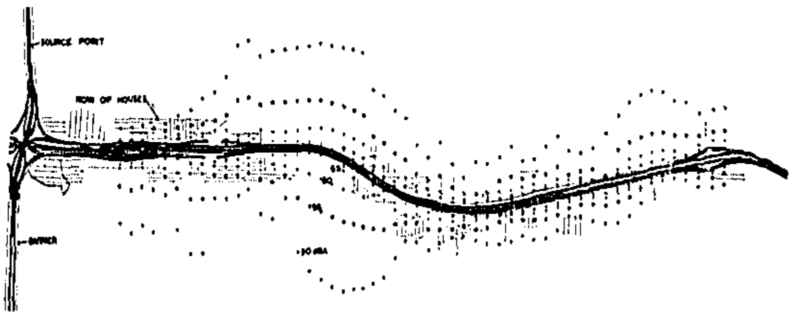

FIGURE 1. SAMPLE COMPUTER PLOT OF I-205 AND L_{50} NOISE LEVELS

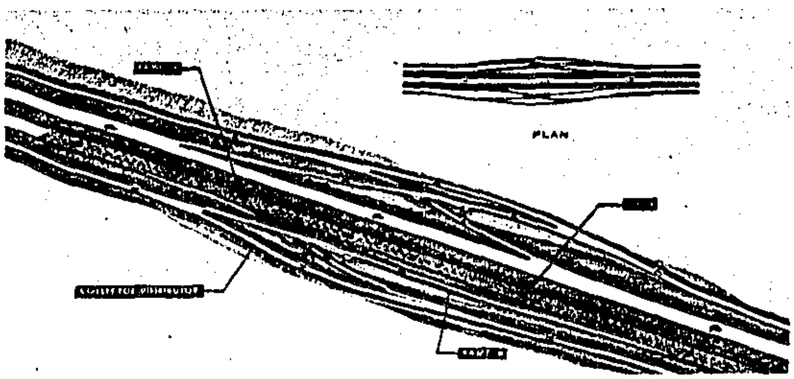

FIGURE 2. TYPICAL LAYOUT BRAIDED RAMPS

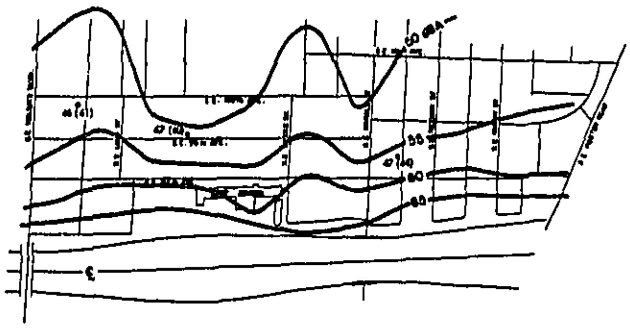

FIGURE 3. PROJECTED L_{50} NOISE CONTOURS FOR LENTS FOR 1990 PEAK MORNING TRAFFIC ON I-205, WITHOUT NOISE CONTROL BARRIERS

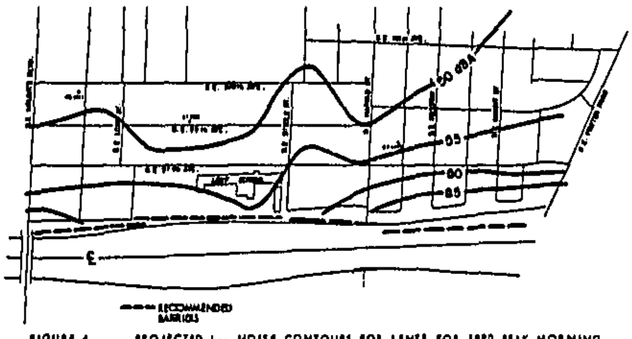

FIGURE 4. PROJECTED L_{50} NOISE CONTOURS FOR LENTS FOR 1990 PEAK MORNING TRAFFIC ON I-205, NOISE CONTROL BARRIERS INSTALLED

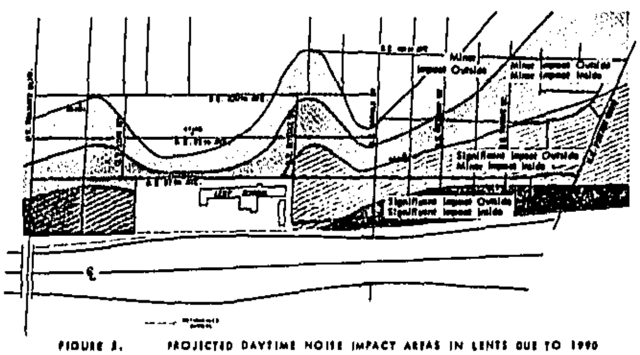

FIGURE 5. PROJECTED DAYTIME NOISE IMPACT AREAS IN LENTS DUE TO 1990 TRAFFIC ON I-205, NOISE CONTROL BARRIERS INSTALLED

THE MEASUREMENT OF NOISE FROM MOTOR VEHICLES ON THE HIGHWAY

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INTRODUCTION

Motor vehicles comprise the most widespread source of noise problems in urban areas today. This is particularly true of those vehicles producing noise levels that "stand out" from the average stream of traffic on highways — the large diesel truck, the noisy passenger or sports car, and many motorcycles. In an attempt to control the increase in noise levels produced by motor vehicles, the California Legislature adopted Motor Vehicle Codes for limiting the noise levels produced by new vehicles, and by existing vehicles on the highway. The responsibility for conducting the necessary highway noise measurements to enforce the Vehicle Code Noise limits was given to the California Highway Patrol (CHP).

The method of measuring vehicle noise on the highway, particularly the selection of suitable measurement sites, was adopted by the CHP from the recommendation proposed by the Society of Automotive Engineers (Reference 1), which was designed for the testing of new motor vehicles. Included in this recommendation is the requirement that the noise measurements should be conducted at a distance of 50 feet from the vehicle path, and that there should be no reflecting obstacles within a 100 foot radius of the microphone or the vehicle at its closest point to the microphone. It soon became apparent that the constraints on the choice of suitable measurement locations imposed by this standard made it almost impossible to take measurements in most urban and residential areas. As a result, a research program was undertaken to provide more substantial evidence on the feasibility of enlarging the number of sites available for highway noise measurements by relaxing site restrictions.*

Examining the feasibility of relaxing these constraints consists first of deciding what error can be allowed in the measurement and second, defining configurations of reflecting surfaces that produce an error no greater than this. Alternatively, if the error due to the presence of reflecting surfaces can be determined, then it can be applied as a correction factor to the noise levels of vehicles measured at that location. If the configuration of reflecting surfaces is too complex for the error to be determined by simple calculation, then the feasibility of determining the error by means of an acoustical calibration must be examined. To verify that these three methods of site selection are valid, and to determine the necessary correction factors, requires a more detailed knowledge of the factors that determine the noise level produced by a passing vehicle. These factors are discussed in the following sections.

FACTOR AFFECTING VEHICLE NOISE MEASUREMENTS

Vehicle Noise Characteristics

In common with most sources of noise, the motor vehicle produces noise as a byproduct of its natural operation. As a noise source, it is much more complex than a loudspeaker, for example, since it contains a large number of moving parts, many of which act as individual producers of noise. Some of these (e.g., the cooling fan, air intake, engine, transmission, exhaust system and tires) are more

*This work was conducted by Wyle Laboratories under contract to the State of California, Department of Highway Patrol (Reference 2).

important than others and by themselves contribute most of the total noise produced by the vehicle. Because these major sources of noise perform widely differing mechanical functions, it is to be expected that the frequency spectrum of the overall noise from the vehicle will be complex. Also, since these sources are distributed about the vehicle to a certain extent, the noise radiated may vary from front to rear or from side to side.

Perhaps more important than these two factors alone is that different types of vehicles will generally display different spectra and directivity due to variations in mechanical design. The degree of variation is sufficiently large to preclude the hypothesis of an "average vehicle" noise characteristic.

The Propagation Path

In the absence of the ground, the sound waves radiated by a typical vehicle will spread out according to the well-known inverse square law (i.e., the noise level will drop 6 dB for a doubling of the distance from the vehicle) except at positions very close to the vehicle. The effect of the ground is to provide a reflected signal at the receiver in addition to the direct, unreflected signal. The resulting interference between the direct and reflected signals can reduce the attenuation, due to spreading of the sound waves, to a value less than that given by the inverse square law. The magnitude of the reduction, however, depends on the dimensions of the vehicle and the spectrum of the radiated noise. Since both these factors vary considerably among different vehicles, the behavior of the noise level with distance is a function of the vehicle type.

At some locations, it might be necessary to take measurements at distances from the vehicle other than the 50-foot distance dictated by the existing standard procedure, so as to monitor traffic in a number of traffic lanes. The noise levels measured at other distances must, of course, be corrected to give the levels that would be measured at 50 feet. The required correction depends upon the rate at which the noise level decreases with increasing distance from the vehicle. To reduce the complications that would occur if each type of vehicle required a different correction factor, some generalizations must be made concerning the propagation characteristics. Experimental evidence (Reference 2) indicates that a correction based on the inverse square law is in order when the measurement distance is less than 50 feet. This correction must be subtracted from the measured noise level. It is recommended that measurements should not be made at distances of less than 25 feet from the vehicle path. At distances greater than 50 feet from the vehicle, a slightly lower correction must be used, based on a decrease of 4 dB for a doubling of the distance. This correction must be added to the measured noise level. The difference in the correction factors for the two ranges of distance is the result of interference caused by reflections from the ground. These correction factors for distance are chosen so as to be conservative for typical highway vehicles, i.e., the corrected noise level is equal to or less than that which would actually be measured at 50 feet.

Meteorological Effects

The propagation of the sound waves through the atmosphere can also be affected by wind and temperature gradients. These have the effect of refracting the waves and changing their direction of propagation, as well as producing possible focusing of many different propagation paths at a single point. In the enforcement of the vehicle noise limits, it is of primary importance to ensure that meteorological effects do not result in an error that increases the measured noise levels. In contrast to absolute measurements of vehicle noise, the introduction of an error that decreases the measured noise level is of secondary importance, since it is the motorist who receives the benefit. Consequently, the only meteorological effects that need to be studied are those that could result in an increase in the measured noise level. Fortunately, for distances of importance in this study, the magnitude of the meteorological variations that can significantly affect the propagation of sound waves are either greater than those normally encountered or lie outside the constraints dictated by the measurement code. Thus, it would appear that no additional constraints or corrections are required to account for effects of weather.

Effects of Reflecting Obstacles

The presence of reflecting obstacles is the source of the most significant error that can occur in noise measurements, and hence received primary emphasis in this study. Since vehicles do not generally radiate sound energy equally in all directions, some reflector configurations will produce a larger error than others. This means that reflecting obstacles can assume an importance that is dependent on the directivity of the noise radiated by the vehicle, and this directivity is not the same for all vehicles. There is therefore no single reflector configuration that will produce the same error in measurement for all types of vehicles. Fortunately, this does not prevent general limits from being formulated for the effect of reflecting obstacles. It can be shown (Reference 2) that the calculated error in the measured noise level of a vehicle which radiates noise equally in all directions represents a conservative estimate of the real error obtained with any other vehicle in the presence of certain types of reflectors, regardless of its directivity. The types of reflectors for which this is true are plane surfaces that are parallel or perpendicular to the vehicle path. This validity also holds in the case of corner reflectors, where surfaces that are parallel and perpendicular to the vehicle path meet. With this simplification, the errors produced by the above types of reflecting obstacles have been calculated and their validity checked by field measurements using typical highway vehicles.

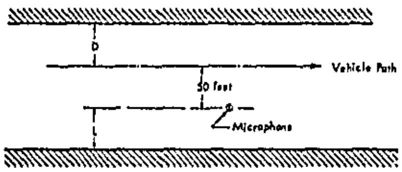
A concise statement of the errors due to reflecting surfaces is given in Table 1 in terms of the increment (increase) in measured noise level that is produced by the presence of the surfaces. Of special note in this table is the fact that a single reflecting surface, parallel to the vehicle path, may be as close as 50 feet from the vehicle or the microphone without affecting the measured noise level by more than 0.5 dB. As the noise level indicated on the scale of a sound level meter cannot generally be read to within an accuracy greater than approximately 0.5 dB, this value is taken as the maximum degree of error allowable without the need for a correction to the noise level. If the single reflecting surface is closer than 50 feet to either the vehicle or the microphone, the error will be greater than 0.5 dB and a correction to the noise level will be required. Table 1 also includes the error or the correction factor required in the case where there are reflecting surfaces on both sides of the vehicle-microphone configuration. These errors are naturally greater than those with just one reflecting surface. It is interesting to note that reflecting surfaces satisfying the constraints in the existing test method (no obstacle(s) located 100 feet from the vehicle and 100 feet from the microphone) could still produce an error of 0.5 dB. The above assumption that no correction to the noise levels is required if the error is less than 0.5 dB is therefore compatible with existing constraints. Thus Table 1 can be used to determine reflector locations such that no corrections to the measured noise levels of vehicles are required, or to determine the correction factor to be subtracted from those levels if the error is greater than that considered acceptable.

There are some types of reflecting obstacles that do not exactly come under the category of a plane surface, yet their effect on the measured noise level can be determined in some circumstances. Some typical examples of such obstacles are embankments, stationary automobiles, small billboards and signs, telephone kiosks, etc. It is difficult to claculate the error that results from reflections from these types of obstacles, and so an experimental program was employed to measure the error. The main conclusion drawn from the results of this program is that reflecting obstacles smaller than 8 feet long and 2 feet high in the immediate vicinity of the measurement site do not produce any significant error in the measurement of the A-weighted noise levels of highway vehicles. Neither, it appears, do bushes or even fairly dense hedges, unless they are situated between the vehicle and the microphone. Under these conditions, a reduction in the measured noise level is possible.

Site Calibration

The configurations of reflecting surfaces covered by the above considerations are sufficiently basic as to be "easily defined." In other words, the suitability of a particular site for highway noise measurements can be checked simply by measuring the distances of certain allowable types of reflecting surfaces from the microhone. Even so, the number of highway sites that can be chosen by this method is limited by the presence of reflecting surfaces that do not come under the description of

Table 1. Numerical Correction Factors in dB to be Subtracted from the Noise Level from Vehicles Measured in the Presence of Reflecting Surfaces



L (feet) \ D (feet)	∞	100	50	40	30	20	10
∞	0	0.2	0.5	0.6	0.8	1.2	1.8
100	0.2	0.5	0.9	1.1	1.3	1.7	2.3
50	0.5	0.9	1.4	1.5	1.8	2.2	2.8
40	0.6	1.1	1.5	1.7	2.0	2.4	3.0
30	0.8	1.3	1.8	2.0	2.3	2.7	3.3
20	1.2	1.7	2.2	2.4	2.7	3.1	3.8
10	1.8	2.3	2.8	3.0	3.3	3.8	4.5

being "easily defined." Thus, in order to expand the method of site selection to include locations having a greater complexity of reflecting obstacles, it is necessary to examine the feasibility of a measurement site calibration.

Basically, the calibration of any unknown system is performed by determining the effect of that system on a known input signal. In this case, the system is the configuration of reflecting obstacles in the immediate vicinity of a highway location at which it is desired to measure the noise level of passing vehicles. The effect of the system that it is required to determine is the change in noise level measured at the microphone due to the presence of the reflecting obstacles. The input, or calibration signal, must be an acoustic signal that to some degree simulates the noise radiated by highway vehicles.

For the purpose of calibration, the source used to generate the signal must be driven along a path 50 feet from the measurement microphone in an area free of all reflecting surfaces and the maximum noise level noted. The same source is then driven through the desired measurement site and again the maximum noise level noted. The difference between the two noise level readings is a correction factor, due to the presence of reflecting obstacles, to be applied to all subsequent noise measurements of highway vehicles. Providing that the noise level of the calibration signal is measured at 50 feet in the area free of reflectors, the location of the microphone with respect to the vehicle path in the desired measurement location is unimportant. In theory, any conceivable highway location and microphone position can be calibrated.

Field tests to verify the validity of the method of site selection by calibration have shown that for a great number of typical highway locations the correction factor obtained is a conservative estimate of the error noted in the noise level of typical vehicles (Reference 2). There are locations where the method does not work, notably where the reflector configuration results in the formation of acoustic resonances. However, it is possible to define such problem locations so that they can be avoided.

Spacing of Other Vehicles

An additional factor in the measurement of vehicle noise at any site, regardless of the method of selection, concerns the noise levels produced at the microphone by vehicles other than the one under scrutiny. Obviously, if the noise from these other vehicles affects the noise level of the measured vehicle, an error is introduced into the measurement which is therefore rendered invalid. The error that is introduced will depend upon how close other vehicles are to the one under measurement; the greater the separation, the lower the error. For the error to be insignificant (e.g., less than 0.5 dB), some constraint must be laid down concerning the distance between vehicles on the highway. Since it is difficult to accurately judge the separation between moving vehicles, it is better for the distance constraint to be translated into a constraint that can be measured directly with the equipment at hand, i.e., a sound level meter. The procedure is very simple and consists of noting the rise and fall of the noise level before and after the maximum level that occurs as the vehicle passes through the site. It can be shown (Reference 2) that the measurement is valid if the noise level rises and falls at least 6 dB about the maximum level.

CONCLUSIONS

The study briefly outlined above shows that the existing constraints on the choice of highway noise measurement sites can be relaxed. As an example, it is possible for a wall parallel to the vehicle path to be as close as 50 feet to either the vehicle or the microphone without introducing an error of more than 0.5 dB in the measured noise level. For two walls, however, parallel to the vehicle path, the existing constraints (i.e., 100-foot distance) still hold. If satisfactory sites still cannot be found, then reflecting obstacles can be allowed closer to the microphone and the vehicle, provided that a correction factor is subtracted from the measured noise levels. Furthermore, measurements can be taken at distances less than or greater than the normal 50 feet from the vehicle at its closest point of approach to the microphone, and a distance correction applied to the noise levels. This approach can be taken whether or not there are reflecting obstacles sufficiently close to require a correction. It is important to realize that these steps can be taken to facilitate the choice of suitable measurement sites purely by inspection of the sites. All that is required is a tape measure and two tables of correction factors. In this way, many desirable measurement sites that hitherto have not satisfied the existing requirements can be used with a minimum of inspection.

It should be noted that errors in the measurements that result in a decrease of the measured noise levels have, within reason, been assumed to be of secondary importance. Furthermore, any corrections have been chosen to be conservative on the high side. This means that the guidelines are not always applicable to the absolute measurement of the noise level from, for example, the new vehicle tests.

In some locations the surrounding obstacles will not conform to those particular configurations amenable to selection by inspection. In these cases, the method of site calibration can be used for a wide range of additional sites which satisfy certain constraints specified in the report. Within these constraints, the method of site calibration will provide a conservative correction factor and can therefore be an extremely powerful technique when properly applied.

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NEW DEVELOPMENTS IN THE CONTROL OF RAILROAD WHEEL SCREECH NOISE.

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I. INTRODUCTION

The noise generated by railroad wheels on sharp turns has been a source of discomfort since the introduction of railroads. The control of this noise source has been approached in the past with auxiliary treatments such as lubrication of the rails, vibration isolators between the shaft and the shoe of the wheel, etc. A series of research and development tests have been carried out in laboratory experiments and, then, in field trials by using the most efficient viscoelastic vibration damping materials, first in homogeneous layers, and then in multiple rings to obtain optimum noise reduction with minimum weight. Test data on the laboratory experiments and extended field tests will be described providing up to 24 dB attenuation with the total weight of the vibration damping treatment of less than 10 lbs. This represents approximately 2% of the weight of the railroad wheel to be treated.

II. DISCUSSION OF TESTS ON WHEELS

The railroad wheel by virtue of its circular symmetry vibrates like a bell, or loudspeaker, with well defined nodal lines of vibration and its radiation effectiveness is well demonstrated as screech noise can be heard for distances over a mile and can be as high as 110 dB inside a typical New York City subway car. From the noise control engineer's point of view we have to evaluate the typical steel railroad wheel most popular in the United States, approximately 28" dia., 550 lb. steel forging, with a cross section shown on Fig. 1.

FIG. 1

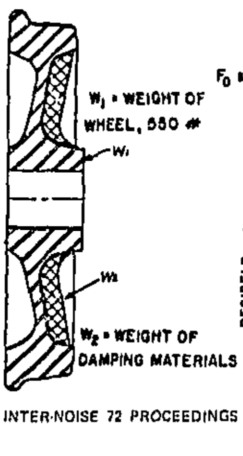
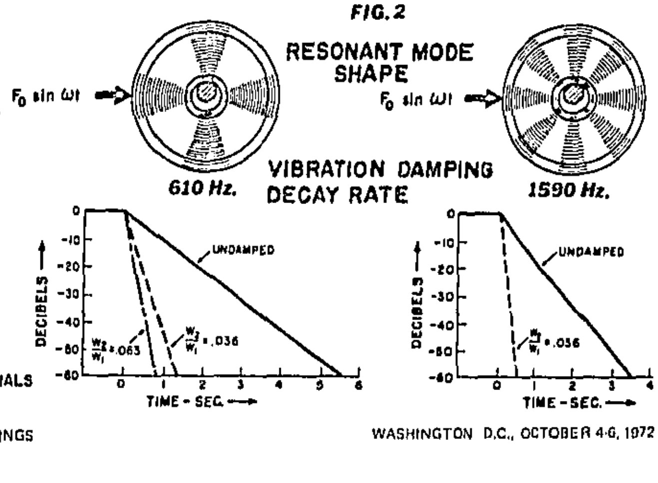


FIG. 2



A typical forged steel wheel weighing 550 lbs. was tested with different vibration damping materials in the Soundcoat laboratory. The railroad wheel was supported on nylon cords in a frame and excited with a vibration exciter to obtain the principal modes of resonant vibration of the untreated wheel. Typical modes of vibration are shown on Fig. 2.

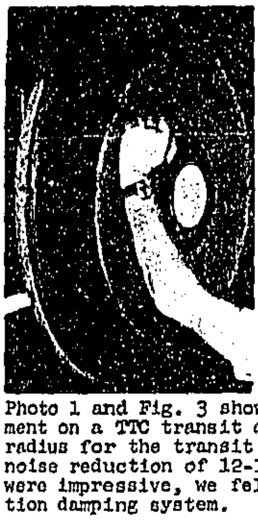
We found from the 1/3rd octave band analysis on a running vehicle that the highest noise peaks were at wheel resonances of 610 cps, and 1590 cps. The laboratory experimental test results are illustrated on Fig. 2; the 610 cps and 1590 cps resonances of the wheel were clean-cut nodal patterns indicating four (4) nodal lines at 610 cps and eight (8) nodal lines at 1590 cps.

Each railroad or transit system has its own problems and the damping or noise control application has to be developed for the specific system, because clearances vary around the wheel from car to car, and lateral space allowed for damping treatment on the wheel as additional thickness at the shoe of the wheel, has to be thin enough, to clear signal system, frog switches, routine maintenance and machine operations required in the grinding of flat wheels.

Within such frameworks of specifications and restrictions, a number of different damping systems were designed for the railroad wheels of the Toronto Transit Commission (TTC), The New York Port Authority Transit System (PATH) and Bay Area Rapid Transit System (BART). These damping treatments comprised homogeneous extensional (free-layer) damping, multiple shear deformation (constrained-layer) damping, and optimum combinations of these.

Field Tests on Extensional Vibration Damping System

PHOTO # 1



"A" SCALE NOISE LEVELS FOR MICROPHONE PLACED FIVE FEET FROM TRACK

FIG. 3

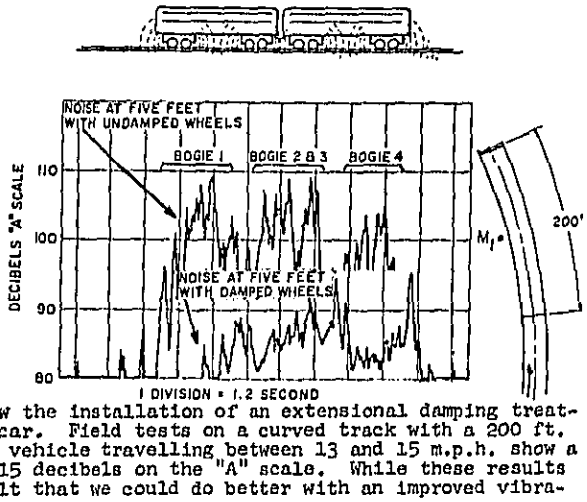


Photo 1 and Fig. 3 show the installation of an extensional damping treatment on a TTC transit car. Field tests on a curved track with a 200 ft. radius for the transit vehicle travelling between 13 and 15 m.p.h. show a noise reduction of 12-15 decibels on the "A" scale. While these results were impressive, we felt that we could do better with an improved vibration damping system.

Design Goals and Development of a Combined Extensional and Shear Vibration Damping Treatment for Railroad Wheels.

Extensive testing was done at the New York Port Authority System (PATH), with the ultimate goal of developing an optimum damping treatment for railroad car wheels. The goals were the following:

a) At least 24 dB reduction in screech noise. (For all practical purposes, this reduction will make the screech noise inaudible compared to the running noise of the car).

b) Treatment to be confined to the wheel rim.

c) Treatment weight not to exceed 22 lbs. (The goal was to limit the treatment weight to 4% of the wheel weight of 550 lbs.).

d) A fail-safe treatment which guarantees adhesion of the treatment for the life of the wheel.

e) Mechanical protection for the treatment so that it would not be damaged by interference from switches, guard rails, track cross-overs, cleaning equipment, etc.

PHOTO #2

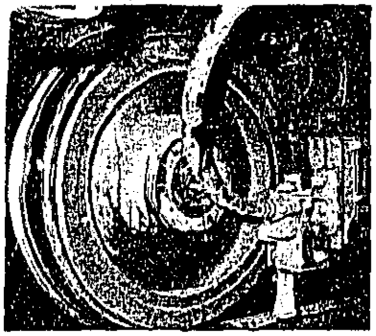


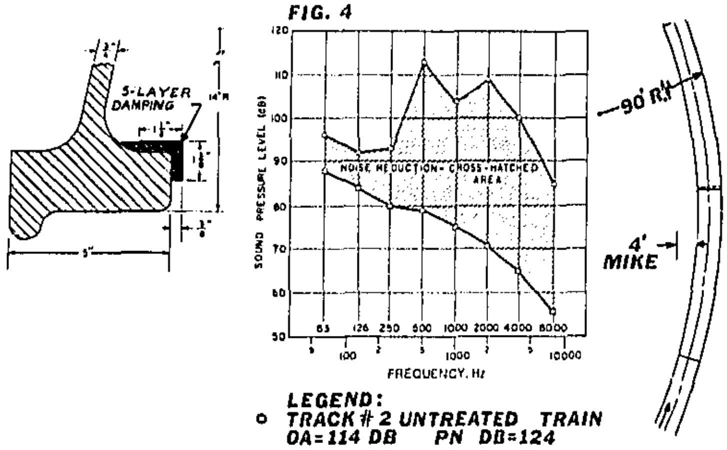
Photo 2 illustrates a five-layer damping ring of combined extensional and shear damping installed on a PATH System transit car in revenue service under test on the sharpest radius curves available in the United States railroad industry...90 ft. radius at Hudson Terminal.

In order to obtain maximum damping efficiency in minimum space and weight, we developed a five-layer ring which is effective over the whole audible frequency spectrum. Thus, excellent damping is obtained for the high modes of vibration, rather than only for the relatively small damping and loss factors which can be obtained with the three-layer system. In addition, from the mechanical point of view, we know that for a ring configuration, the constraining layer does not shift the position of the neutral axis in the right position for maximum damping. We have split our ring treatment, which allows us to take advantage of the extensional and shear damping as well, and provides a very effective non-linear damping system, because of the large elliptical deformation during vibration in the steel wheels.

Noise Reduction Results:

The noise reduction obtained in field trials is given (Fig. 4) for a five-layer damping treatment, which meets all our design goals in terms of weight, space limitations, temperature extremes and mechanical strengths. Noise levels with and without treatment on the sharpest curve available in the United States railroad industry (90 ft. radius at Hudson Terminal in New York City), are illustrated for two untreated and two treated cars (with 16 wheels each) rounding the 90 ft. radius curve. The amount of noise reduction in overall noise levels is 25 dBs. In PN dBs (Perceived Noise Levels) untreated it is 125 PN dBs and with damping it is 93 PN dBs, or a reduction of 31 PN dB. The noise reduction in the octave bands centered at 500 cycles is from 113 dBs to 79 dBs, and at the next highest screech peak in the 2000 cps octave band it is from 109 to 71 dBs. Tests were also run in the open and in a tunnel where the highest noise levels can be expected due to the reverberations of the tunnel walls, and a similar noise reduction was confirmed in all three cases.

FIG. 4



LEGEND:
o TRACK # 2 UNTREATED TRAIN
OA=114 DB PN DB=124
□ TREATED TRAIN (DAMPED WHEELS)
LEFT TO RIGHT SLOW SPEED
OA=89 DB PN DB=93

Vibration Damping for BART System Field Test:

On September 14, 1971, our company was approached by the system designer and acoustical consultant for the BART system, describing screech noise problems occurring on curves with a radius of 540 feet. This system was laid out specifically with large radius curves to avoid the generation of screech noise. Nevertheless, at the speeds which these new trains were operating, considerable screech noise was being generated. This system used a combination aluminum and steel shoe wheel, where the hub section is made of cast or forged aluminum and a steel shoe is press fitted to provide a long wearing, running surface. This wheel weighed 450 lbs. The acoustical consultant further stated that (a) the problem in this case does appear to be the noise radiation from the wheel web and, therefore, applying damping to the wheel web may give the best results, (b) a simple and easy installation of the rings is desired, (c) accomplishing this project in a short time (30 days) is essential, (d) a 20 dB reduction will be more than adequate in the high frequencies to bring about the desired inside noise levels for passenger comfort.

The vibration damping treatment which accomplished all of these objectives was designed in eight (8) days, and manufactured in fourteen (14) days. This design is shown with the resultant noise reductions on Fig. 5.

FIG. # 5.

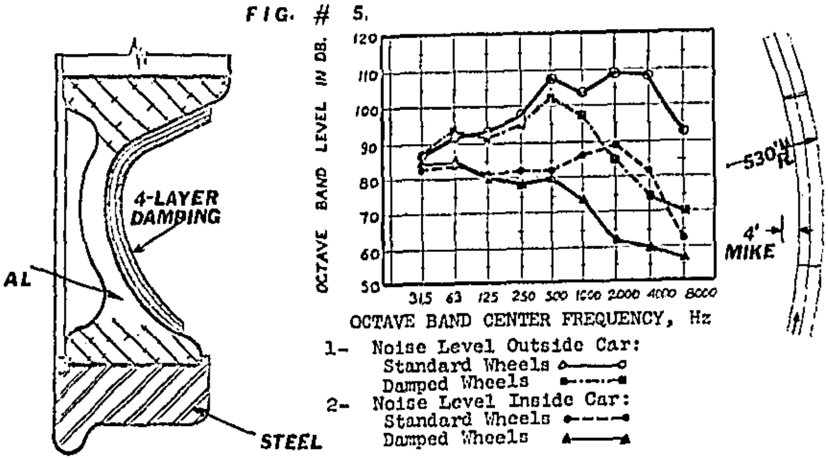


Photo 3 shows the installation on one of the BART transit cars. You will note the treatment was designed to be held in place by three (3) bolts in addition to the adhesive, attached to the shaft which carried three (3) pre-drilled holes for the attachment of the Odometers or similar measurement instruments. The total weight of this treatment is 9½ lbs., and its total thickness is about 1/4". The noise reduction obtained inside and outside the car is as follows:

	2000 Cycles (dB)	4000 Cycles (dB)
Inside Car	27	25
Outside Car	24	32

These tests refer to speeds at 18 m.p.h. on the 530 ft. radius curve. Similar noise reductions were obtained at 540 ft. radius curves in other areas.

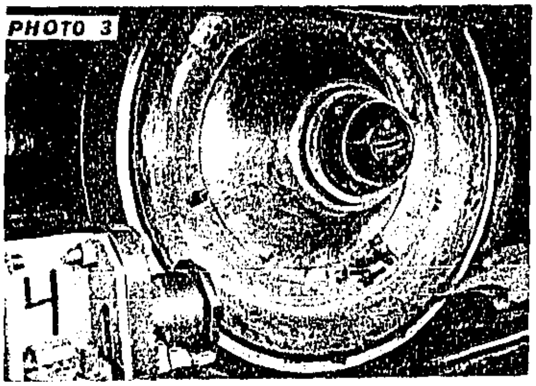


Photo 3 illustrates a four-layer damping ring of combined extensional and shear damping installed on a BART System transit car.

III. CONCLUSION

The rapid development of vibration damping technology is illustrated in the three (3) tests or in the three (3) development models of vibration damping treatments for railroad wheels in three (3) different transit systems...each with its own problems...each requiring a different treatment and a different solution. The present state of the art of vibration damping technology allows us to provide vibration damping treatment for heavy forged steel or aluminum railroad wheels weighing up to 550 lbs. with rim thicknesses of five (5) inches and providing noise reductions of 24 dB or more when required. In effect, a vibration damping system can be designed for railroad wheels which will provide 3 dB of noise reduction per pound of added weight, and that for a railroad car weighing as much as 200,000 lbs.

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ACOUSTIC ENGINEERING CONSIDERATIONS IN THE DESIGN OF A TRACKED AIR-CUSHION VEHICLE (TACV)*

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The TACV seems to be leading the field of potential high-speed ground transportation systems in the United States. This paper briefly discusses the acoustic design of a proposed 150 mph, 60 passenger prototype vehicle by LTV Aerospace Corporation (Ref. 1). The arrangement of the vehicle is shown in Figure 1, with a typical cross section in Figure 2. It was some 120 feet long and intended to travel in a channel-section guideway. The vehicle was levitated by a continuous series of air-cushion arrays, underneath, with a similar arrangement along the sides to provide guidance. Two fans at the front supplied air to the cushions via a duct down each side of the vehicle. The cushion concept comprised a shallow, inverted pan suspended from a flexible air-bag (Figure 2). The gap between the pan lip and the guideway was 0.14". Stringent environmental requirements were imposed by DOT to allow operation in urban areas, and to provide unexcelled passenger comfort. The specified acoustic levels (Ref. 2) are tabulated below.

50 ft. sideline		Passenger Compartment	Crew Compartment
Terminal	150 mph		
63 dBA	73 dBA	65 dBA	75 dBA

EXTERNAL NOISE ANALYSIS

Sources. The sources considered in the analysis were as follows:
- (i) The fan inlet;
- (ii) The air-cushion discharge air;
- (iii) The turbulent boundary-layer; and
- (iv) Secondary sources; wake turbulence, structural radiation, power collection shuttle, LIM rail, on-board equipment, brakes and dump air from the supply ducts.

Each of the primary sources is discussed in the following paragraphs.

Fan Inlet. The 400 hp, single-stage, axial fans each delivered 46,500 cfm. The radiated acoustic power was estimated to be 133 dBp (re 10^{-12} watts). Assuming hemispherical radiation, this led to a 50 ft. level of 101 dBA. To meet the terminal goal, an attenuation of some 40 dBA was indicated. The form of the muffler selected is shown in Figure 4. It comprised 60" splitters, 2" deep with a 2.75" air gap. The length was dictated by low frequency requirements and the gap by fan performance. The splitter data were obtained in the laboratory without airflow. However, limited tests were made with the flow opposing the sound at velocities up to 130 ft/sec. (Ref. 3). With these data the laboratory results were modified to give the curve in Figure 5. The associated spectrum is included in Figures 6 and 7, for a level of 66 dBA. Because of the many uncertainties involved in the design, and the relative importance of the contribution of the noise source, it was considered imperative that a full-scale test should be made as soon as possible in the vehicle development phase.

Air Cushions. Referring to Figure 2, it is seen that the lift cushion discharge air is entrapped within the guideway; however, the top edge of the guidance cushions is close to the top of the guideway parapet and thus is a potential source of noise, both in the far-field and the passenger compartment.

This source represented a new variation on an old theme and some experiments were made in the Vought Aeronautics Company anechoic chamber with a 1/8" x 24" slot. Two configurations, sketched in Figure 3, were tested at pressures from 4" to 28" of water. The overall sound pressure level varied as velocity to the sixth power, with a spectrum related to the slot width, see Figure 3. The directivity was uniform to 90° from the jet axis, then reduced rapidly to zero. Following the format for a circular jet, corrected for the increased periphery, a constant $k = 1.6 \times 10^{-6}$ was derived from the measured power. The acoustic power is then given by,

$$W = k\,A\,\rho\,V^6/2c^3 \text{ watts}$$

Where, A = area of orifice; V = exit velocity; ρ = air density; c = speed of sound. From the modified formula, and the measured octave band spectrum, the power per foot was calculated. This was then treated as a line source in which the mean square pressure is given by,

*This work was largely performed under contract to the Urban Mass Transportation Administration (UMTA) of the U.S. Department of Transportation

$$p^2 = \frac{2W_s}{\pi d} (\tan^{-1} \frac{L}{2d}),\ (lb/ft^2)^2$$

Where, W_s = watts/ft; d = sideline distance; 2L = source length. The predicted 50 ft. side line level for full lift was 59 dBA with the spectrum shown in Figures 6 and 7. No attempt was made to investigate for the effect that forward speed or crosswinds might have on the power developed.

Turbulent Boundary Layer. In predicting the noise radiated by this source the method employed by Eldred (Ref. 4) in an analysis of jet fighter glidebys was used. That used an analogy with an air-jet to calculate the acoustic power developed per unit area of vehicle surface. The equation derived was,

$$W_A = 2.62 \times 10^{-6}\ \rho\ U^8/C^5\ \text{watts/ft}^2$$

Where, U = vehicle velocity. As in the case of the cushions, the sideline level was estimated for the line source assuming uniform radiation. The octave band spectrum was assumed to follow that of the surface pressures found by empirical methods (Reference 5). The resulting 50 ft. sideline level was 68 dBA with the spectrum shown in Figure 7. This proved to be a most important contributor to the noise level.

INTERNAL NOISE ANALYSIS

The objective in the acoustic design was to achieve a good balance between the noise reduction of each transmission surface; i.e., floor, roof, sidewalls, windows and doors. To minimize on-board equipment noise, potentially noisy items were placed in a compartment at the rear of the vehicle. Intervening compartments were used for baggage and passive electrical components. Interior absorption was provided by a carpet and pad, seats, trim panels, and a perforated treatment in the ceiling. To minimize energy transmitted structurally and via flanking paths, liberal use was made of viscoelastic damping in the design. All skin panels and frames in the passenger compartment were to be spray-coated with a damping compound.

The sound pressure levels from each of the sources is calculated from —

$$SPL_n = PWL_n - \log S\bar{\alpha} + 16\ \text{dB, where, } S\bar{\alpha} = \text{total absorption}$$

Sources. The sources of interior noise were:

(i) Boundary-layer and buffet;
(ii) Fan noise in the duct;
(iii) Air-conditioning;
(iv) Secondary sources; adjacent highway vehicles, air-cushion discharge, on-board equipment and structure-borne vibrations.

The methods of determining the contributions from the primary sources are discussed below.

Aerodynamic. The overall power radiated by the structure due to excitation by the boundary layer was estimated using empirically developed methods, based on the work reported in Refs. 7 and 8. The total power radiated was calculated from,

$$PWL_T = PWL_P + \log \frac{A}{K} \frac{(P_T + P_P)}{P_P} \quad \text{dBp (Ref. } 10^{-12} \text{ watts)},$$

Where, PWL_P = power radiated by a unit panel; A = area of radiating surface; P_T = 2 (length of stiffeners); P_P = total periphery of panel; K = experimentally determined constant ≈ 2. The spectrum limits are controlled by the fundamental panel resonances and the coincidence of pressure field convection velocity (≈ 0.8U) and the flexural wave speed. The acoustic power is uniformly distributed by octave bands between these frequencies. A 10 dB increment was applied to the corner roof panels to account for buffet along the roof line.

The overall of 76 dBA indicated the need for a minimum of 11 dBA reduction. This was accomplished primarily with an isolated inner wall of lead/vinyl, and the use of double-glazed windows. The A-weighted spectrum is plotted in Figure 8, the overall equalled 61 dBA.

Duct Noise. The floor design, for noise control, was set by the low frequency fan noise in the duct. The estimated overall SPL = 133 dB. Reduction at the source was addressed through the use of low frequency resonant absorbers and splitters. A 6" gap between the splitters was dictated by fan performance needs. The duct itself was lined with a 1" layer of "Ultraliner." The floor was some 6" deep and afforded space for a full treatment of lead/vinyl and glass fiber blankets. However, structural flanking paths remained that could not be interrupted, and to ascertain the full effect of these would have eventually demanded tests. As mentioned above, all of the structural surfaces were coated with a viscoelastic compound to minimize these effects.

Air-conditioning. This source is controlled

(1) By limiting flow and discharge velocities to a maximum of 1,200 ft/min.,
(2) By the use of mufflers on delivery and return ducts to remove mechanical noise, and
(3) By ensuring that the duct walls are not exposed to other noise sources which would shortcircuit the treatment.

The estimated contribution of the air-conditioning source to the interior noise of the vehicle is shown in Figure 8.

RESULTS OF ANALYSIS

The analyses made in the design of the proposed vehicle, to the specified requirements, showed two of the noise sources to be particularly important. These were:

(1) External analysis, the radiated boundary layer; and
(2) Internal analysis, the fan noise in the ducts.

The first is fundamental and clearly represented a subject for future research. The second presented a challenge in the reduction of low frequency noise and the elimination of flanking paths, for a vehicle in which the duct is an integral part of the structure. Two approaches are suggested for further study to minimize that challenge, namely,

(a) To employ several smaller fans rather than one large fan per side, and
(b) To structurally isolate the supply ducts from the occupied compartments.

From the community noise viewpoint, the predicted external levels showed the proposed vehicle would be significantly quieter than freeway traffic during cruise. Measurements made 50 ft. from the edge of a freeway pavement indicated 80 dBA for a traffic flow of 50/55 cars per minute. The predicted levels in the terminal are within the range of average to noise residential areas; i.e., 50 dBA – 63 dBA.

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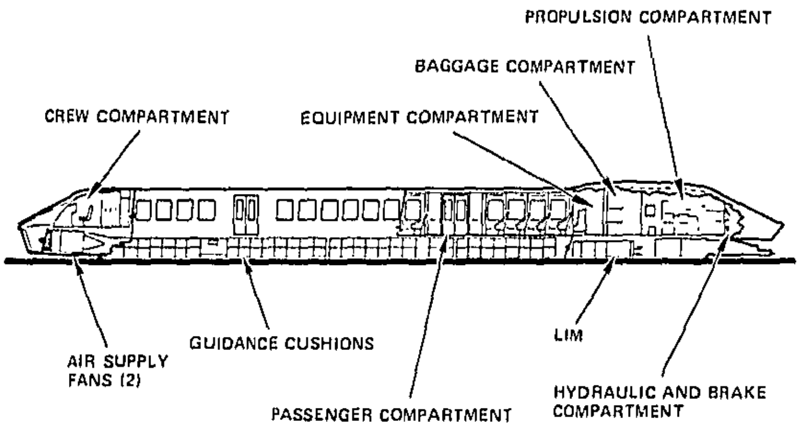

Figure 1 Side View Showing Vehicle Arrangement

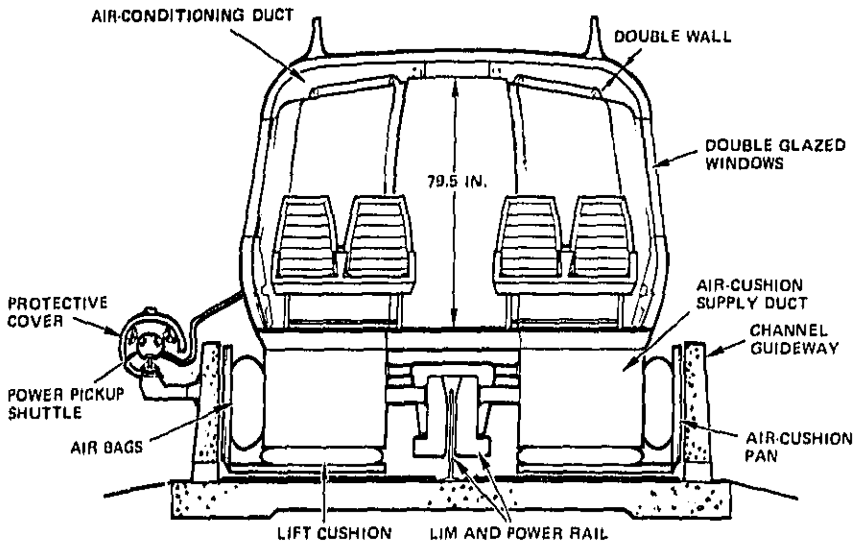

Figure 2 Cross Section of Vehicle

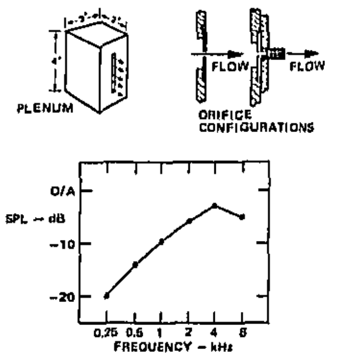

Figure 3 Slot-Jets and Spectrum

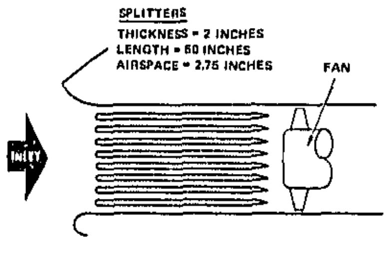

Figure 4 Air-Cushion Air Supply Fan Inlet Treatment

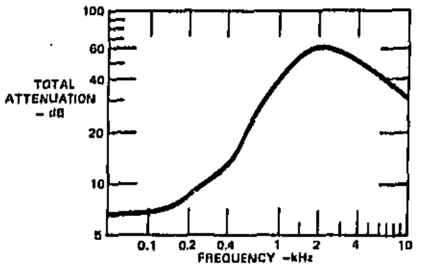

Figure 5 Estimated Total Attenuation of 60-Inch Splitter

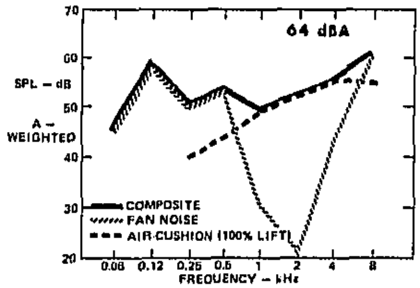

Figure 6 External Noise 50 Ft Sideline Terminal Area

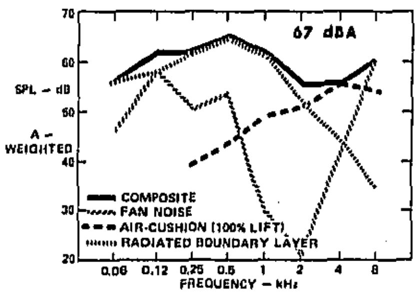

Figure 7 External Noise 50 Ft. Sideline 150 mph

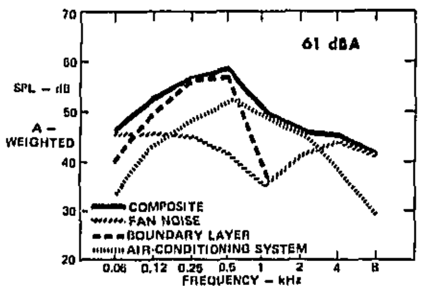

Figure 8 Passenger Compartment Levels 150 mph

STUDYING THE EFFECTS OF SNOWMOBILE NOISE ON WILDLIFE

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INTRODUCTION

At the present time, there are over two million snowmobiles in North America. One million machines have been sold in the United States during the past five years. Although the growth rate is levelling off, it can be expected to continue at the current annual rate of 500,000 to 600,000 units until the end of this decade. Recreational management methods and organized facilities for the sport generally have not kept up with its rapid increase in popularity. As the demand for additional facilities rises, the need for environmental design criteria and guidelines, for both vehicles and trails, becomes apparent. The most important area of attention in this regard is that of snowmobile noise emission. The noise produced may have any of four types of potentially significant effects:

- The effects on the hearing of drivers or passengers in the form of temporary or permanent noise-induced threshold shift.
- The effects on people engaged in indoor activities in the proximity of snowmobiles.
- The effects on people engaged in outdoor activities other than snowmobiling.
- The effects on wildlife due to the impact of increased noise levels in the animals' habitat.

Research into the effects of noise on wildlife in its natural habitat has been virtually non-existant. This paper presents some of the results of studies carried out at The University of Wisconsin during the past winter dealing with the effects of snowmobile noise on the movements of wild rabbits and deer. Separate studies were run for each of the two types of animals.

The Purpose of the Studies

The hypotheses tested in both cases were the following:

- Will snowmobiles cause the animals to move outside their normal home ranges?
- Will snowmobiles cause the animals to seek shelter and stay there until the disturbance stops?
- Will the animal activities be significantly different between periods when snowmobiles are or are not present?

Efforts were also made to isolate effect of noise versus the effects of snowmobile presence, lights, and exhaust fumes.

Methods

For each study, a number of animals were trapped and equipped with radio transmitters enabling animal positions to be determined with antennas and receivers.

The animals were followed without snowmobiles present to establish normal habits and home ranges. Snowmobiles were then run in the area in various patterns and noise levels were recorded for later analysis.

THE INVESTIGATION

The Rabbit Study

Seven wild rabbits were studied in an isolated woodlot covering 14.5 acres of The University of Wisconsin experimental farm near Lancaster, Wisconsin. There had been no previous snowmobiling in the immediate area. The group of seven rabbits was comprised of one adult female, two adult males and four juvenile females. Each rabbit was equipped with a radio-transmitting collar operating at a different frequency between 52.50 and 53.50 MHz. The transmitters had lives of 125 to 255 days and ranges of approximately one half mile.

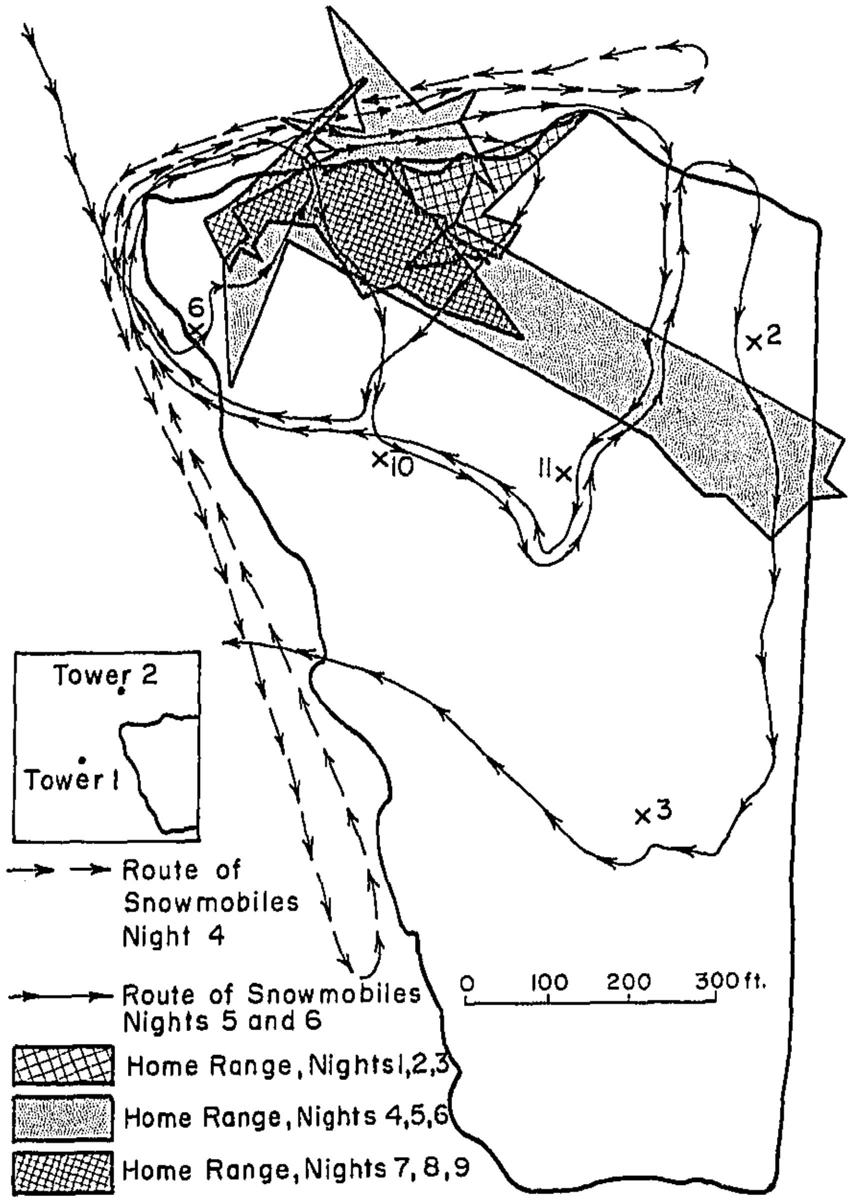
The transmitted signals were monitored at two towers with receivers connected to three-element yagi antennas placed ten feet above the ground. A pair of antennas was used at each tower. The antennas were stacked so as to produce a sharp null in the received signal when the antennas were pointed at a given rabbit. The angular positions of the rabbits were established by reading pointer positions on a 40 inch diameter protractor. The readings were determined to be accurate to within ± 1 degree by transit locations taken on two fixed reference transmitters.

The rabbits were tracked during nine consecutive evenings and nights from approximately 6 p.m. until 6 a.m. with locations being taken every twenty minutes. Thus an average of 200 locations per night was obtained.

During the first three nights, no snowmobiles were present. During the fourth night, six snowmobiles were run along two outside borders of the woodlot from 7 p.m. until midnight. The machines were run for twenty minutes at a time with twenty minute intervals between runs. The average running speed was between 18 and 20 mph. On the fifth and sixth nights, the six snowmobiles were run on a network of trails inside the woodlot. On the fifth evening machines ran from 6 p.m. until midnight and on the sixth evening from 4 p.m. until 8:30 p.m. Each run lasted 25 minutes and 20 minute breaks were taken between runs. Average machine speed was 7 to 10 mph. The primary reason for taking the long breaks was to permit rabbit locations to be established. In the course of the initial studies, ignition noise from the snowmobile interfered with the signals from the rabbit transmitters. This problem was eliminated in the latter deer studies by the use of special suppressor spark plugs. Figure 1 is a map illustrating the rabbit study woodlot with snowmobile trails. In addition, the home range areas of one rabbit for the duration of the study are outlined.

Noise Levels

The noise produced during snowmobile runs was recorded at five points within the woodlot. Complete runs, both inside and outside the woodlot were recorded at each point. These points are numbered 2, 3, 6, 10 and 11 on the map of Fig. 1. Since all the rabbits inhabited 8 acres in the northern half of the woodlot, the above set of points was deemed to be adequate to describe noise variations throughout the area of interest. The noise levels are described in terms of their distribution in time during a complete 20 or 25 minute run. The 10 percent level is the noise level which is exceeded 10 percent of the time. The 50 and 90 percent levels are similarly defined. These levels were determined in both dBA and octave bands having center frequencies between 63 and 8000 Hz. Although the dBA weighting may be unrelated to animal noise sensitivity, it is presented in Table I as the most widely used single-number measure of overall noise levels. Furthermore, the use of the linear scale outdoors results in



Rabbit Study Area

Figure 1

oversensitivity to low frequency wind variations and gives a less accurate indication of snowmobile noise levels.

TABLE I

Range of Overall Noise Levels During Runs (dBA)

		L_{max}	L_{10}	L_{50}	L_{90}	L_{min}
Run Outside Woodlot	High	79	71	61	55	46
	Low	68	58	54	38	35
Run Inside Woodlot	High	95	72	56	46	41
	Low	89	66	54	42	38

The median (50 percent) noise levels ranged from 54 to 61 dBA when machines were running along the outside of the woodlot and from 54 to 56 dBA when machines were run on the inside trails. The ambient noise level varied from below 20 up to 25 dBA when snowmobiles were not operating with occasional levels to 45 dBA as cars and light trucks passed along a nearby highway. The high-low range of levels encountered for a number of runs while outside the woodlot was much greater than for the inside runs. This difference in range can be attributed to the fact that running the machines outside the woodlot resulted in a greater variation in distance between the machines and microphone locations. The maximum noise levels during the inside runs were considerably higher than those encountered in the outside runs because the snowmobiles passed very close to the microphone at various points along the trails.

The range of noise levels in octave bands are shown in Figs. 2 and 3. Figure 2 shows the ranges of 10 percent levels for both inside and outside runs. Figure 3 shows the ranges of the 50 percent levels. The high levels are very close for both cases but as with the overall levels, the variation in level is greater for the outside runs. It should also be noted that the spectra are quite flat.

Rabbit Movements

The rabbit movements were evaluated in two ways. Firstly, the distances between consecutive locations was totaled for each evening or group of evenings. These distances are not, of course, the actual distances that the rabbits moved. The measure is, nevertheless, related to rabbit movement. The second measure is the rabbit home range. This is the area in which the rabbit stayed for a given period. The home range for one rabbit has been superimposed on the map of Fig. 1. The nine day test period has been broken up into three-day periods. The home range polygons are formed by joining the extreme location points for each three-day period. The rabbit whose movements are illustrated in Fig. 1 exhibited the largest changes in home range of the test group. Representative figures of overall rabbit movements are best illustrated by the data in Table II.

TABLE II

Summary of Rabbit Movements

Day Number		Distance Moved (Averages of 7 Rabbits) ft/hr	Home Range (Average for 7 Rabbits) acres
1	No Snowmobiling		
2	No Snowmobiling	108	0.76
3	No Snowmobiling		
4	Runs Outside	126	
5	Runs Inside	197	1.84
6	Runs Inside		
7	No Snowmobiling		
8	No Snowmobiling	137	1.37
9	No Snowmobiling		

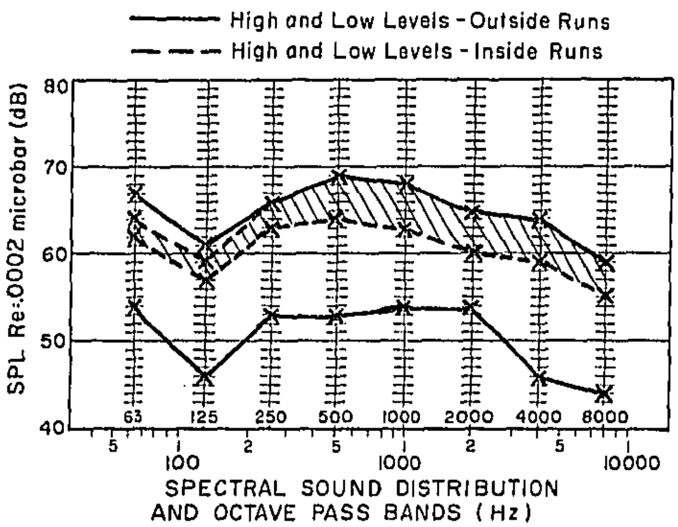


Figure 2
Ranges of 10 percent Levels - Octave Bands - Rabbit Study

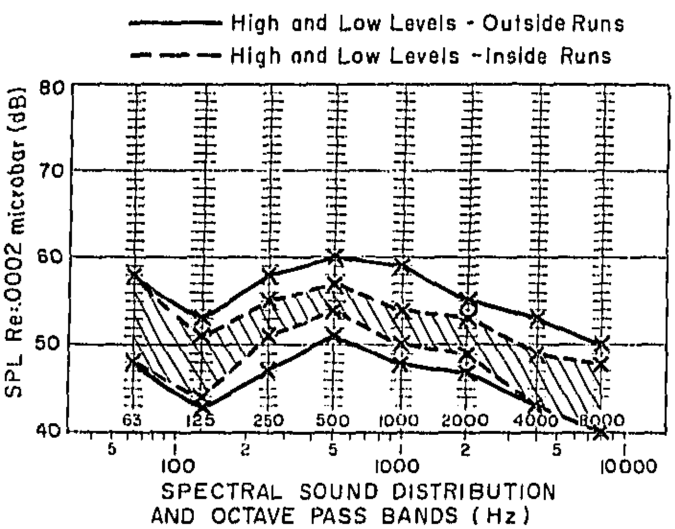


Figure 3
Ranges of 50 percent Levels - Octave Bands - Rabbit Study

When distance moved (ft/hr) is used as a measure of the level of rabbit activity, there was a significant increase in activity during the evenings when snowmobiles ran inside the woodlot compared to evenings when no snowmobiles were present. However, there was no significant increase in activity between the evening when snowmobiles ran outside the woodlot and when no snowmobiles were run. Thus, one could conclude that the increased noise levels of machines not operating in close proximity of the animals did not have a significant effect on activity. It is not clear, however, whether the increased levels of movements obtained with machines operating on trails within the woodlot, was due to higher noise peaks, exhaust fumes, machine lights or presence.

When home range is used as a measure of activity, an analysis of variance indicates that the home range size was increased during periods of snowmobiling. Analysis of the data for all rabbits further indicated that the home range size decreased after snowmobiling ceased, but remained larger than originally measured before snowmobiling.

The Deer Study

The study of snowmobile noise effects on deer was carried out in a 3000 acre area of tamarack swamp near Lewiston, Wisconsin. Twenty deer were trapped and trasmitting collars operating at frequencies close to 150 MHz were placed on eight does and doe fawns. The collars could not be placed on twelve bucks due to the expansion of their necks during the fall season. The most intensive part of the study was concentrated in a 140 acre section of the swamp which was normally inhabited by four of the radio-tagged deer.

Snowmobiling was carried out in the area on eight afternoons during February and March of 1972. Six snowmobiles were run from one to four hours each afternoon. The animal movements were monitored before, during and after snowmobiling. The snowmobiles ran along the perimeter of the 140 acre section of swamp as well as on three inside trails. The most remote point within the 140 acre section was approximately 1000 feet from the nearest trail.

Only on the first day of the study did one of the deer leave the area. This deer returned by the following day and, for the remainder of the study, none of the four deer left the area while snowmobiles were present.

Based on the animal location data, it is, however, likely that deer movements did increase within the swamp during snowmobiling. A complete presentation of the deer study data will be made in the near future.

CONCLUSIONS

The hypotheses tested in these studies have only been examined for short term effects. It is not possible to predict how the animals would behave after more frequent or lengthier exposures. The snow cover during the deer and rabbit studies never exceeded six inches, and generally remained between two and four inches. This amount of snow cover would not be sufficient to inhibit animal movements. Furthermore, both study area are located in farming regions and close to lightly travelled highways. The reactions of animals inhabiting remote wilderness regions with deep winter snow cover may be different from reactions measured during these studies.

The results of the studies do indicate, however, that neither the noise nor presence of the snowmobiles caused the deer or rabbits to leave the areas which they normally inhabited. Furthermore, the animals did not remain stationary at a single location while machines were running. The presence of snowmobiles did appear to increase animal movements within and near their normal home ranges.

ACKNOWLEDGEMENTS

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ON PUNCH PRESS DIAGNOSTICS AND NOISE CONTROL

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INTRODUCTION

The interest in the noise problem in factories has grown recently as a result of several factors: the increase in factory noise levels as faster machines and processes are introduced; the newly documented results of the adverse effects of high noise levels on performance and psychological states; the established relationship between high noise levels and hearing damage and the resulting legal suits for compensation; and finally the role that the government has assumed in enforcing specific noise regulations. Factory noise is a problem when such machines as punch presses, forging presses, textile machines, large compressors, printing presses, wood chipping are used. The study of one type of these machines - presses - is discussed here.

Most manufacturing facilities use punch presses of one size or another. Presses vary in capacity from 40 tons to 1,300 tons; in speed from 20 to several hundred strokes per minute; in stroke length from 1/2-inch to 5 inches. Some presses are horizontal or inclined but the majority are vertical. They use single-step dies or progressive dies, and process metals from steel to aluminum to brass of various thickness. The impulsive forces of presses cause two distinct problems: structural vibrations, and noise affecting the operator. The noise at the operator's position is a widely-recognized problem in industry. To remedy the situation several approaches are possible:

1) Use partial enclosures for the following purposes: a) enclose each machine partially, b) enclose the press area from the rest of the plant. In the first instance the purpose would be to interrupt the direct path of sound from the machine to the operator and may result in 3 to 6 db reduction at his position. In the second case the reduction in the far field would be improved (See Reference 1).

2) The complete enclosure of the machine. This approach has been used successfully in few cases, particularly where one or two machines exist in a plant, and where space and simplicity of operation allows the complete enclosure of the machine.

3) The eventual requirement is the design of a quieter machine. For this it is necessary to understand the mechanisms of sound generation and to design the machine such that less vibration and less sound radiation would take place. Among the parameters directly affecting the noise level of presses are the operating speed, the material processed and its thickness. It was found (1) that a 70% increase in machine speed raised the overall level 5 to 10 db, and that hard steel in a press generates about 5 db more noise than mild steel. The question we are seeking to answer here is, for a given process, material, and speed, what changes can be made in existing machines, and possibly in new designs, to make the process less noisy.

Among the papers written on the subject, the two by Bruce (1) and Hoover (2) dealt with the noise environment in metal stamping areas, the criteria for their measurement, and their effect on communication and on the human

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MACHINERY NOISE (II)

ON PUNCH PRESS DIAGNOSTICS AND NOISE CONTROL

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Among the papers written on the subject, the two by Bruce (1) and Hoover (2) dealt with the noise environment in metal stamping areas, the criteria for their measurement, and their effect on communication and on the human ear mechanism. The present study concentrated on diagnosis of presses to identify the noise causes and to point to the possible solutions at the source.

ANALYSIS OF PRESS NOISE

A. Type of Measurements

In the present studies the following types of measurements were made:
1) Record the sound levels with a sound level meter set on the fast linear response. Sound was recorded at the operator's position and at distances of 6, 12, 24, and 36 feet.

2) Record acceleration at different positions in the press. These included the upper die, the stripper plate, the lower die, and the machine frame.

3) Record acceleration on the foot of the machine as well as that of the floor at the foot of the machine, and at several locations away from the machine.

4) Simultaneously record the position of the moving part of the press while recording the acceleration and sound.

The above measurements were repeated while only one machine was operating in normal conditions, and after making changes in the process as will be discusses later (See Section C2).

The accelerometers used have flat response up to 18,000 Hz. The data was recorded on an FM tape recorder at speeds of 60 i.p.s. which has linear response up to 20,000 Hz. The press position was measured using a Linear Velocity Displacement Transducer with a sensitivity of 2 volts/inch and linear range $\pm$ 3 inches.

B. Data Analysis

The objectives of the analysis were to determine the machine components that contribute most to vibrations and sound, investigate the details of the process to identify the events causing excitation forces and vibration, determine the importance of room reverberation on the noise situation in the near field and far fields of the machine, and determine if sound is transmitted in the room through air or through the floors, to evaluate the effectiveness of machine isolation.

Three methods of data analysis were used:
1) Waveform Analysis. The signal is displayed versus time to see the sequence of events, see for example Figure 1. In this display the sound is compared with the output of several sensors on the die and the frame. The advantages of this method is that it points out the different impacts that take place within one cycle, and gives the relative amplitude of each impact force, and the relative importance of each impact in producing sound. The signal was then filtered into one-octave bands, and again the waveform analysis was repeated. One result is shown in Figure 2. In several cases a summation was made of a number of impacts to bring out the major features of the events and eliminate the variations in operation.

2) Frequency Analysis. The signal is analyzed in the frequency domain and this reveals any periodicities in the accelerations and in sound. This is particularly significant if noise sources include items such as gears or bearings whose mechanical signatures have strong periodic components in the audible frequency range. An example of this analysis is shown in Figure 3.

3) Cross-Correlation Analysis. The sound signal and the acceleration signals of different machine components were cross-correlated after filtering into octave bands from 63 Hz to 2000 Hz using a cross-correlator with total time delays τ from 10 to 100 msec. with $\Delta\tau$ (delay time resolution) 1% of τ and averaging time of 12 seconds. Example of the correlator's output is shown in Figure 4. The results of this analysis

indicated that certain machine elements such as the lower die have stronger normalized correlation with sound than the stripper plate which has the highest level of vibrations. This raises the question as to the relative importance of each of the two criteria in determining the offending component.

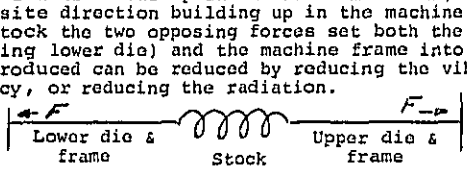
C. Results
This study covered a large number of machines with the following results being common to all: a) The noise levels of the punch press areas are generally the highest in industrial plants, and are a major part of the industrial noise problem; b) The noise problem in most cases is localized, with only those operating the presses or in the immediate vicinity are subject to the high levels of the presses. In some cases, however, the proximity of the press area to other manufacturing sections in a plant causes the noise levels there to be quite undesirable; c) In press areas where an operator is exposed to the noise of his machine as well as those adjacent to him, nearly half of the noise energy reaching him is due to his machine and when this is turned off the sound pressure level at his position would drop 3 db, but the level is still well above that desired. That part of the sound energy due to other machines can be reduced - to some degree - by improving the room absorption on ceiling, walls and by barriers or partial enclosures; d) The noise reduction required in many press areas to meet present and projected government standards ranges between 10 and 20 db (See Reference 2). This requires a reduction of the sound pressure of between 67 and 90 %; e) The sound spectra in all cases are broad and exhibit little or no sharp peaks, indicating that the excited structures are vibrating in many modes. The maximum levels are not always at the same frequencies, as will be discussed later; f) Little sound energy is transmitted through the mountings of the machine. In general, the mountings reduce the force transmitted from the machine to the floor by 20 to 25 db in the higher frequencies and 5 to 10 db in the frequencies below 100 Hz; g) Where air ejectors are used, these add to the overall level only 1 db. In general the noise due to the ejector (or exhaust) air can be well above the 90 dbA level but considerably below the levels due to the mechanical noise.

Different types of presses had specific results that will be discussed below:
1) High Speed Blanking Presses: Tests made on several makes of these presses pointed out that: a) The blanking operation itself generates a large amount of sound energy. In one test the press, in a quiet shop, was run normally with steel stock being blanked, then with the stock removed, and finally with the stock fixed in place so that no blanking was taking place but all other events (such as clamping) occurred. The sound level during blanking was 5 to 7 db above the other two conditions; b) in the time domain sound energy appears to have two bursts, one of which corresponds to the blanking and the other is due to one of different events; c) the die components had considerably higher levels of vibration than the machine frame or component housings, as shown in Figure 3; d) in viewing the motion of the punch (Figure 5) measured by attaching a displacement sensor to the upper die, the simple sinusoidal movement is briefly interrupted near the bottom of the stroke either due to the resistance of the material to shear or due to the slack in the crank arm pin or bearing; e) in some cases the sound energy of presses operated with no stock in them was considerable, with sound levels only few db below the level during blanking; f) the frequency composition of the sound energy is such that the highest levels were at frequencies below 1000 Hz. The level drops about 6 db per octave above that frequency (Figure 6).
2) Presses With Progressive Die: In several of these presses the tests indicated: a) the noise levels when processing metal was equal to or slightly less than the noise levels of the machine operating with no stock. It is possible that the presence of stock adds to the damping due to friction at the interfaces. It is evident however, that the forming and blanking of the material are not necessarily the major source of sound. This is also explained by the impacts of the stripper plates preceeding and following the punching or forming action; b) there were two distinct,

nearly equal, sound energy bursts for each stroke; c) the highest levels of vibrations were those of the stripper plates which have low damping qualities. This was true even in presses whose noise during blanking far exceeds the noise with no stock; d) The frequency composition of the sound energy in progressive dies shows the highest energy levels to be at frequencies between 600 and 6000 Hz. Because of their higher frequency content (against blanking dies) they have higher overall level on the A-weighted network.
3) Forging Presses: These presses are usually run at much slower speeds, and their noise problem is due to the following conditions: a) the impact sounds during forging are intermittent and should be measured either with an impact level meter or with the regular meter on the C Fast weighing network on the 130 db range. The law requires that the sound should not exceed 140 db on the impact level meter or 125 db on the C-weighted scale. In presses the impacts generally do not exceed the limits specified above, but when their impact noise is added to the continuous machine noise it is quite significant and need be considered in assessing the required noise reduction; b) the machine components, such as gears, bearings or other moving parts, can be excessively noisy. This becomes true as these components become worn or if poorly maintained. In such cases, particular tones and harmonics will be detected in the sound and vibration signals, and it will be possible to pinpoint the offending machine element. The noise generated by these parts will be continuous as long as the machine is on, regardless of whether there is any production and as such they are more irritating.

D. Solutions
Noise problems can be eliminated by requiring everyone in the area to wear ear plugs, which in most cases are uncomfortable and impractical, except for short periods, to cover the machine with a cover lined with acoustical damping, sound absorbing material, or, the best way to eliminate the noise is to prevent it at the source, that is at the moving parts that cause vibrations and radiate sound. We will discuss the last two methods in some detail.

1) Press Noise Elimination by Modifications: a) Gear Noise. The noise from gears depends on the natural frequency of the teeth, the damping in the body of gear, the quality of surfaces, and the tolerances which affect the dynamic forces during meshing. The approach to reducing the noise is: To maintain smooth surface and tighter tolerances to reduce the dynamic loads. This can best be achieved through better lubrication, such as immersing the meshing teeth continuously in a thick oil film. To re-design the teeth so that their natural frequencies would be less than the meshing frequency. This can be done by, for example, selecting their material to have low modules of elasticity. To reduce the ringing of the gear body which is the main radiating surface. This can be done by using material with better damping qualities. Research has been made on this subject and significant successes have been reported (3). Also reducing the area of the gear body (within the strength requirements) would reduce the sound radiated. b) Punching Noise. During the blanking operation a large force is developed, and stress builds up in the stock material, with an equal force in the opposite direction building up in the machine frame. Upon the fracture of the stock the two opposing forces set both the stock material (with the supporting lower die) and the machine frame into transient motion. The noise produced can be reduced by reducing the vibration amplitude, its frequency, or reducing the radiation.



Vibration reduction can be achieved by minimizing the maximum impact force. This was tried by slanting the blanking punch or the die section (Figure 7) with significant reduction of the vibration and sound. Reduction can also be obtained by using materials with higher damping coefficients in parts directly involved in the impacts, namely the stripper plate and the lower die. In tests with plates subjected to impacts similar to those in the press the sound was reduced 14 db by changing the construction material of the plate from steel to steel-lead composition (Figure 8). c) Progressive Die Press Noise. These presses produce their sound energy through the vibration of metal plates upon impact, independent of the punching action. The parameters affecting the vibration and sound radiation are: the velocity of impact, which can be reduced using hard rubber mounts (snubbers); the composition of the vibrating plate, which can be changed to reduce the vibration and noise. This was proven experimentally by obtaining nearly 84% reduction in sound by replacing a plate with geometrically-similar, laminated and more massive plate; the size of the plate, which is reduced will radiate less sound energy than a larger plate vibrating with the same amplitude. One technique is to cut-out areas of the plate that do not perform any necessary work functions.

2) Noise Elimination by Enclosures: A survey of the literature and the commercially available materials as well as practical applications have shown that, a) enclosures can be very effective in limiting the proliferation of machine noise if the necessary design and planning is made to insure nearly complete enclosing. The performance of the building blocks (panels) is usually given for idealized test conditions where they are placed between two adjacent reverberation rooms. The resulting transmission loss should not be used directly to calculate the expected noise reduction from a complete enclosure. b) For large plants, using large numbers of noisy presses the cost of enclosures can add up to very large sums of money. c) The operation of the machine is interfered with when it is enclosed and this may be the single most undesirable factor in the use of enclosures at this time. In addition, they could not be used directly in many plants where machine layouts do not allow sufficient space for separate enclosures. d) In instances where interference with operation and cost problems are sumounted, the use of an enclosure is a very attractive solution to the noise, since 20 to 30 db reduction can be achieved immediately and the efforts are directly visible to all concerned.

CONCLUSIONS

1) The study shows that since it is advantageous to deal with the problem at its roots by aiming at the modification and design of inherently quieter presses and processes, it is necessary to have a thorough knowledge of the noise generation mechanisms in presses. The methods used here are the waveform analysis, frequency analysis and the cross correlation analysis with the objective of identifying the causes of sound generation and selecting the components to be treated. These methods are effective in studying noise sources in punch presses and in a large number of other machines. 2) To reduce vibrations and noise, one can a) reduce the forces exciting the structure, by slanting the die or the punch; b) reduce the response by adding mass or damping. Increased damping is most effective at resonances, and increasing the mass is useful at the higher frequencies; c) break the transmission path from the primary disturbed element of the machine to other radiating members. between the ram and the rest of the machine or between the die and the press frame. 4) Enclosures are good solutions when economics and spacing allow. 5) Barriers can be used effectively to reduce the noise radiated from the press area to the rest of a plant.

REFERENCES

1) R.D. Bruce, 2) R.M. Hoover, and 3) W. Compton - unpublished papers.

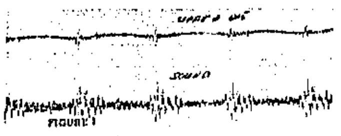
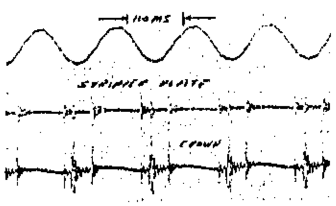
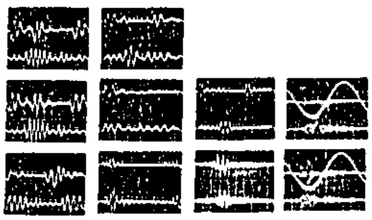


Fig. 1 Waveform of sound and component vibrations





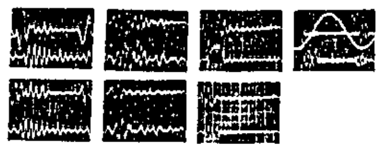


Fig. 2. Filtered signal, columns: 63, 125, 250, 500 Hz
Rows = vibrations of components against sound.

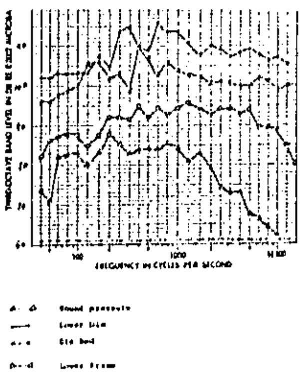
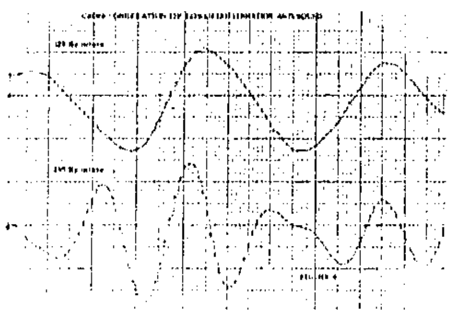
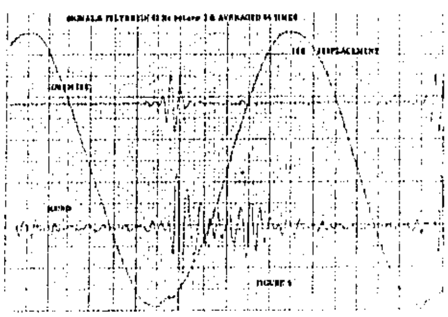


Figure 5. Sound and Acceleration Levels of a High Speed Blanking Press





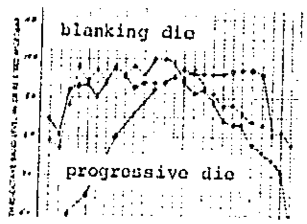


Fig. 6 Sound Analysis

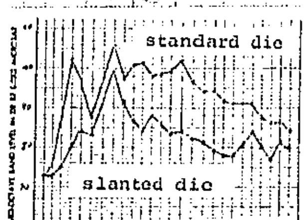


Fig. 7 Sound Analysis

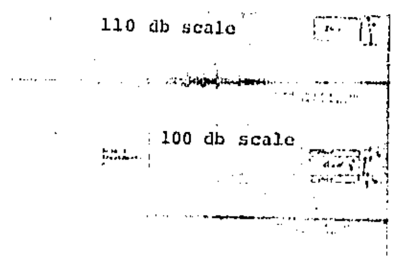


Fig. 8

THE ORIGINS OF NOISE IN HYDRAULIC PUMPS

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ABSTRACT

Results are presented from a fluid dynamic/acoustic research aimed at identification and analysis of noise sources in positive-displacement pumps. The investigation included noise generation in the suction and discharge lines, unsteady volume displacement by the rotating elements, cavitation at low inlet pressure, the influence of large fluid viscosities and of the pump operating parameters: shaft speed, pressure rise, volume flow, etc. All important noise sources were traced down and quantified, with theoretical support in most instances. This work proves that a competent fluid dynamic/acoustic model can be constructed for all the important noise sources in this class of fluid machinery. Further, design optimization, relative to hydrodynamic performance and noise, leads to significantly quieter pumps. This paper is believed to be one of the first basic investigations of noise generation in this class of fluid machinery. The previous literature is devoid of the fundamental understanding needed by the designer to improve the noise characteristics of hydraulic pumps.

INTRODUCTION

The Waukesha Foundry Company builds two types of positive-displacement pumps, the WR and DO, which are lobe-type and lug-type respectively. Despite the fact that these machines have many geometrical details in common and are driven by the same gear box, they are fundamentally different in some of their hydrodynamic and noise characteristics.

It is not the intention of this paper to exhaustively record pumping performance and other features. Rather, our purpose is to point out the fundamental noise and pulsation aspects of the WR and DO and how these are revealed by the pump's "language" and performance.

These two pumps have a rich "language", full of information on their internal hydrodynamics. By understanding the pump's speech more fully, the applications engineer will be able to apply these machines with more precision. This paper is a primer for interpreting the noise or "language" of rotary, positive-displacement pumps. While gear, vane, and the many related types are not considered explicitly here, all of the phenomena discussed occur in them too, with similar noise and pulsation outputs.

FLUID DYNAMIC FUNDAMENTALS OF ROTARY PUMPS

Rotary pumps are nothing more than devices for generating and destroying volume. They are very analogous to reciprocating positive-displacement pumps whose action is much easier to understand.

When the piston of a positive-displacement pump moves down the cylinder, volume is generated between the piston and the head. When the piston runs back up the cylinder, this volume is destroyed. There is a tendency for fluid to fill the newly-generating volume, if there is any connection between the volume and some source of fluid. Likewise, when the volume is being destroyed, the fluid will run out of the volume if there is any egress.

In a reciprocating compressor, the connections to reservoirs are provided through valves, which are essential to the operation of the machine. In contrast, the rotary positive-displacement pump has no valves at all; its valving action is produced in an ingenious fashion. Nevertheless, the rotary pump is basically a device for manipulating volume just like any other positive-displacement machine.

VOLUME MANIPULATION BY THE LUG-TYPE DO PUMP

Pump volume manipulation is much easier to understand for the lug-type (DO) than for the lobe-type (WR). In Figure 1, a DO pump is seen at various rotor positions. New volume growing between the trailing edge of the lug on the lower rotor and the cylindrical surface of the lug on the upper rotor can be seen clearly. And on the discharge side of the pump, it is quite easy to see volume being destroyed between the leading edge of the lug of the lower rotor and the cylindrical surface of the upper rotor.

As volume is made on the pump's inlet side, it tends to fill with fluid from the inlet pipe. And, as volume is destroyed on the discharge side, the fluid that was contained therein tends to flow out to the discharge pipe.

Of course, the newly-made and filled inlet volume in Figure 1 must be transported to the discharge side of the pump without allowing fluid to flush back. This is done by carrying the volume from inlet to discharge against the cylindrical surface of the casing; in this way, the pump's "valving" action is accomplished.

So the entire pumping process of this machine can be described as the continuous creation of new volume on the inlet side, the carrying of this newly-formed volume from inlet to discharge, and the destroying of the volume on the discharge side.

So long as all the new volume is liquid filled, the pump is a "positive-displacement pump" except for a small leakage or "slip". That is, the volume of liquid pumped per revolution should be independent of the pressure rise across the machine. Figure 2 illustrates that indeed this is so for actual machines. The slight slope to the performance curve is caused by leakage past the rotors from the discharge to the inlet side of the machine.

THE LOBE-TYPE (WR) IS SUBTLE

Basically the WR pump generates volume, transports it and destroys it in the same fashion as the DO, but the process of volume generation in the WR is much less obvious than it is in the DO. In Figure 3, we demonstrate the WR's volume generation.

The volume produced by the WR may be envisaged to flow out from the contact between the two rotors. This is hard to see in diagrams of the pump rotors, but proceeds essentially as in the DO. Likewise, on the discharge side of the WR, volume is destroyed in the same fashion. Of course, the volume so made is carried over from inlet to discharge sealed against the casing in the swept-through volume shown in Figure 3c.

Despite these similarities, there is a fundamental difference between the WR and DO; although minor in magnitude, it is important in its consequences. Analysis and tests show that the DO generates volume at a steady rate in time. That is, for every unit of time passing, the DO produces the same amount of new volume on its inlet side and destroys the same amount of volume on its discharge side. This means, theoretically, that the DO should be a "rock-steady" pumper.

In contrast, the WR does not generate or destroy volume steadily. In Figure 4 a comparison between these two pumps is shown. Note that the WR's volume curve oscillates about a mean, while the DO's is straight.

The slight difference between these two pumps in the rate of volume generation has important consequences on pump performance even though the WR's unsteadiness amounts to only a small percentage of the volume flow rate. This small oscillation produces important fluid dynamic effects, which can cause pressure pulsations of several hundred psi.

THE "STEADILY" PUMPING DO AND THE "UNSTEADILY" PUMPING WR

We have asserted that the DO pump displaces volume steadily. While this is true theoretically and can be proven exactly from geometrical analysis, the DO is in fact not a steadily-operating pump. Now why should this be so?

In order to give a satisfactory answer to this puzzling question, we must look closely at just where fluid is trying to go on the inlet and discharge sides of the pump. If the fluid should spread out uniformly into the inlet region of the pump, then one would not expect large pressure changes. If, on the other hand, the fluid rushes through a small opening, the fluid pressure can be severely changed. So we must seek to determine just what sort of inlet flow pattern is actually produced by the DO.

Figure 1 shows the inlet side of the DO as fluid flows in. The lugs of the DO rotors can be regarded as pistons. As these pistons are withdrawn from the inlet region of the pump, they urge fluid to flow in and fill the displaced volume they have evacuated. Figure 1a illustrates the early part of this process when the lug of the lower rotor is just beginning to move out of the inlet volume.

The cylindrical outer surface of the lugs does not retreat from the fluid, but merely slides through it and so does not demand volume to fill evacuated space. Only the trailing surface of the lug demands that something occupy the space it has left. This realization proves to be exceedingly important in understanding the fundamental characteristics of the DO.

One can see in Figure 1a that all of the flow coming into the pump at this particular time is attempting to rush into the little cavity lying between the rear of the lug of the lower rotor and the cylindrical inactive surface of the upper rotor. This little cavity is attempting to swallow the entire flow rate of the pump!

Remember that such pumps may deliver up to 200 gpm; a rather enormous flow must gush through the small gap between the "horn" of the lower lug and the inactive surface of the upper lug. The small gap causes a large change in pressure of the fluid as it attempts to crowd between the lugs.

For example, we can calculate that if the total flow rate of the pump did pass through the tip gap of the size 25 BDO at 1000 rpm, the fluid velocity required at minimum clearance between the trailing tip of the lower lug and the inactive surface of the upper lug would be about 110 ft/sec. To produce this velocity requires a pressure drop (as may be readily calculated from Bernoulli's equation) of 84 psi. In itself, a pressure drop of 84 psi would not necessarily be serious, but liquids may boil catastrophically or cavitate after a pressure reduction.

Every fluid at a given temperature has a critical pressure (p^*) called the vapor pressure, below which the fluid will convert to vapor. If an attempt is made to reduce the pressure below this value, the production of vapor will be copious and the fluid will become more vapor in volume than liquid. It is well known that for water at 70° F the volume of the vapor exceeds the volume of the liquid by some 600 times. So it takes only a miniscule volume of water to evaporate into a large volume of vapor.

The vapor pressure of water at 70° F is 0.36 psia. We calculated above that a pressure drop of 84 psi would occur if all the flow of the BDO rushed through the minimum gap between its two lugs. Simple addition shows us then that the inlet static pressure of the fluid must be more than 84.36 psia if the pump's internal pressure is to be above the water's vapor pressure. For any lower inlet pressure, the fluid will surely reach the vapor pressure in small gaps between the lugs and will boil inside the little cavity. The copious quantities of vapor produced will fill the cavity between the lugs and satisfy the pump's demand for volume.

This action would surely be of academic interest only if inlet pressures were always above 84 psi above the vapor pressure, but these pumps often need to operate with inlet pressure closely approaching the fluid's vapor pressure, as is true particularly when pumping hot liquids. The closer the pump can be made to approach the vapor pressure at its inlet, the better machine it is for pumping hot liquids and from low pressure tanks. Less submergence of the pump below the level of a boiling liquid would be needed; that is, less net positive suction head (NPSH) is required to give satisfactory operation.

A pump which will operate right down to the vapor pressure would be ideal; at the vapor pressure all liquid flow must cease. But, the unfortunate pressure reduction through the gap between the lugs of the DO spoils its ability to achieve ideal pump performance. Actually, of course, the restrictive position of the rotors is only realized for a very short part of the total pumping cycle of the machine, although it occurs four times per revolution in a two-lobe pump. But, despite the short time interval of this malfunction, it has long-term and dramatic effects on the entire system. We shall explore this further.

Now we have seen that if sufficient inlet pressure is not provided (as it rarely is), the full flow of the pump cannot actually get into the space between the two lugs in the position shown in Figure 1a. Since there is no other place for the fluid to go, the normal flow cannot pass into the pump and the pumping rate is severely reduced during this part of the cycle. And in order that the flow rate average about the mean, there must be some part of the cycle when the flow is above the average delivery rate. It follows that the flow rate through the DO must be oscillatory in nature because of the unfortunate tip blockage between its lugs.

Considerable effort has been spent attempting, geometrically and otherwise, to find means of avoiding the tip blockage, but it is tied intimately to the sealing overlap of the machine and cannot be avoided completely. Improvement can be won by optimum choice of geometry, but the basic problem cannot be eliminated or even much alleviated.

While it is incontestible that the flow through the DO must be oscillatory, perhaps that is of little consequence. Why should we be concerned (except in special metering applications) that the inlet flow rate of the DO is not steady?

The reason is just this: there is always a mass of fluid enclosed in any piping system. Forces are required to change this fluid's velocity; the reason follows simply from Newton's Law, $F = Ma/g_o$. A certain change of fluid velocity in a certain time implies a certain acceleration a; to achieve this acceleration, the mass of the fluid M (which may be quite large in a normal piping system) demands a force in proportion to the mass times the acceleration. The needed force can only come from the fluid pressure. So we see that unsteady flow through the pump requires unsteady pressure forces in the inlet and outlet piping.

These pressure fluctuations can have two consequences. First, they can cause hammering and pounding noises in the pipes and, secondly, if the fluid in the inlet piping is near its vapor pressure, the transitory depression of pressure required to accelerate from minimum flow, can lead to intermittent boiling of the fluid. Now if the vapor would disappear as the pressure rose again, this inlet boiling would be of no great consequence (although the vapor collapse is noisy). But large bubbles of vapor are produced which require considerable time to collapse; in fact, an amazingly long time. We have seen such bubbles travel out of equilibrium with the liquid for many feet down piping; rough calculations show that they could travel as far as 1000 ft. in certain circumstances.

Why is this important? The main effect is that the pump will ingest both vapor and liquid, yielding two consequences. First, swallowing vapor has an influence on the liquid pumping rate of the machine since its volume appetite is being partially satisfied with gas. And secondly, this vapor cannot exist in the high pressure discharge of the machine; it must condense and will do so rather violently as the pressure rises from inlet to discharge side. This rather violent collapse of the vapor leads to the familiar and irksome problems of noise and cavitation damage.

So we see that the difficulties compound upon themselves; a restriction in the pump leads to the production of vapor which satisfies the pump's thirst for volume. This causes the flow rate in the inlet line to vary in time, which in turn can depress the pressure in the inlet line to the vapor pressure causing more vapor to form, leading to more unsteadiness in the pump's behavior.

Every engineer who has dealt with lug-type pumps is well familiar with the pounding of their inlet lines. Strangely, the pounding reaches maximum intensity at pressures considerably higher than the critical inlet pressure, where the pump's flow rate breaks off as shown in Figure 5. This oddity is rather perplexing to explain at first.

How can it be that a cavitation noise is more irksome at high inlet pressure than at low? (Of course, as we pointed out above, there is an inlet pressure above which all these effects disappear.) We shall leave at the moment the explanation for this somewhat strange acoustic behavior to later discussion below.

In passing, we note that the WR pump produces no inlet disturbances. This is rather surprising because, as we have said, the WR is theoretically an unsteadily pumping machine. The chief difference between the two machines is that the WR suffers no major flow restriction on its inlet. Its flow geometry is quite "clean" as one can see in

Figure 6 which compares the WR and the DO rotors in their worst inlet flow configurations.

THE DO'S QUIET ZONE

The sorts of noises one hears are illustrated in Figure 7 by the curves showing the sound level and the little diagrams showing the pressure pulsations in the piping. Before the crescendo of noise commences at p*, the DO has a remarkable quiet zone on Figure 7. In this zone, the pump's noise level reaches a minimum. Since most of the pump's noise is due to cavitation, how can it become quieter as the inlet pressure is lowered toward the fluid's vapor pressure?

Figure 7 hints an answer to this puzzling question. We note that above the quiet zone the pressure pulsations in the inlet line of the DO are severe, but these disappear as the inlet pressure is decreased. In contrast, the DO's discharge line pulsations are minor until the inlet pressure goes below p*. So at high inlet pressure, the DO's noise comes from its inlet line, while at low inlet pressure it comes from the discharge line. At intermediate inlet pressure, both lines are quiet--that is, the "quiet zone."

In order for the inlet line to quiet down, flow and pressure oscillations in it must cease, and they do, as is shown in Figure 7. Perhaps this means that somehow the tip blockage effect, which causes all the inlet furor at higher inlet pressures, somehow is alleviated. But this is not right; the tip blockage cavitation actually gets steadily worse as the inlet pressure lowers. What then is going on?

The answer is found in the vapor made in the inlet by boiling. The vapor actually has a good effect which compensates for its bad effects. It does this by acting as a spring or a pulsation damper or capacitor. In other words, once enough vapor is made in the inlet regions of the pump, the fluid is no longer incompressible but can expand and contract under the impetus of small pressure changes.

So instead of fluid being jerked from the inlet line and then abruptly halted by tip blockage, the vapor voids in the inlet can pulse in volume and thereby satisfy the pump's insatiable volume appetite. This odd occurrence is responsible for the "quiet zone." In effect, the pump has made there its own pulsation damper.

Now let's test this explanation on the WR. Since it has no tip blockage and consequent inlet pulsations, they cannot be damped to produce a quiet zone and, indeed, test data demonstrates none for the WR.

THE DISCHARGE PROCESS

So far only the inlet process of these pumps has been discussed and nothing has been said about their discharge. If the discharge should be unsteady in time, severe pulsations will be caused in the discharge piping of the machine.

Right away, we wish to emphasize that discharge pulsations and inlet pulsations are fundamentally and basically different. They occur for different reasons and at different places in the pump's operating map. The applications engineer should learn, as his first language lesson, to distinguish between these, for their corrections differ.

It seems ironical that the WR pump, which has a very steady and smooth inlet process, is the worst actor on the discharge side. And, the DO, which is a very bad inlet actor, has the best discharge characteristics. However, both pumps suffer from an identical discharge malfunction under certain operating conditions as we will explain.

Discharge pulsations, like inlet pulsations, are caused by an unsteadiness in the delivery rate of the machine. As we showed for the inlet, an unsteady flow in the discharge piping system requires large forces to accelerate and decelerate the fluid. These accelerations and decelerations occur four times per revolution in a two-lobe pump and so are reasonably high in frequency. Since the time to change the velocity is short, the acceleration must be quite high.

Any effect which causes an unsteady delivery will lead to pulsations, although some effects produce higher pressure amplitudes than others. In order to calibrate the reader, we have measured pulsations of 250 psi peak pressure in the discharge of the WR, operating at a mean discharge pressure of 10 psi. Similar amplitudes are found for the DO under certain operating conditions. In contrast, inlet pulsations of the DO have been measured as high as 1000 psi at a mean inlet pressure of 12 psia.

The level of these pulsations is by no means inconsequential and can lead to intense noises, mechanical disruption of piping, and fatigue failure of attached devices--not to speak of the effect upon the pump itself, causing the fretting of splines, cavitation damage, torsional oscillation of shafts, fatigue of gear teeth and mechanical noise.

The discharge antics of these pumps may be broken down into two clearly different regimes. First, at <u>high inlet pressures</u>, the flow in the discharge is always liquid. In this regime, the discharge of the DO is completely steady in time, but that of the WR is not. The second regime occurs at <u>lower inlet pressures</u> for both WR and DO when uncondensed vapor is swept over into the discharge region of the pump. We will now consider the first regime.

<u>HIGH-INLET PRESSURE DISCHARGE PROCESSES</u>

<u>The DO</u>

Consider the simpler case of the DO first and then the more complex WR. The DO is, as we have emphasized before, theoretically a steadily-pumping machine. In other words, on both its inlet side and its discharge side, it makes geometrical volume at a steady rate. Now, we have pointed out on the inlet that this is not sufficient to ensure a constant flow rate because the fluid can be ruptured under adverse conditions and is, in fact. On the discharge side, however, the fluid pressure is usually too high to allow any rupture.

Again, we may think of the change of volume on discharge as produced by a piston (the lower lug in Figures 1d, e, and f) moving into the discharge volume of the machine and thereby expelling liquid. As before, the only surface of the lug which causes any displacement is the non-cylindrical leading and trailing surfaces. So one can see in Figure 1 that all the flow produced by the pump is coming out of the little gap between the two lugs.

This fluid flowing through the restriction between the lugs of the BDO needs the same pressure drop at 1000 rpm that we calculated when considering the inlet process--that is, 84 psi. This means that the pressure inside the cavity now must be <u>above</u> the mean discharge by this amount. The 84 psi <u>increase</u> does not cause any change in the physical properties of the fluid as on the inlet side. Only a temporary increase in the torque of the rotors occurs as the fluid is squeezed out through the gap. We have measured these pressures and confirmed the predictions of theory.

Since the fluid coming out of this little cavity is all incompressible liquid, at pump discharge pressures normally used in practice with the DO, the delivery rate of the fluid to the discharge <u>remains constant</u>. The only effect is to produce an unsteady force on the shaft of the machine and to reduce its efficiency. But there is <u>no unsteady</u>

delivery to the discharge except that which is produced by a slight unsteady leakage back to the inlet of the machine caused by the overpressure in the little cavity between the lugs. And this unsteady leakage does not seem to cause any severe discharge oscillations. So while the discharge of the DO is momentarily restricted four times per cycle, this does not cause any pounding in the discharge.

The WR

Now the WR is a different beast for, as we have emphasized above and shown in Figure 4, it does not, in fact, make and kill volume steadily in time. These small oscillations in flow rate would seem to be very inconsequential. Of what moment is an 8% fluctuation of the flow rate of the machine? Well, it proves to be a considerable matter because a sizable mass of fluid must be oscillated in the piping on the discharge in most installations. Even if the pipe on the WR were only 10 ft. long, we can calculate that pressure pulsations of several hundred psi are caused by only a 3% variation in flow rate.

Extensive studies have been made of the geometry of the WR with an aim to minimizing these unsteady effects. In fact, the geometrical properties of the WR type machine have been completely analyzed and can be represented cogently in graphical form. Rotor shapes for the WR have been calculated on computers. At best, the inherent volume-making unsteadiness of the WR can never be eliminated.

There are means contemplated, however, which essentially balance the unsteadiness of two or more WR type machines against one another to give a steady discharge. One simple means is to skew the rotors about their axes so that, at different depths in the casing, the machine is in different parts of its cycle. But, for rotors that are not skewed, the unsteadiness can never be eliminated by changing the rotor shape.

So we see that the standard WR cannot deliver fluid steadily to its discharge pipe and will cause pressure pulsations therein. These pulsations increase in magnitude as the discharge pressure and the shaft speed increase. The WR inherently has a noisy discharge; in contrast, the DO inherently has a quiet discharge.

LOW INLET PRESSURE DISCHARGE PROCESS

At low inlet pressures, the discharge processes of both machines suffer dramatic changes. The change occurs at the "critical inlet pressure" denoted by p^*; it is the most important point on the performance map (Figure 5) and represents the lowest practical inlet pressure of the pumps. Below this point, cavitation damage is severe, greatly attenuating life. There never should be a good reason to operate one of these pumps for any length of time below its critical inlet pressure.

At the critical inlet pressure, the flow rate of the machine starts to decrease rapidly as Figure 5 shows. What causes the liquid flow rate to decrease? Only one thing--the pump is carrying vapor to the discharge in its swept-through volumes. If the swept-through volumes delivered entirely liquid to the discharge (providing the leakage is not high) the flow rate of the machine would be constant, independent of inlet pressure. So the only way the delivery rate can decrease dramatically is for larger and larger amounts of vapor to be present in the swept-through volumes when they open to discharge. This conclusion is incontestible. Before we go on to explain the consequences in the discharge line, we should dwell on the cause of vapor in the swept-through volume of the pump.

As Bernoulli's equation explains, any increase in velocity of the fluid depresses its pressure; if the velocity increase is large enough, boiling commences.

The lower the mean inlet pressure of the machine, the more likely that the minimum pressures within the inlet region will reach the boiling point. Any sharp corners, sudden accelerations, restrictions (as between the DO lugs), rapid opening of new spaces, will all cause marked depressions in fluid pressure. As we lower the inlet pressure, the vapor pressure will eventually be reached at one of these locations. In fact, these unfortunate flow perturbations are so strong in some of the standard machines that vapor is formed even when the inlet pressure is atmospheric. This is particularly true of the WR machine.

Now the mere presence of a small amount of vapor in itself may not be very serious. It may leave characteristic "fingerprints" on the rotors and casing of the pump caused by cavitation damage where the vapor normally collapses. This is true of the WR slightly toward the discharge side of the centerline against the casing and of the DO on the trailing tips of its rotor lobes, for example. But, small quantities of vapor, which collapse before the swept-through volumes open to discharge, do not affect the pumping rate or the discharge hammering of the machine.

Leakage causes the pressure to rise gradually in the swept-through volumes from the time they close to inlet until they open to discharge. This gradual rise in pressure allows vapor ingested into the swept-through volume to collapse before the swept-through volumes open to discharge. On the other hand, if there is more than a critical amount of vapor ingested, the leakage process is incapable of filling up the voids.

So the processes of vapor generation in the inlet must become sufficiently vigorous in order to "flood" the swept-through volumes with more vapor volume than can be filled by leakage. Since the leakage in these machines amounts to only a few percent of their delivery rate, it takes only a few percent, by volume, of vapor in the flow to "flood" the swept-through volumes. Once this occurs, vapor is present in these volumes when they open to discharge. This occurrence is definitive at the critical inlet pressure, p^*, and its effects are catastrophic.

Below p^*, the pump is not making its full flow rate; it is delivering less liquid than its volume displacement. The swept-through volumes appear on the discharge side only partially filled with liquid. The voids, of course, cannot persist in face of the relatively high pressures in the discharge region; something must fill them; that something is liquid taken from the discharge line. So the volume flow rate of the machine is reduced temporarily as liquid from the discharge rushes back to fill the vapor voids. Once they are filled, then the discharge proceeds at the usual rate.

The temporary reduction in discharge flow needed to fill these vapor voids is sudden, occurring immediately when the swept-through volume opens to discharge. The suddenness of the process requires large accelerations of the discharge flow and leads to severe pulsations.

Now it is quite interesting with the WR that the level of these pulsations below p^* is not much larger than those at high inlet pressure caused by the WR's basic geometrical unsteadiness. There are explanations for this in the very presence of vapor in the discharge side below p^* lending a cushioning effect to the discharge process and softening its violence.

In the case of the DO, on the other hand, since it is inherently discharge-pulsation free at high inlet pressures, moving below the critical inlet pressure causes discharge pulsations to appear for the first time (Figure 7). They are of the same magnitude and are caused for exactly the same reason as with the WR.

SUMMARY - DISCHARGE PROCESS

To sum up, the discharge pulsations of these positive-displacement pumps come from two sources--the geometrical displacement unsteadiness (only with the WR) and the delivery of vapor voids to the discharge of the machine. The first problem occurs typically at high inlet pressures although it is theoretically present across the entire operating range. The second problem only appears, and appears rather suddenly, at the critical inlet pressure. As the inlet pressure is further reduced, this kind of discharge pulsation persists and may even become more severe down to very low inlet pressures when so much vapor is delivered that the discharge line becomes so "soft" that it cannot transmit high pressure bangings.

DIAGNOSIS AND CURE

If a DO pump is producing discharge pulsations, then one knows immediately that it is operating below its critical inlet pressure. If a WR pump produces discharge pulsations, one cannot be certain that it is operating below its critical inlet pressure, although there is a definite change in characteristic noise at the critical inlet pressure.

The easiest test which can be applied to establish the cause of the discharge pulsation is to vary the inlet pressure, starting from a high value and working down. With the WR (and the DO too), a characteristic tapping will be heard just above critical inlet pressure. This tapping is caused by the first, and somewhat random, carrying over of vapor to the discharge side of the machine. Just below the first tapping commences a steady roar of discharge hammering caused by the regular delivery of vapor. This point is always distinguishable and serves to separate the two kinds of discharge pulsation phenomena of the WR.

If it is proven that the sweeping over of vapor is the cause of the pulsation, then one knows immediately that the pump is operating below its critical inlet pressure. There is nothing one can do to the pump to correct this short of changing the design of the pump or its shaft speed in order to alter the critical inlet pressure. In practice, one should raise the inlet pressure (perhaps by improving the inlet piping) or reduce the pump's rotative speed, or change the fluid's vapor pressure by changing its temperature.

Some other devices have been developed which can attenuate the severity of pulsations of the discharge and inlet lines. They are pulsation traps or line-isolating capacitors. However, their use is not of much interest below the critical inlet pressure because the internal parts of the pump are subjected to a severe cavitation attack as the vapor collapses on the discharge side. The pump will deteriorate rapidly, even if it does not excite its discharge lines.

On the other hand, the WR produces unsteady pulsations in its discharge lines even at high inlet pressures. Here is a region where pulsation traps are useful in order to prevent unfortunate excitation of discharge piping. This excitation, as is well known to the applications engineer, can cause failure of piping, fatigue failure of associated equipment, and disgruntled customers.

CONCLUSION

We have now covered the most important hydrodynamics of rotary lobe-type and lug-type pumps. All of their small idiosyncracies have not been treated, but those neglected are not often of much concern.

We have not tried quantitatively to relate the pumps' behavior to their design. Needless to say, the quantitative measures of pump performance can be altered, to a degree, by design changes. But the interesting fact is that the basic aspects, which we have discussed herein, still remain.

These pumps speak, through their sounds, a clear language, telling rather well just what is transpiring within them and in their attached piping. If he is not expert already, the pump applications engineer can learn to accurately diagnose the rotary pump's maladies by just listening. We hope this paper will help him in doing that.

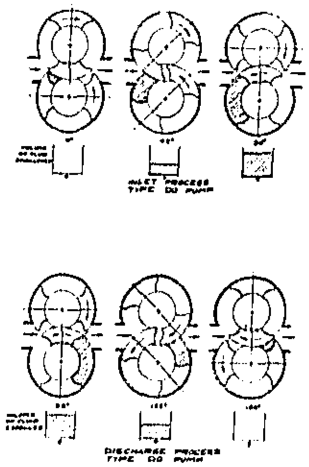
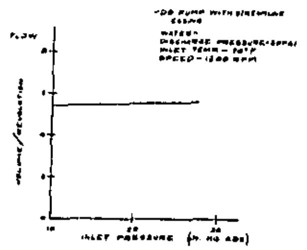


Figure 1



PUMPING CHARACTERISTIC OF DO
(ABOVE CRITICAL INLET PRESSURE)

Figure 2

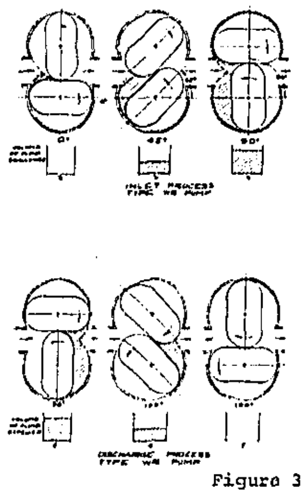
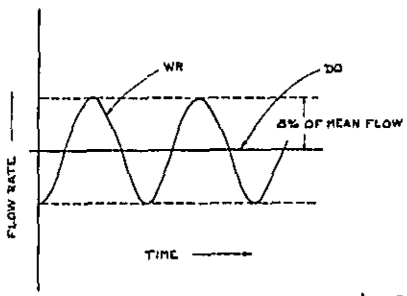
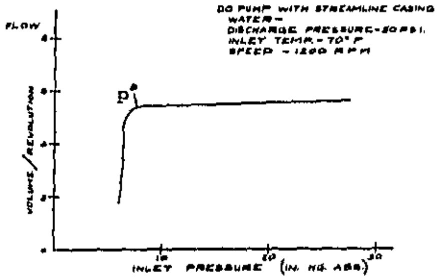


Figure 3



INSTANTANEOUS FLOW RATE OF DO & WR

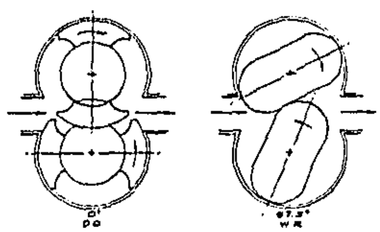
Figure 4



COMPLETE PUMPING CHARACTERISTIC OF DO

p* Indicates critical inlet pressure

Figure 5



DO AND WR AT ROTOR POSITIONS WITH WORST INLET FLOW RESTRICTIONS

Figure 6

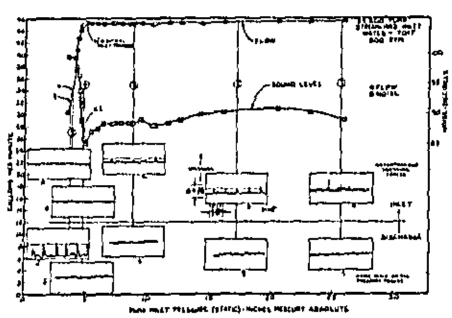


Figure 7

ACKNOWLEDGMENT

For sponsorship of the extensive research investigation behind this summary paper and for permission to publish it, the author thanks the Waukesha Foundry Company, Inc.

THE CORRELATION OF MACHINE STRUCTURE SURFACE VIBRATION AND RADIATED NOISE

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The relationships between structure surface vibration and the resultant noise radiation have always appeared rather unattainable, complex and vague. Anyone who has compared noise with a single vibration measurement knows that there is about a 50-50 chance of them showing any similarity at all! The effect of acoustic environment, directivity, non-uniformity of vibration response and phase over the structure and the complexity of the exact mathematical models all combine to make other methods of analysing the problem more attractive. This paper sets out to show that in considering the problem for one type of structure (load carrying cast machine structures), a simple relationship exists between radiated noise and the mean square surface averaged vibration level.

The vibration and noise of structures is a complex and well documented subject (1, 2, 3), and the interaction of both the vibration and noise of complex structures by classical methods has been studied by Lalor (4), who shows that the methods can be applied but that the computation times involved can become a major problem. Partly to avoid this, the concept of the statistical energy methods has been developed from the work of Smith (5) and Lyon and Maidanik (6). This method assumes that the exact vibration and noise interaction of any structural mode of vibration with a room mode is represented by the properties of an 'average' mode. In this paper the approach lies between the classical and statistical energy methods by using averaged quantities but applying them to simple classical models. Thus the general interaction characteristics can be shown but the exact detail characteristics cannot.

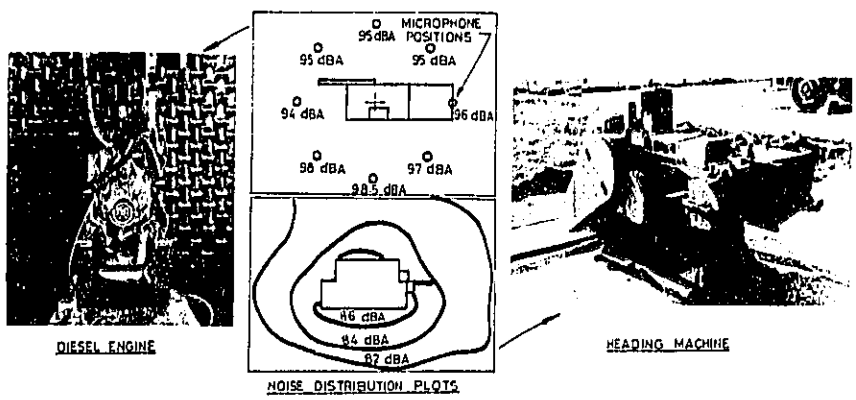

FIGURE 1 - TYPICAL MACHINE INSTALLATIONS

MEASUREMENT OF MACHINE NOISE

There are a large number of national and international standards concerning the measurement of machine noise. Generally the recommended method is to assess the acoustic power output of the machine under anechoic conditions. However, the practical difficulties of this 'ideal' procedure are recognised by the British Standard (B.S. 4196:1967) and the 'near field' method (B.S. 4196:1967) is also recommended. This latter method has been used extensively by the I.S.V.R. both in research and industrial work for the noise assessment of a variety of machines.

MACHINE TEST CONDITIONS

The machines tested have been installed in hard floor foam wedged rooms as shown in Figure 1. Some have been installed out of doors. Where I.C. engines have been measured, air inlet and exhaust noise have been eliminated by piping them away: with other machines compressed air sources etc. have been dealt with similarly. In this way, investigations of the noise radiated by the main load-carrying cast iron/aluminium component surface vibration, has been investigated.

The noise has been assessed at various points around the engine at a distance of 3 ft. The vibration by measurement of vibration acceleration over a rectangular grid with some 60-100 measuring points over the surface (Figure 2). The noise distribution around the machines varies only slightly, as shown in Figure 1.

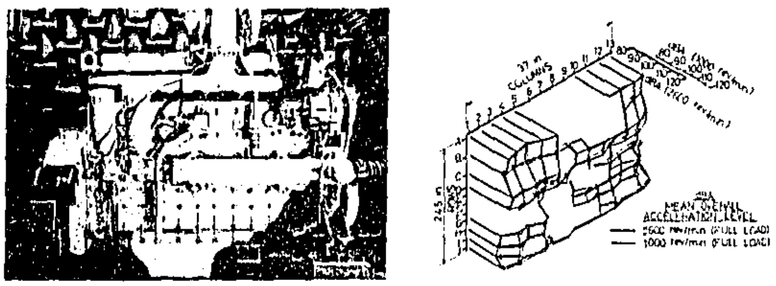

FIGURE 2 - VIBRATION MEASUREMENT GRID AND TYPICAL OVERALL VIBRATION 'PATTERNS'

EFFECT OF ACOUSTIC ENVIRONMENT ON 'NEAR FIELD' MEASUREMENTS

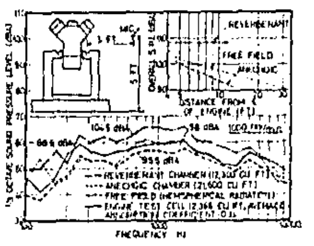

FIGURE 3 - EFFECT OF ACOUSTIC ENVIRONMENT ON 'NEAR FIELD' MACHINE NOISE

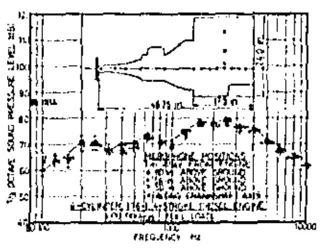

FIGURE 4 - VARIATION OF NOISE OVER MACHINE SURFACE

To examine the effect of the acoustic environment on the noise produced by machine surfaces a diesel engine of about 20 sq.ft. surface area was installed consecutively in the I.S.V.R. anechoic, reverberant and diesel test cell facilities and also on an outside pad. Figure 3 shows the effect on the noise spectrum measured 3 ft. from the engine surface, and also the variation in overall noise with distance. The noise spectrum at 3 ft. is increased greatly under reverberant conditions but for the other conditions little effect is measured in the important frequency range (300 Hz-10 kHz), particularly between the moderately absorptive test cell and the free field (hemispherical) conditions. As the microphone is moved away from the machine surface the effect of the acoustic environment becomes greater.

Figure 4 shows the typical detail variation of noise as the microphone is moved around the surface of an engine installed in the test cell. It is concluded that provided moderately absorbent test conditions apply (average absorption coefficient not less than 0.25, minimum room volume 1500 cu.ft.) then the effect of the acoustic environment on 'near field' measurements taken at 3 ft. distance from the nearest surface is less than 3 dBA.

VIBRATION CHARACTERISTICS OF CAST MACHINE STRUCTURES

All structural vibration is characterised by the presence of resonances (modes) which occur at various frequencies. These modes of machine structural vibration are dictated by the particular form of the machine design, the distribution of mass, stiffness and damping determining their frequency and spacing (1). A detailed analysis of the response magnitude and phase can be obtained for each machine structure mode, but unless only a few modes predominate, this can be a time consuming and difficult task.

Many machines are designed for constant speed operation and consequently the forces exciting the structure cannot be varied. The I.C. engine is a complex variable speed machine and results from these machines can be used to show the effect that varying the exciting forces has on noise and vibration (2). Figure 2 shows the effect of a change in machine speed on the overall vibration acceleration ('A' weighted) of the machine surface. The surface vibration 'pattern' remains the same but the magnitude is increased with the increase in machine speed.

CORRELATION BETWEEN MACHINE SURFACE VIBRATION AND RADIATED NOISE

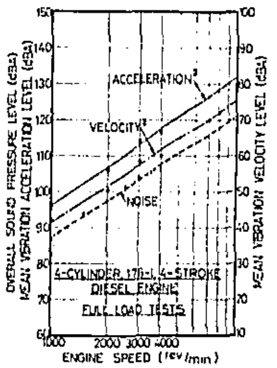

FIGURE 5

VARIATION OF OVERALL NOISE AND VIBRATION WITH MACHINE SPEED

The machine surface vibration can be represented by the single value of average mean square vibration acceleration or velocity of the surface. Figure 5 shows the relation between noise and mean surface vibration for the cylinder block and crankcase of a diesel engine as the engine speed is increased. Figure 6 shows this relationship plotted in more detail as a function of frequency. These results suggest that the comparison of dB(SPL) with either dB (average acceleration) or dB (average velocity) is very appropriate.

Measurement of the vibration and noise produced by 10 different diesel engines (Table 1) varying in surface area from 1000 in^2 to 4000 in^2 has been carried out. Both 1/3 octave band and overall levels were recorded, the vibration being measured over the load-carrying cast structure, the engine block and crankcase. The noise measurements were taken 3 ft from the centre of the engine side surface along a line perpendicular to the surface. The acoustic velocity response, based on sound power flow from a non-directional source into the free field, is presented in Figure 7a for all ten engines. The sound power radiated W(rad), is given by (3).

$$W(rad) = \rho c S(rad)\sigma(rad)\langle v^2\rangle$$

where

$\sigma(rad)$ = average radiation radio for the structure

$\langle v^2\rangle$ = space-time average mean square velocity of the structure ($= \langle a^2\rangle(2\pi f)^2$)

$\langle a^2\rangle$ = space-time average mean square acceleration of the structure

f = Frequency - Hz

Thus for measurements 3 ft from the machine surface; the acoustic velocity response is:

Acoustic Velocity Response = Sound Pressure Level − 20 $\log_{10}$ (Average Surface Velocity)

$$- 10 \log_{10}\left(\frac{\text{Total Machine Surface Area}}{\text{Surface Area of Measuring Sphere}}\right)$$

$$= 10 \log_{10}\sigma(\text{rad}) + 53$$

where reference velocity is: 8.68×10^{-4} in/sec and units are inches and pounds force

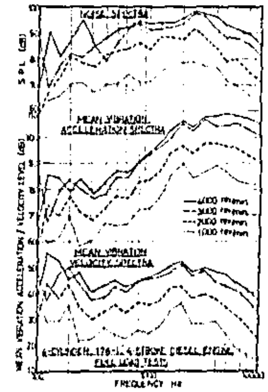

FIGURE 6 - COMPARISON OF MACHINE NOISE AND SURFACE VIBRATION SPECTRA

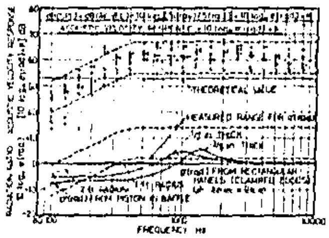

FIGURE 7 - (a) ACOUSTIC VELOCITY RESPONSE FOR 10 ENGINE STRUCTURES

(b) MEASURED AND CALCULATED RADIATION RATIO

TABLE 1 - LIST OF ENGINES

Engine	Symbol	No. of cylinder	Capacity (cu. in.)	Stroke	Total surface area (in^2)	Engine side area (in^2)	dBA (SPL)	dBA (Av. Acc.)	dBA (Av. Vel.)
In-line A	●	6	500	4	3318	1320	92.5	94.0	39.0
" B	♣	6	500	4	3318	1320	94.5	97.5	43.5
" C	✱	6	500	4	3318	1320	91.00	95.0	40.5
" D	+	6	363	4	3520	1082	88.0	94.0	40.5
" E	Δ	6	426	2	4150	1410	98.0	111.0	49.5
" F	■	6	400	2	3400	880	98.5	105.0	48.0
In-line G	×	4	107	4	1220	400	86.5	96.5	42.0
" H	▲	4	154	4	1827	630	85.0	95.0	40.5
" I	▼	4	154	4	1513	630	84.5	94.5	35.0
Vee J	∇	8	568	2	4265	909	98.5	101.5	45.5

The general shape of the acoustic velocity response shows that at frequencies above 400 Hz there is a direct relation between noise and average surface velocity. Below 400 Hz the response decreases at the rate of about 25 dB/decade indicating a direct relationship with average surface acceleration. There is a ± 6 dB scatter which remains constant through the frequency range.

Figure 7b shows the data plotted to show the measured values of the radiation ratio (σrad).

The very general form of the sound power relations prevents the calculation of the exact radiation ratio for any given structure, and a theoretical value can only be obtained for the high frequency region where the value of the radiation ratio reaches unity. This theoretical value is indicated in Figure 7a, and shows that the measured acoustic velocity response values lie in the 12 dB band above the theoretical value. The relation assumes that free-field conditions exist and that the source is non-directional. Any deviation from these assumptions will tend to increase the values of the measured acoustic velocity curve.

RADIATION RATIO

To calculate the radiation ratio of a structure it is necessary to know the detail dimensions and vibration characteristics. This is only generally possible for fairly simple structural elements. Two well established cases are those for the piston-in-baffle and for thin clamped rectangular panels (2, 3, 7). Applying these methods to the calculation of the radiation ratio of the smallest machine tested (total surface area 1220 in^2, side area 400 in^2) gives widely varying results which are plotted on Figure 7b.

The radiation ratio calculated from considerations of coincidence effects in rectangular panels changes from unit value at a higher frequency than the measured data and also shows a lower rate of fall off to the low frequencies. Consideration of the piston-in-baffle case shows a change from radiation ratio unit value in the right frequency range, but again the rate of fall off with frequency is not correct. These comparisons indicate that the radiation ratio for the cast structure appears to depend mostly upon the size of the structure, rather than on any coincidence or wave effects in the surface.

CORRELATION OF OVERALL VIBRATION AND NOISE OF CAST STRUCTURES

Since there is a relatively simple relationship between sound pressure level and average square surface velocity for machines with surface areas in the range 1000 to 4000 sq.ins. then the overall sound pressure level and overall average square surface velocities should also show a simple relation. In particular if an 'A' weighted overall level is used, then the acoustic velocity response of a cast structure can be considered completely independent of frequency.

Since the surface vibration is usually measured by using an accelerometer and converting to vibration velocity, there is a strong case for using overall acceleration level rather than overall velocity. Although there is no direct conversion from overall acceleration level to overall velocity, (it will depend on the spectrum shape) for the range of structures tested, the variation was found to be within the scatter of results. Therefore for convenience, both velocity and acceleration levels are quoted.

If the space-time averaged overall 'A' weighted quantities are used, the following empirical relations for the overall noise ('A' weighted) measured 3 ft from the machine surface can be put forward. The relations are accurate to ± 3 dBA and are valid for machines whose total surface area exceeds 1000 in^2 or cross sectional area exceeds 400ins^2.

(i) Based on machine total surface area

$$\text{dBA(SPL)} = \text{dBA(Average Velocity)} - 10 \log_{10} [S(\text{trav})/S(\text{rad})] + 15$$

or

$$\text{dBA(SPL)} = \text{dBA(Average Acceleration)} + 10 \log_{10} [S(\text{trav})/S(\text{rad})]$$

S(trav) = Surface area of measuring sphere, in^2.
S(rad) = Total structure surface area in^2.
ref velocity = 8.68×10^{-4} in/sec
ref accel'tion = 2.17×10^{-2} in/sec^2.

(ii) Based on machine cross sectional area

$$\text{dBA(SPL)} = \text{dBA(Average Velocity)} + 10 \log_{10} S(\text{side}) + 20$$

$$\text{dBA(SPL)} = \text{dBA(Average Acceleration)} + 10 \log_{10} S(\text{side}) - 35$$

where S(side) = Structure side surface area in^2.

These relations are given in two forms to correspond as closely as possible with the two most common practical considerations - sound power of a machine and the sound pressure level radiated by one surface of a machine. Thus with comparatively little experimental data, the overall characteristics of these structures can be assessed.

APPLICATION TO THE ASSESSMENT OF MACHINE STRUCTURE SURFACE RADIATED NOISE

The relations given in the previous section can be used as a basis for the evaluation of noise radiated from the various sections of a machine. If one structural element is bolted to another, then the average vibration of each element can be considered separately. This provides an alternative method to the lead shielding technique (9) used for assessing the individual contribution of various surfaces to the total noise. Table 2 shows the method applied to the noise radiated from the side of a machine with three separate bolted structures directly facing the microphone. Two of the surfaces had vibration reduction treatments applied to them and the noise was remeasured. The predicted and measured results were in good agreement.

TABLE 2 - CALCULATION OF NOISE RADIATED BY A MACHINE WITH THREE RADIATING SURFACES

	STANDARD MACHINE				MODIFIED MACHINE			
Source surface area in^2.	Average accel'n dBA	Predicted noise at 3' dBA	Total predicted noise at 3' dBA	Measured total noise at 3' dBA	Average accel'n dBA	Predicted noise at 3' dBA	Total predicted noise at 3' dBA	Measured total noise at 3' dBA
120	97.4	83.2	92.0	92.0	85.4	71.2	90.7	90.5
809	91.5	86.5			91.5	85.6		
168	96.0	83.6			92.0	79.6		
Other sources		88.5				88.5		

CONCLUSIONS

For machine noise caused wholly by the vibration of the machine outer surfaces:

1. The noise radiated by machine load carrying cast structures tends to be non-directional when considered in terms of overall levels.

2. There is a relatively simple relationship between the noise radiated by a machine structure and the mean square vibration velocity of its surfaces.

3. The mean square sound pressure radiated is proportional to mean square vibration acceleration below 400 Hz and proportional to mean square vibration velocity above this frequency, for structures with surface areas in the range 1000 to 4000 in^2.

4. The radiation ratio of the machine structures is dependent on the machine size, rather than on coincidence effects occurring in the outer surfaces.

5. The overall sound pressure level radiated by the surfaces of a cast machine structure can be calculated from the space-time averaged overall mean square vibration level. This vibration level can be expressed either as a velocity or an acceleration.

6. The empirical relations for correlating overall noise and vibration level can be used as an additional method for assessing the relative contribution to the total noise of the various structural elements of a machine.

ACKNOWLEDGEMENTS

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PROPAGATION OF MACHINE GENERATED SOUND WITHIN AND AROUND A PROCESS PLANT.

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This paper describes the results of a series of measurements of the way in which sound travels through a plant, and the amount of attenuation it suffers in the process. The original intention of the study was to evaluate the magnitude of any absorption or reverberation effects which occur in a process plant. The total amount of attenuation suffered is that due to presence of the plant, and that due to hemispherical propogation and atmospheric absorption. All the results are plotted as excess attenuation, that left after subtracting attenuation due to distance and atmospheric absorption. The plant used was the recently completed refinery of Chevron Oil Belgium at Feluy. The measurements were made over a period of three nights in conditions of little wind, before plant start up, but at a time when the plant was substantially complete.

NOISE SOURCES

Two noise sources were used, a broad band "continuous" source and an impulsive source. The sound power output of the sources had to be limited to avoid annoyance to surrounding residents, and the sources had to be independant of external power supplies. The broad band source consisted of a pair of chains which rotated inside a steel drum with corrugated sides. The motive power was gravity acting on a weight attached to a cord wrapped round the shaft to which the chains were attached. This source gave a useful sound power output in the octave bands from 250 to 4000 Hz. with a consistency of ± 1 dB between tests. The impulsive source was a sounding board approximately 48 inches by 12 inches which was struck by a wooden hammer propelled by a spring and released by a trigger mechanism. The sources were placed at three positions within the refinery. In two of these positions measurements were made with the sources 3 feet above grade and 40 feet above grade. At the third position, underneath a pipe rack, the low level position only was used. Using two precision grade B & K sound level meters feeding two Nagra III tape recorders simultaneous measurements were made at a position close to the source, to act as a check on the consistency of the source output, and at many positions radiating out from the source. The distant measuring positions were selected so that the effects of sound passing through dense areas of equipment, and areas where there was no equipment, could be assessed. Where possible the more distant positions were well outside the plant area where hemispherical radiation conditions would be expected to exist.

To deduce absolute values of excess attenuation it was necessary to measure the sound power output of the continuous source. This was done in the large reverberation room at I.S.V.R. It was assumed that the mean sound power found in a series of tests there was the same as the mean power generated on site, and the on-site results were all normalised to that power level. The directivity characteristics of the source were measured in the large anechoic room at I.S.V.R.

SITE LAYOUT

The refinery layout at Feluy is shown in Figure 1. This placed some restrictions on the number of measurement positions which could be used. The most satisfactory conditions existed to the south where the ground was flat and approximately at plant grade level. Around the other three sides there is an earth bank rising to approximately 30 feet at its highest point to the east. Whilst this prevented distant measurements at grade it did allow the effect of receiver height to be assessed. Plant plot 1 contains only a moderate density of equipment, and plots 2 and 3 contain a high density of equipment, with rows of pumps beneath the pipe racks and fin fan coolers above. Plot 4 is only of moderate density, but has fin fan coolers above.

RESULTS

The tape recorded results from the continuous source were analysed into octave bands, and the impulsive source results were displayed as oscillograms. The continuous source results were used to compile graphs of excess attenuation against distance, examples of which are shown in Figures 1,2 and 3. The impulsive results were used to deduce the method of sound propagation, and to attempt to deduce an effective reverberation time for the more enclosed parts of the plant.

Figure 2a has been selected to show propagation through the longest path available and indicates the excess attenuation suffered during passage of sound through the dense arrays of equipment, and approximately zero excess attenuation across the spaces between plant blocks. The most distant point shown is elevated approximately 30 feet above grade level, and shows the effect of flanking transmission across the top of the plant.

Figure 2b shows negative excess attenuation as sound is propagated southwards between two plant blocks, indicating a beaming effect due to reflections from large items of equipment, with a resultant negative excess attenuation.

Figure 3 is the result of the source beneath the pipe rack on plot 2 and shows considerable attenuation, given by the passage of sound southwards between two large box furnaces and the reflection of much of the incident sound energy from the ends of the furnaces back towards the source. In the opposite direction, as shown on Figure 4, there is very little excess attenuation while sound travels down the length of the pipe rack, followed by a sharp increase in attenuation caused by the earth bank surrounding the plant. The former behaviour is very similar to that reported in reference 1 for sound in long rooms. The impulsive source was used to measure the reverberation time of the space beneath the pipe rack, which was 0.7 secs. at 250 Hz, 1.0 secs. at 500 Hz, 1.1 secs. at 1000 Hz, 1.0 secs. at 2000 Hz, and 0.8 secs. at 4000 Hz. This space can be likened to a room by assuming that the pipe rack and fin fan coolers above are acoustically solid, and that the room size is effectively the height of the air coolers by the width between rows of pumps on either side by a length equal to twice the width. Assuming an absorption coefficient of 0.02 for the concrete floor, 0.05 for the steel surfaces, and 1.0 for the ends of the room and the free spaces between equipment, the reverberation times indicate that 60% of the sides of the assumed room should be open space. As this is approximately true the room model appears to be reasonably valid. The reverberation times could be useful for estimating the reverberant build up which should affect sound sources such as electric motors located under the pipe rack, but it was found by comparison of measurement of electric motor noise made in a semi-anechoic test shop with measurements of the same motors installed on site that local reflections from objects close to the motors were more important than gross room effects.

Results from the impulsive source at the positions outside the plot 2 pipe rack indicated that any absorption effects which may occur are negligible by comparison with the absorption of the sky above. Reverberant effects were found to be a function of the source directivity pattern and the arrangement of reflecting surfaces around the source. In the majority of positions the only significant sound energy reaching the receiver came from the direct sound and a small number of discrete reflections so it is not surprising that classical room reverberation effects, in which the reverberant sound field is a function only of the room acoustics, (due to multiple reflections) cannot be applied. However, in some receiver positions a block or more from the source the direct sound pulse was negligible by comparison with reflected pulses from major reflectors. In these receiver positions it was often possible to detect by ear which of the large vessels, ducts, etc. were acting as the major reflecting surfaces. The impulsive source tests and the continuous source tests both showed that there was negative excess attenuation, or sound build-up due to reflection, at positions close to the source, but except in the case of sound beamed out southwards, (Figure 2b) there was, fortunately, sufficient excess attenuation to more than cancel out the build up by the time the sound reached measurement positions outside the plant.

By taking all the values of excess attenuation measured outside the plant area, it was possible to deduce the amount of excess attenuation produced by the passage of sound through various densities of equipment. Equipment density was rated in terms of numbers of plant blocks traversed. Detailed results for the 1000 Hz. band are shown in Figure 5. Circles indicate source positions between plant blocks. Squares indicate source positions beneath the pipe rack. Measurement positions in which beaming of the sound took place, such as shown on Figure 2b are included. Diagonal passage of sound through a plant block is considered as equivalent to 1½ blocks of attenuation, and zero represents line of sight propagation. Omitting measurements made with the source beneath the pipe rack it can be said that non line of sight minimum equipment density propagation paths give 5 to 6 dB excess attenuation at 1000 Hz, and most other propagation paths give about 10 dB excess attenuation. Exceptionally high excess attenuations up to 14 dB can arise from propagation between large solid objects such as the box furnaces.

Also shown on Figure 5 are the remote excess attenuations derived in a similar manner for the other octave bands considered in the tests. The curves are similar for 500, 1000, 2000 and 4000 Hz, and considerably lower for 250 Hz, as might be expected, because the typical size of many obstructions in the path of the sound is considerably less than a sound wavelength at 250 Hz. It is also possible to plot remote excess attenuation against distance from the source as in Figure 6, which shows that the height of the source above grade does not appear to be significant, that the presence of the earth bank around three sides of the refinery does not appear significant, both presumably because of the propagation of sound reflected from equipment at various heights, and that the average remote excess attenuation is 10 dB at 1000 Hz, with a scatter of ± 4 dB. Alternatively it can be deduced from Figure 5, making the assumption that a third of the refinery sources suffer minimum attenuation, a third average attenuation and the remainder maximum attenuation, that the average excess attenuation for all sources is about 8.5 dB, for the 500 to 2000 Hz. octave bands, and about 3.5 dB for the 250 Hz. band. This concept can only be applied to the very large number of small broad band noise sources in a plant. Large sources such as furnaces and air coolers could not be expected to behave similarly, nor could sources such as compressors with strong tonal components.

COMPARISON OF RESULTS WITH PUBLISHED INFORMATION.

The O.C.M.A. guide (reference 2) gives a method for estimating the excess attenuation due to varying degrees of screening by the inclusion of a factor K_2 related to frequency and distance. K_2 is said to account for normal in plant attenuation effects plus screening effects external to the plant. For minimum external screening at a distance of 800 feet the values of K_2 suggested are 6.5 dB at 250 and 500 Hz, 4.5 dB at 1000 and 2000 Hz, and 3 dB at 4000 Hz. These are substantially different from the results of the experiment at Feluy.

A second source of data is reference 3, Figure 7, which indicates excess attenuation of 4 dB at 250 Hz, 2.5 dB at 500 Hz, 2 dB at 1000 Hz, 1.5 dB at 2000 Hz, and 1 dB at 4000 Hz at a distance of 800 feet. These include in plant effects and out of plant ground attenuation.

There is obviously a considerable diversity of opinion, and lack of sufficient measured data to allow a firm prediction of in plant excess attenuation to be made. Further experiments on other sites of the type carried out at Feluy would be valuable.

ACKNOWLEDGEMENTS

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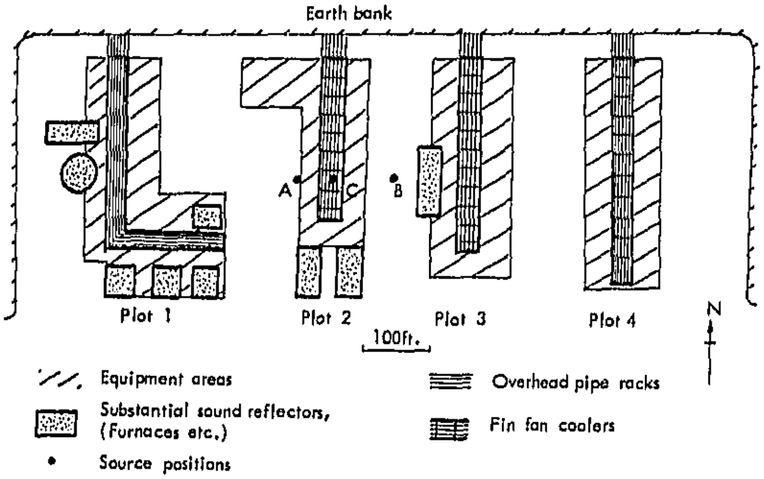

FIG. 1 REFINERY LAYOUT.

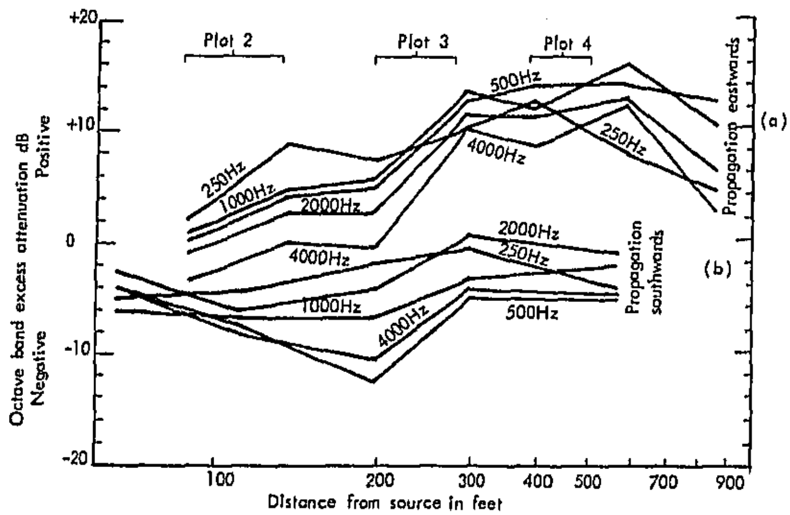

FIG. 2 EXCESS ATTENUATION WITH SOURCE IN POSITION A - 40 FEET ABOVE GRADE.

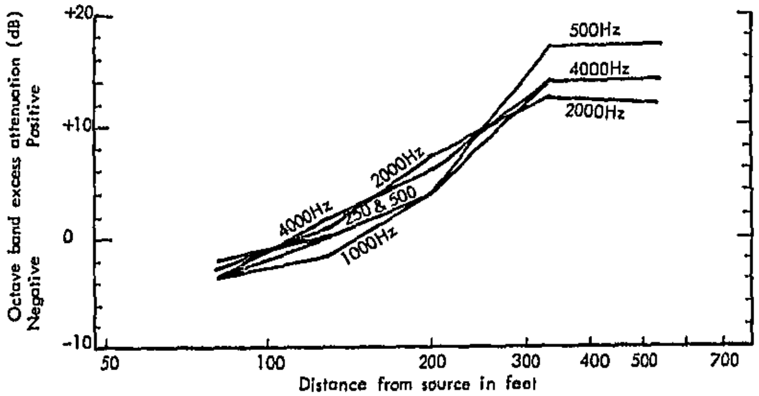

FIG. 3 SOUND PROPAGATION SOUTHWARDS FROM SOURCE AT POSITION C.

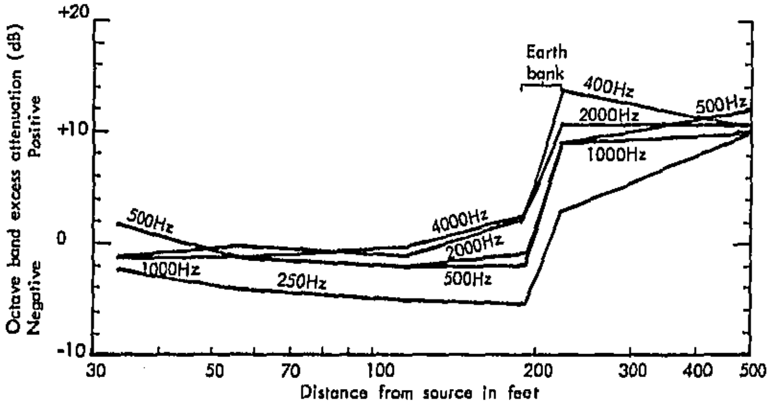

FIG. 4 SOUND PROPAGATION NORTHWARDS FROM SOURCE AT POSITION C.

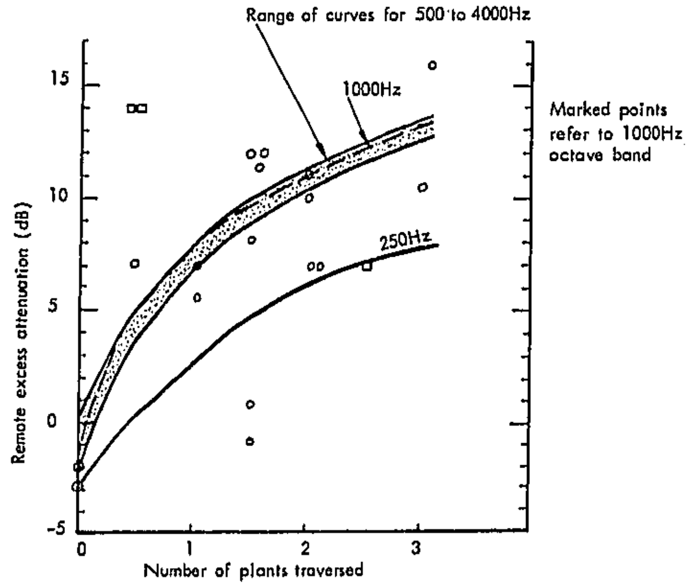

FIG. 5 REMOTE EXCESS ATTENUATION RELATED TO QUANTITY OF EQUIPMENT IN SOUND PATH.

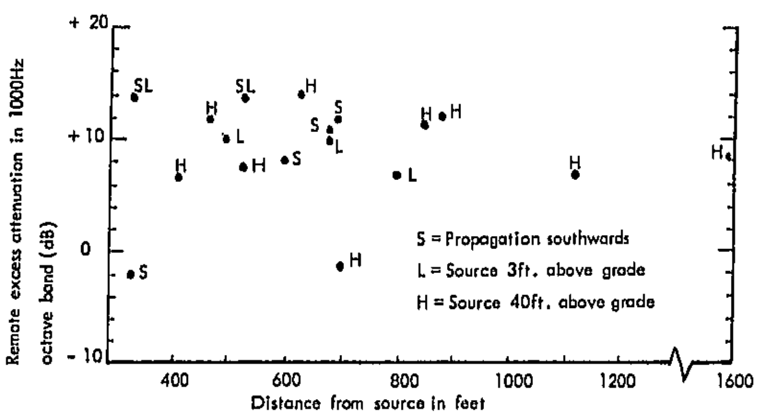

FIG. 6 EFFECT OF DISTANCE ON REMOTE ATTENUATION IN 1000Hz OCTAVE BAND.

ROOFTOP CONCRETE BLOCK HOUSES FOR MUFFLING OF LARGE INTERNAL COMBUSTION ENGINES

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INTRODUCTION

This paper describes the development of rooftop expansion chamber mufflers for large internal combustion engines. These rooftop expansion chambers are made of lightweight concrete blocks covered with a steel top, Figure 1. They have the advantages of being relatively light, easy to install, and very effective.

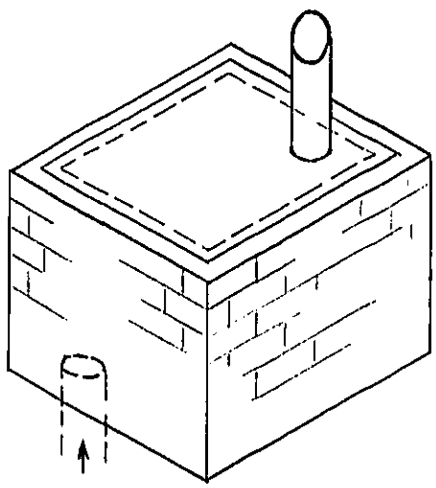

FIGURE 1

CONCRETE BLOCK HOUSE

They have a further advantage in that they can be installed after the engine is in operation. This is extremely useful when the frequencies of the engine exhaust are not known (more on this below), and/or the effectiveness of attentuation between the engine and discharge is not accurately predictable, as was the case of mufflers incorporated in exhaust gas heat exchangers.

The remainder of this paper contains the details of the development, design, construction, and effectiveness of two installations. The first installation was to quiet a set of engines located in downtown St. Louis. The second installation was to optimize the muffling of the exhaust gas heat exchangers by supplementing them with block houses after the engines were operating. In this installation the engine exhaust was not noisy, even without the blockhouse, but the intake noise had set up a standing wave which bothered neighbors a quarter mile away. Concrete blocks were also used to make an expansion chamber on the inlet, reducing this noise appreciably.

INSTALLATION NUMBER 1

Exhaust gas heat exchangers were to provide muffling of the engines, as they had been in an earlier successful application using identical engines. Due to space limitations the heat exchangers were made more compact which caused them to be much less effective mufflers, and the exhaust noise was excessive. At the normal operating load the sound pressure level was 125 dbC, taken parallel to the end and three inches radially out from the twelve inch diameter exhaust pipe. The C scale was used exclusively in order to identify frequencies below 100 hz. After detuning the exhaust pipe length and installing partly satisfactory resonator type mufflers, the sound pressure level was 115 dbC, taken at a location similar to the previous readings.

The decision was made to encircle the exhaust pipes, which now ended 24 inches above the roof, with a wall of Soundblox, a commercially available resonator type concrete block. A further suggestion was made by Vito Cerami, the consultant for architectural acoustics of the installation, to put a top on the wall and use it as an expansion type muffler.

So Soundblox were stacked at a height of 56 inches, approximately a quarter-wave length of the predominant engine exhaust frequency of 90 hz. A plywood cover was placed on top and weighted down. An opening was left at a corner equal to the exhaust pipe area. At a location 9 feet from the installation the sound pressure levels were 100 dbC without the blockhouse and 92 dbC with the blockhouse. This was with a somewhat loose stacking of blocks and a plywood cover.

The removal of one course of Soundblox from the original height increased the sound pressure level 3 dbC. The addition of one course decreased the sound pressure level only 1 dbC, so the original height was made permanent. The interior size was 48 inches by 48 inches.

The blocks were mortared, a steel top and a stack 50 inches long, approximately one quarter-wave length, were installed. The final reading taken parallel to end of the stack three inches out from the twelve inch exhaust pipe was 98 dbC. This was a transmission loss of 115 - 98 = 17 dbC attributed to the blockhouse.

With three engines operating the noise level of air handling equipment for the building and the rattling of adjustable louvers was greater than the engine exhaust. Street traffic was the predominant noise.

This had been a particularly bad acoustic situation because the engine exhaust was through the third floor roof of a low wing of a thirty story office building. The nearby buildings were tall, so the engine exhausts were literally in the bottom of a canyon of surrounding buildings. When the engines were first started a standing wave at 90 hz could be heard in the immediate neighborhood. With the blockhouses installed the engines are barely discernable.

A very interesting side issue in this installation, referred to in the introduction, was the fact that the predominant frequency was 90 hz, while the expected frequency was 120 hz from the 12 cylinder, 4 stroke cycle gas engines. The analysis of why this occurred and the procedure for prediction of engine exhaust frequencies has been presented earlier [1]*.

INSTALLATION NUMBER 2

The experience gained in the first installation required better knowledge of the frequencies present in the exhaust of the engine to be used, allowing better design of muffling in the heat exchangers. The science of the design of muffling into the heat exchangers had also progressed appreciably.

The engine to be used was an eight cylinder two stroke cycle dual fuel engine operating at 720 revolutions per minute, so the predominant frequency ordinarily calculated would be 96 hz. By this time the procedure to calculate the predominant frequencies had been developed and large amplitudes at 48 hz and 144 hz were predicted. The exhaust pressure of an identical engine was recorded and its spectrum determined, which did establish that the predominant frequencies were truly at 48 hz and 144 hz. These frequencies are the 4th and 12th harmonics of the engine speed, while the usual calculation would predict a single predominant frequency at the 8th harmonic, 96 hz.

The heat exchangers were designed to attenuate the predominant frequencies to a level which should be satisfactory. At this time it was decided to build a concrete roof top block house after the installation was on line. In this way exact amplitudes and frequencies could be measured and the block house design could be optimized for attentuation using quarter wave expansion mufflers. Even though it is not easy to build one expansion chamber to muffle more than one frequency, changes could easily be made in the height and area of the concrete blockhouse to achieve effective muffling.

Shortly after the power plant came on line there were complaints that the engines were noisy. A trip to the location to build blockhouses was made. It was immediately apparent that the greatest noise was coming from the engine intakes, and expansion chamber mufflers were made for both exhaust and inlet. Soundblox had been considered but calculations showed that they would not be effective because of the small area exposed, so lightweight concrete blocks were used.

The exhaust spectrum before and after the exhaust block house was built is shown in Figure 2. The height of the muffler was made four feet, which is about one eighth the wave length for 48 hz and three eighths the wave length for 144 hz. The exhaust temperature is approximately 350°F. The overall sound pressure level was reduced from 96 dbC to 83 dbC, and from 86 dbA to 74 dbA, all measured at the same location, ten feet from the finished block house. With the lower noise levels before the blockhouse was built, it is doubtful that this supplementary muffler was needed.

Quieting the intake noise was not the responsibility of the authors, but was an interesting acoustical problem, so attention was turned to it. The spectrum of the intake noise is shown in Figure 3. The frequency of 72 hz was the same as that identified as discernable at an apartment complex a quarter mile away, so attenuation of it was undertaken. The intake filter had a horizontal entry shroud which extended about three feet beyond the

*Numbers in brackets refer to references listed at the end of the paper.

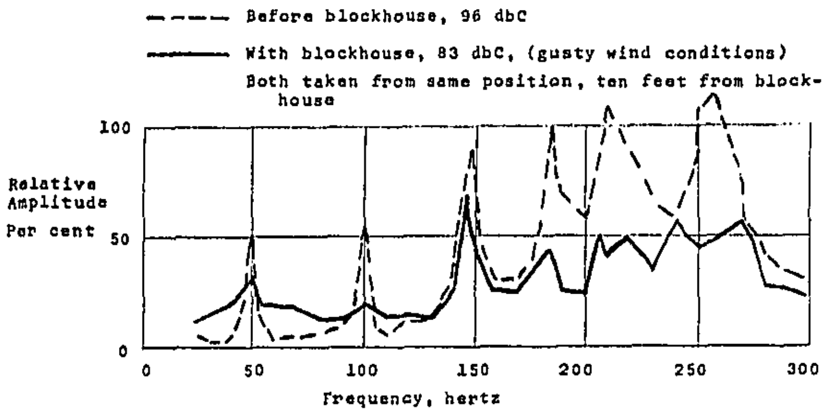

FIGURE 2

SPECTRUM OF SOUND PRESSURE BEFORE AND AFTER BLOCKHOUSE INSTALLATION

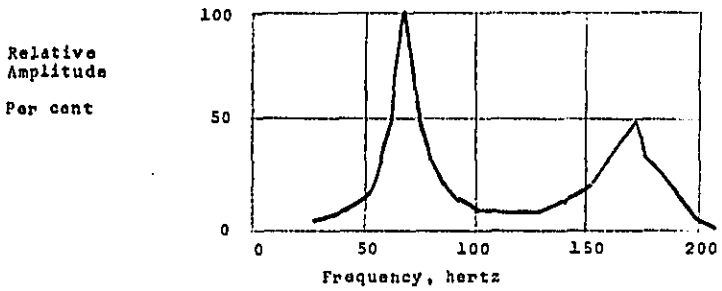

FIGURE 3

SPECTRUM OF INTAKE NOISE BEFORE EXPANSION CHAMBER INSTALLATION

Relative peaks were equal after installation filter, so concrete blocks were stacked in front of this, forming an expansion chamber muffler. Concrete blocks were turned sideways to form the air inlet. Just enough were turned so that the inlet manometer reading at the engine was not affected. The sound pressure level was reduced from 95 dbC to 87 dbC, and from 84 dbA to 78 dbA, all taken 13 feet from the air inlet. This was still a problem because this frequency could still be discerned at the apartments, and was attributed to a standing wave caused by the direct sound from the air inlet and the reflected sound from the one hundred foot long by thirty foot high engine house wall located approximately ten feet from the air inlet.

CONCLUSIONS

1) Concrete block houses are effective mufflers for large internal combustion engines. They can be mortared or strapped with stainless steel straps. Covers can be made of steel or reinforced concrete similar to park bench tops, for weather resistance purposes.

2) The frequency spectrum of the engine exhaust must be known from measurement or calculation. The traditional calculation of firing frequency is seldom correct. Narrow band wave analyzers are needed to measure the spectrum.

3) The intake noise of engines can be a serious and must be considered along with the exhaust noise.

REFERENCES

[1] Diboll, W. B., "The Measurement and Prediction of Exhaust Frequencies in Large Interal Combustion Engines," A.S.M.E. Paper No. 72-DGP-2, presented at the Diesel and Gas Power Conference, St. Louis, Mo., April 16-20, 1972.

ACOUSTIC HOOD DESIGN IN THEORY AND PRACTICE

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ABSTRACT

An existing theory for the performance of acoustic hoods has been shown to conservatively predict the insertion losses of experimental hoods, particularly for broad band noise machines.

1. INTRODUCTION

Apart from Jackson's (1,2) work on unlined acoustic hoods the only truly analytic assessment of acoustic hood performance was given in Junger's work on lined-hoods (3) in 1970.

It is the intention of this brief discussion to compare this latter theory with experiment and to show the effect of using a more accurate measured lining impedance in the theory. The idealized model for the enclosure and enclosed machine is shown in Fig. 1 and will be briefly described.

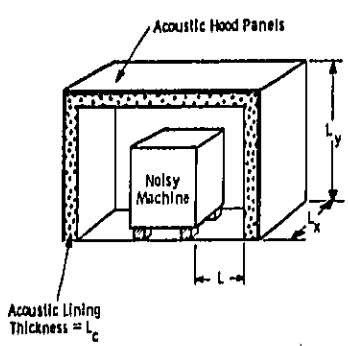

Fig. 1 – Model for acoustic hood and enclosed machine

2. THEORY

Junger has given the following equation for the discrete frequency insertion loss of a lined acoustic hood:

$$\text{I.L.} = -20\log_{10}\left[\frac{64\ (K/M)\,|kL/\sin kL|}{\pi^4(2n+1)^2(2m+1)^2\,|\bar{\omega}_{mn}^2-\omega^2|\,|F(\alpha)|}\right]_{\omega\neq\bar{\omega}_{mn}} \quad (1)$$

The important parameters involved are defined in the appendix. The term $F(\alpha)$, is a function of the lining impedance and is given by

$$F(\alpha) = 1 + i\cot(kL)\coth \Pi(\alpha - i\beta) \quad (2)$$

where α and β are derived from the mechanical impedance, Z, of the hood lining. According to Junger(4), the lining impedance ratio can be approximately expressed as

$$\frac{Z}{\rho c} = 1.6\,L_c + \frac{i\ 2200}{fL_c}\left[1-\frac{f^2}{f_n^2}\right] \quad (3)$$

where

$$f_n = \frac{1.7}{L_c}\ (\text{kHz}) \quad (3a)$$

One should note here that the normal impedance of the hood lining is modeled by an approximation which depends on the frequency of interest and the lining thickness. The validity of this expression will be dealt with later.

Regarding the insertion loss equations, one notes that the insertion loss is, in all cases studied, governed by the fundamental panel mode of vibration. Thus, the theory assumes that the major portion of acoustic energy is transmitted by the fundamental vibration mode of the hood panel.

3. EXPERIMENTAL OBSERVATIONS

Tests were conducted in an anechoic chamber with similar sized hoods using a 3/8 inch electric hand drill and a hydraulic pump as sound sources. Each hood was formed by a 5-sided (6th side being the floor) steel or wooden box isolated from the machine foundation. The steel box panels were 0.035 inches thick and were joined together by bolting them to 1x1x1/8 inch angle iron channels. The wood box panels were 3/4 inch thick plywood and were nailed and glued together. In all cases the hoods were isolated from the machine base and were lined with 1" thick acoustic foam.

3.1 STEEL HOOD

The geared 3/8 inch electric hand drill produced a broad band noise spectrum which peaked near 2000 Hz and fell off at about 10 dB/octave on each side of that frequency. The measured insertion loss of the hood is shown in Fig. 2 for a foam lining bonded to the hood panels. The typical machine-to-panel distance in this case was 9 inches. The sound pressure levels were measured with a 12 Hz bandwidth analyzer.

Also shown in the figure is the insertion loss predicted by Junger's discrete frequency formula, equation (1).

The theory is conservative below, and liberal above 2000 Hz for this case and, on the average, is within 4 dB of the measured insertion loss.

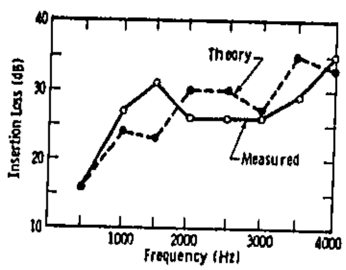

Fig. 2 – Insertion loss for steel hood when enclosing an electric drill

The oil pump used produced a discrete frequency spectrum. The typical machine-to-panel distance was 8 inches. The measured and predicted insertion losses are shown in Figure 3. For this case, the theoretical insertion loss is seen to be 7 dB (on the average) less than the measured insertion loss.

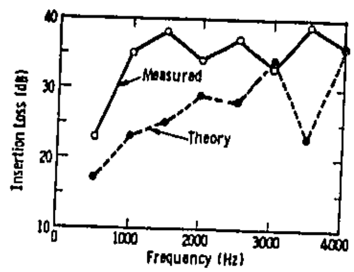

Fig. 3 – Insertion loss for steel hood when enclosing an oil pump

3.2 WOOD HOOD

The typical machine-to-panel distance for this hood was 9 inches when used with the electric drill. The insertion loss is shown in Fig. 4. For this case there is fair agreement between the theoretical and measured insertion loss-over most of range. For 1000 Hz and above, the average theoretical insertion loss is seen to be about 2 dB less than the measured insertion loss. Thus, the theory is definitely applicable to materials other than steel.

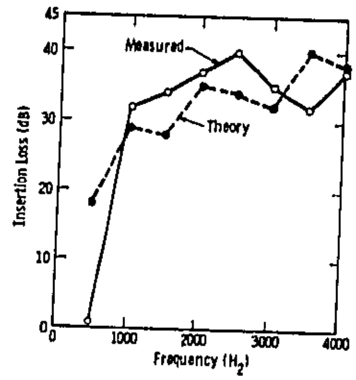

Fig. 4 – Insertion loss for a wood hood when enclosing an electric drill

4. MODIFICATIONS TO THEORY

In an attempt to improve on the theory, the normal impedance of the hood lining was measured and used in place of Junger's approximate value given by equation 3. In this case, slightly better agreement between theoretical and measured insertion loss was obtained. For example, Fig. 5 shows this modified prediction for a 1" foam lining.

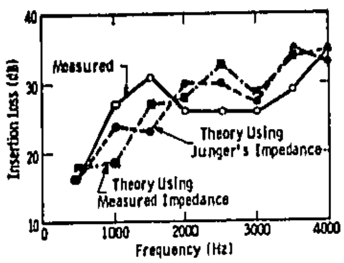

Fig. 5 – Insertion loss for a steel hood when enclosing an electric drill

This result is considered strange because the real and imaginary parts of the impedance calculated from equation 3 do not agree well with measurements as can be seen from Table I. The conclusion is that the theoretical insertion loss is relatively insensitive to changes in the lining impedance, which does not agree with practice.

TABLE I

Real and Imaginary Parts of the Normal Specific Acoustic Impedance for 1-inch thick Acoustic Foam

Freq. (Hz)	R/ρc Measured	R/ρc Eq.(3)	X/ρc Measured	X/ρc Eq.(3)
250	1.7		-7.1	
500	0.89	1.6	-3.7	4.0
1000	0.60	1.6	-1.4	1.4
1500	0.66	1.6	-0.92	.32
2000	1.8	1.6	0.014	-.42
2500	1.1	1.6	0.35	-1.0
3000	1.9	1.6	0.36	-1.6
3500	2.5	1.6	0.28	-2.0
4000	2.8	1.6	-0.21	-2.5

One further test was made on the impedance used in the theory, namely the sign of the imaginary part in equation 3 was made negative. This modification at least makes the approximate impedance have the right sign. Again, no significant change was found in the calculated insertion losses, as seen in Fig. 6.

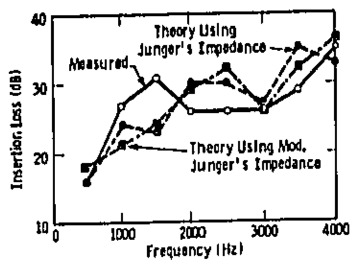

Fig. 6 – Insertion loss for a steel hood when enclosing an electric drill

5. SUMMARY

The theory was found to generally predict conservative insertion losses and was markedly better at predicting insertion losses for hoods covering broad-band noise machines. The accuracy in such cases is good enough for the theory to be used as a regular design tool for acoustic hoods.

If any improvements in the theory are contemplated they should be directed at improving the insertion loss prediction for machines emitting discrete frequency noise.

REFERENCES

(1) R. S. Jackson "The Performance of Acoustic Hoods at Low Frequencies", Acustica, Vol. 12, 1962, pp. 139-152.

(2) R. S. Jackson "Some Aspects of the Performance of Acoustic Hoods", Journal of Sound and Vibration, Vol. 3, 1966, pp. 82-94.

(3) M. C. Junger "Sound Transmission Through An Elastic Enclosure Acoustically Closely Coupled To A Noise Source", A.S.M.E. paper no. 70-WA/DE-12, 1970.

(4) M. C. Junger, Personal Communication.

APPENDIX

(all linear dimensions in inches)

c = sound velocity in air

f = frequency, Hz

h = panel thickness

k = wave number ω/c

k_s = structural wavenumber of panel mode $(k_m^2 + k_n^2)^{1/2}$

m,n = 0, 1, 2, . . .

K = cavity stiffness = $\rho c^2 L_x . L_y ./L$

L = machine to panel distance

L_x, L_y = panel side lengths

M = static mass of panel = $L_x . L_y . \rho_s . h$

ρ, ρ_s = air and panel material density

ω = circular frequency, rads./sec.

$\bar{\omega}_{mn}$ = "kettledrum" natural frequency of panel

PROGRESS IN SUPPRESSING THE NOISE OF THE PNEUMATIC ROCK DRILL

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INTRODUCTION

The Coal Mine Health and Safety Act of 1969, Public Law 91-173, states in Section 206 that "on and after the operative date of this title the standards on noise prescribed under the Walsh-Healey Public Contracts Act...shall be applicable to each coal mine and each operator of such mine shall comply with them." The Act further stipulates that "In meeting such standard under this (noise) section the operator shall not require the use of any protective device or system, including personal devices, which the Secretary or his authorized representative finds to be hazardous or to cause a hazard to miners in such mine." Hence, the Act is considered to be restrictive in the possible use of ear muffs and plugs to provide ear protection, and emphasis is placed on the reduction of operational noise to acceptable levels. In the mining industry the percussion drill is rated as one of the severest noise hazards. In 1971 an environmental noise survey in 21 coal mines by the Bureau of Mines showed that pneumatic rock drilling for the installation of roof bolts exposed individuals to the highest sound levels observed--a range of 104 to 118 dBA.[1] In hard rock mining the pneumatic rock drill is used worldwide and even higher noise levels are commonly sustained for most of the operators' shift time.

Task. In implementing the Act through a program of research on coal mine noise problems, the Bureau assigned to the Rolla Metallurgy Research Center the task of modifying the rock drill to reduce its noise level to acceptable limits. The use of damping alloys and noise suppressing materials was stressed, wherever feasible, in external covers and muffling devices.

Previous efforts to reduce the noise of the pneumatic percussion drill have been made, from time to time, by enlightened mine managers, by equipment manufacturers, and by Government agencies. As a result, several types of mufflers can be purchased to attach to drills and some drills come equipped with integral mufflers.[2,3] Noise suppressing kits to enclose drills, exhaust vents, and drill steel have been devised and marketed, and they are used to good effect in some urban construction work.

Early Rolla Work. In the early 1960's a research group at Rolla under W. C. Miller made a study of the magnitude and significance of the noise from pneumatic rock drills. In their first detailed report published in 1963[4] they identified the major noise sources as (a) the impact noise from the percussion train, particularly from the impact of the piston on the drill steel; (b) the exhaust noise; and (c) the sonic resonance set up in the drill steel and other metal components. Two subsequent reports by Miller, Chester, and DeWoody, in 1964, reflected their concern with the exhaust noise as the major problem. The first report[5] described electrical analogy studies and how they were used in designing an efficient reactive muffler for rock drills; the second[6] presented practical results of applying these methods in the construction of mufflers. They concluded that a comparatively efficient muffler could be incorporated in the shell of a hand-held drill without increasing the weight or size of the drill to an objectionable extent. It was found that the shape of the cavity of the reactive element is not critical in a muffler of this design.

ANALYTICAL AND EXPERIMENTAL APPROACH

In essence, the objective assigned to the project was to produce in 1 year a prototype drill that would meet the Walsh-Healey standards. This could only be accomplished by working with the available drills, making such modifications and component substitutions as were possible in the limited time. At the outset the principle was adopted that functional performance requirements and operator acceptance would have major consideration in the planning and adoption of modifications to the existing equipment.

Functional Requirements. From the standpoint of function, it was accepted that the modified drill should operate without serious loss of drilling speed or efficiency. It should not overheat or ice up to the extent that it becomes inoperable. It should be rugged and durable enough to stand the normal rigors to which drills are subjected in mining applications without excessive breakage or maintenance.

Operator Acceptance. This factor is an important consideration because the economic incentives for production by the individual tend to outweigh the incentives to protect the ears. High noise levels are generally equated with high production and the natural aversion to a quiet tool that this engenders would be heightened by other features considered objectionable in a modified rock drill. Therefore, points to be stressed in the designs included minimum added weight; avoidance of projections, particularly mufflers and other devices that could catch and break off; no interference with operating controls; no maintenance, such as replacement of muffler liners; and capability of taking the same abusive treatment as the standard drill.

Method. The approach taken in the investigation was to make major noise reduction efforts in each of the three distinct noise categories that had been identified, namely, the exhaust noise, the machinery (impact) noise, and the drill steel resonance noise. It was apparent from the beginning that, to reduce the overall noise level of the drill to 90 dB, each of these sources would need to be suppressed well below 90 dB, a formidable task. It was also anticipated that many lesser noise sources that were undetected at high noise levels, say 105 to 110 dB, would become prominent as problem noise sources at 90 to 95 dB. These considerations made it essential that adequate sound measurement and analysis equipment be available and that methods be developed to isolate and evaluate, insofar as possible, the separate noise contributions of exhaust, machinery, and drill steel.

Experimental Facilities. The sound laboratory was provided with a portable sound level survey meter, a sound and vibration analyzer with graphic level recorder, laboratory and portable tape recorders, a real-time spectrum analyzer and digital integrator with X-Y plotter, and accessories for making integrated applications of these devices. The equipment is housed in a laboratory constructed within a large roofed enclosure, so that drilling experiments can be performed under shelter adjacent to the sound lab. Compressed air at 70 psi is provided from a remote compressor and for pressure regulation a ballast tank was installed at the test site. Granite blocks, about 3 feet on a side, were obtained for drilling tests simulating the mine environment, and noise measurements of the drilling operations were made in accordance with the standard method prescribed by the Compressed Air Gas Institute.[7]

An available Atlas Copco BBC-16* drill was selected for use in the prototype because it is convertible to a muffled version, BBC-17, by substituting a patented silencing cylinder for the conventional one. This cylinder releases the exhaust through a ring of small holes into an expansion chamber, a modification which, according to the manufacturer,** effects a 75-percent reduction of the exhaust noise. The drill weighs 59 pounds, has a piston diameter of 2.75 inches and a stroke of 2.125 inches. At an air pressure of 85 psi the BBC-16 is rated at 2,300 impacts per minute, and the BBC-17 at 2,200.

For evaluating the noise contributions from each of the major noise domains, special methods were devised. For example, to eliminate the noise from the drill steel a test was used in which a short stub of drill steel was inserted in the rotation chuck, and the exposed part was enclosed in thick-walled rubber tubing. This stub was inserted into a large lubricated, close-fitting hole in a large rubber block. The drill could then be operated in the normal way, for brief periods, with virtually no noise to contend with from the drill steel. When an exhaust hose was added, this setup permitted an analysis of the levels and the frequency spectrum of the noise emitted from the drill machine components and radiated from the case.

An acoustical enclosure was also provided in which a drill could be suspended, with no drill steel in place; the exhaust was vented outside through a pipe. The high transmission loss of this enclosure virtually eliminated the drill noise for studies of the effectiveness of mufflers on the exhaust noise of the drill.

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* Reference to specific equipment is made for identification only and does not imply endorsement by the Bureau of Mines.

**Manufactured by Atlas Copco, Inc., Hackensack, New Jersey 07602. Patented.

MODIFICATIONS AND INNOVATIONS

The course chosen for the investigation was to develop the best suppression that could be devised for each of the noise domains, and then to assemble these modifications and innovations into a prototype model of a quiet pneumatic rock drill.

Exhaust Noise. Laboratory tests confirmed that the Atlas Copco Model BBC-17 drill, with the patented built-in muffler, had a lower noise level, by about 6 dB, than the standard Model BBC-16. In Table 1 are presented some observed sound levels for these drills as various expedients were employed to reduce the noise.

TABLE 1. - Sound Pressure Levels. BBC-16 and BBC-17 drill, various conditions at 70 psi

Drill and condition	Sound pressure level, dB
BBC-16 drilling in granite	122
BBC-17 drilling in granite	116
BBC-16 with no drill steel in place	117
BBC-17 with no drill steel in place	110
BBC-17 with no drill steel in place, exhaust hose attached	100
BBC-17 with no drill steel, exhaust hose attached, front end plugged	90

The use of an exhaust hose as a silencer in the mine is not considered to be an acceptable practice for the model drill, and the need is apparent for an integral muffler to reduce the exhaust noise by 15 to 20 dB from the initial conditions shown in Table 1. It was evident, for the prototype development, that it would be advantageous to start with the BBC-17 drill because of the effectiveness of the muffler that was incorporated in the case with only a slight increase in size and none in weight. Additional muffling built into an acoustic cover or case to enclose the entire drill was chosen as the logical modification for the quiet drill. Either a reactive muffler or an expansion type could be used, or a combination of them. A muffler study was assigned to M. G. Barth, employed on the project as a graduate fellow, at the University of Missouri - Rolla, the objective being to design a practical muffler for a pneumatic drill. Barth's work resulted in design of a modified expansion chamber which reduced the exhaust noise of a drill from 113 dBA to 87 dBA. Back pressure and icing characteristics of this muffler were assessed as excellent.

An experimental muffler of this type was built to attach to the BBC-17 drill, and a reactive muffler modeled after the design data of Chester, Miller, and DeWoody[6] was also made to fit this drill. In comparative tests the data in Table 2 were obtained, showing that the reactive muffler is more effective in reducing the exhaust noise of this "premuffled" drill. This can be explained, in part, by its better suppression in the frequency range 200 to 500 Hz. Therefore, the reactive design was selected to be incorporated in the prototype case.

TABLE 2. - Comparison of Drill and Exhaust Noise. BBC-17 drill on rubber pads, expansion chamber or pi-type muffler in experimental case, alloy or steel chuck, 70 psi

Experimental conditions	Noise level dB	Noise level dBA
Expansion chamber muffler, steel rotation chuck	109	99
Expansion chamber muffler, alloy rotation chuck	106	98
Pi-type reactive muffler, steel rotation chuck	98	95
Pi-type reactive muffler, alloy rotation chuck	97	93

Machinery (Impact) Noise. Two innovations were undertaken to suppress the noise radiated from the drill machine proper. Most of this noise is generated by the impact of the piston on the end of the drill steel, which causes resonance to be set up in many of the parts and in the external case. One suppressive measure was to substitute energy absorbing alloy components where possible to quell the impact shock waves; the other measure has already been mentioned--the use of an exterior case with high noise absorption characteristics.

Damping Alloy. Alloys consisting primarily of manganese and copper are noted for their ability to damp vibration, and their mechanical strength is comparable with that of mild steel.[9] The energy they absorb is converted to heat and the alloys lose their damping capacity at temperatures in excess of 125° C. Attempts by Miller[4] to substitute a manganese-copper piston for the steel one were successful from the standpoint of noise reduction, but heating of the piston caused it to expand to the point of jamming, and to lose its effectiveness as a damper. It may not be feasible to substitute high-damping materials directly in the percussion train of a pneumatic drill because of the severe stresses encountered and the intense localized heating that can occur because of the damping which converts mechanical energy to thermal energy. The rotation chuck, on the other hand, is a strategic component of the drill that is not in the percussion train and it was selected to be made from the alloy. The rotation chuck encases the volume in which the piston strikes the end of the drill steel. It was postulated that the damping alloy would absorb much of this shock and noise. The rotation chuck is fitted at one end with a hardened steel bushing into which the drill steel is inserted, and at the other end with a brass spline nut in which the male spline of the piston slides. Through the spline, rotation is transmitted to the chuck and thence through the hexagonal bushing to the shank of the drill steel. The stresses on the chuck are not as severe as those imparted to other components of the drill which are composed of steel of exceptional strength and hardness. The alloy selected for testing is a patented casting alloy called Sonoston* which was cast in approximately the shape of the chuck and machined to the proper dimensions.

The damping alloy rotation chuck proved to be a very effective suppressor of noise, more effective than any external covers that were employed to suppress the radiated noise from this region of the drill. An alloy air valve has also been in use in the BBC-17 but a reduction in noise by this change from a steel valve has not been detected.

Noise-Suppressing Cover. The features desired in a case, besides the high absorption for noise, are numerous and varied. It must conform well to the mating surfaces of the drill, because small air leaks are noise sources. It must be rigid, hard surfaced and durable, capable of taking hard knocks. To prevent either icing or overheating it is desirable that it be thermally conductive. To find the novel combination of materials to construct such a case required the selection and testing of numerous contemporary commercial products, and the following design was chosen for the prototype. First a sheet of 1/2-inch-thick, flexible aluminum honeycomb (foil thickness 0.0019-inch) was cut and shaped around the drill, and around modeling clay buildups of resonance chambers for the reactive muffler. The honeycomb cells were then all filled with viscoelastic material, either a liquid rubber, Flexane 85,** or a pour-in-place polyurethane, Eccofoam VIP***. To impart additional rigidity and a hard cover surface, this honeycomb layer was then coated with a commercial plastic aluminum (epoxy type) in which was embedded expanded aluminum metal sheet. Mating surfaces of the split halves of this cover and where the cover contacted the drill were formed of liquid rubber, which made a very tight joint when the case was clamped to the drill. Two vent tubes 10 inches long and 5/8-inch ID were inserted at the extremity of the case where the reactive muffler cavity was exposed to the air. In Table 2 are listed the noise levels observed when the BBC-17 drill was operating on the rubber test pad with the type of cover described above. The pi-type reactive muffler has an advantage of 4 or 5 dB over the modified expansion chamber. The alloy rotation chuck accounts for a further reduction of 1 to 2 dB over the steel chuck.

Drill Steel Noise. The resonance noise from a standard 4-foot chisel bit drill steel is of the same order of magnitude as the exhaust noise, and it is essential that it be controlled. This noise has a bell-like quality, and tones at the frequencies of the transverse vibration of the steel dominate the sound frequency spectrum (fig. 1). By the criteria used to develop the prototype drill, it was decided that the treatment of the drill steel should be an integral part rather than a detachable cover or separate item of equipment. Although it had been reported that a manganese-copper insert had been effective in reducing the noise of a short paving-breaker tool,[10] it did not appear feasible to line a long, hollow drill steel with the alloy. Experiments with rubber wrappings and coatings on the exterior of the steel showed promising noise reduction and out of them evolved the constrained-layer damping treatment. A thin-walled metal tube is slipped over the drill steel, its diameter such that an annular space of about 1/8-inch thickness is left between tubing and drill steel. This space is then filled with a viscoelastic material that adheres well to both components and is tough enough to withstand the violent vibration during drilling. This material must also be soft and pliable enough to absorb the vibrations of the drill steel rather than transmit them to the outer tube where they would be radiated into the surroundings. In the prototype model, a thin-walled steel tube is used, 1-3/8-inches OD with 0.065-inch walls. Of a number of commercial rubber and plastic materials tested for the viscoelastic filler, two were selected. One is a liquid rubber that hardens

* Stone Manganese Marine Limited, London S.E. 7, England.

** Devcon Corporation, Danvers, Mass. 01928.

***Emerson and Cuming, Inc., Canton, Mass. 02021.

to durometer 35 (Flexane 35), and the other is pour-in-place (syntactic) polyurethane foam (Eccofoam VIP). The real-time spectra of noise from drilling in granite with the prototype quiet drill (figure 1), show in graphic detail the reduction in the noise associated with the drill steel resonance after the VIP foam-filled damper was installed on the drill steel.

PROTOTYPE AND PERFORMANCE

The prototype quiet drill, which has been delivered for evaluation to the Pittsburgh Mining and Safety Research Center of the Bureau of Mines, is an Atlas Copco Model BBC-17 (muffled version) pneumatic rock drill with the following modifications:

A. The main air valve and the rotation chuck, normally of tool steel, are made of high-damping manganese-copper alloys.

B. The drill is encased in a highly efficient noise suppressing cover which also encloses a reactive muffler for the exhaust noise.

C. The standard 48-inch-long chisel bit drill steels have been modified by the addition of constrained layer dampers to suppress the resonance noise.

The effectiveness of these modifications in reducing the operating noise of the complete drill assembly is evident from an examination of Table 3, which shows that in tests at air pressure of 70 psi the sound pressure level of the original BBC-16 drilling in granite is 121.3 dB and the level observed with the prototype quiet drill under the same drilling conditions is 102.7 dB. On the "A" scale the noise level has been reduced below 100 dB. Sound levels measured when the drill was operated with air pressure of 85 psi (Table 3), are not significantly higher.

TABLE 3. - Noise From Drilling in Granite. Air at 70 and 80 psi, microphone centered on drill at 1 meter, 4' chisel-bit drill steels, average of three drilling tests

Test No.	Air at 70 psi Sound level, dB SPL	Air at 70 psi Sound level, dB "A"	Air at 85 psi Sound level, dB SPL	Air at 85 psi Sound level, dB "A"	Type of drill BBC-16	Type of drill BBC-17	Type of drill Prototype	Rotation chuck Steel	Rotation chuck Alloy	Drill steel Standard	Drill steel Damped[1]
1	121.3	115.0	120.5	116.5	*			*		*	
2	116.5	112.0	117.7	113.0	*			*			*
3	117.3	113.5	118.5	113.5		*		*		*	
4	110.2	106.5	111.0	107.5		*		*			*
5	117.5	111.0	117.2	113.0			*	*		*	
6	104.2	99.5	106.0	101.5			*	*			*
7	115.2	109.0	115.5	111.0			*		*	*	
8	102.7	98.5	104.2	99.5			*		*		*

[1]Constrained layer of Eccofoam VIP.

Charts comparing one-third octave band analyses of runs corresponding to the tests shown in Table 3 at 70 psi are included as figures 2-4. Figure 2 compares tests 1 and 8, the original standard drill and the complete prototype; figure 3 compares tests 6 and 8, demonstrating the effect of changing from the steel rotation chuck to the alloy chuck when all other improvements are already completed; the 1 dB improvement is obtained, for the most part, at frequencies above 400 Hz. The noise reduction resulting from the substitution of the constrained-layer damped drill steel for the standard steel appears in figure 4, and again it is seen to occur most dramatically in the upper frequency ranges. This corresponds to tests 7 and 8.

The appearance of the prototype drill (figure 5) is clean-cut and smooth, although slightly bulkier than the normal drill. The operator acceptance factor is believed to be very satisfactory. The weight of the drill is increased 9 pounds by the case-muffler enclosure. From the performance standpoint, there is some loss of drilling rate, mainly associated with the damping system on the drill steel. In field tests on early models of the prototype drill, it was observed that the adherence of the damping medium to the constraint tube and to the drill steel is an important consideration in the choice of materials. Also, it appeared that the hardness or elasticity of the medium influenced the loss in drilling rate. In the laboratory, the drilling rates of the BBC-17 drill and the final prototype were compared by a series of 2-minute tests of the penetration in granite of the drills with identical gross loads on the chisel bits. The results (Table 4) reveal that the addition of the case to the BBC-17 drill has little effect on the drilling rate, but the use of the damped drill steel results in a loss of more than 28 percent in penetration rate.

This research is being continued with the aim of reducing the noise level of standard rock drills to 90 dBA. Investigation of the factors in constrained-layer damping of the drill steel that can be adjusted to cause minimum loss in drilling rate will be stressed. Further substitution of damping alloys in the drill mechanism is also being actively pursued.

TABLE 4. - Drilling Rates. Steel rotation chuck, 4' chisel-bit steels in granite, 70 psi

Drill steels	BBC-17 drill		Prototype drill	
	Drilling rate, in/min	Decrease, pct	Drilling rate, in/min	Decrease, pct
Standard	10.9	0.0	10.7	1.8
Damped	7.8	28.4	7.8	28.4

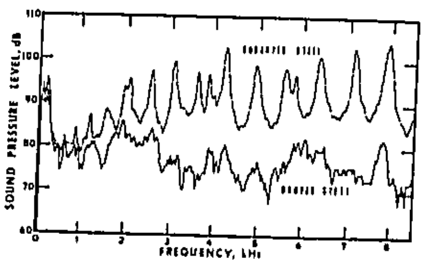

Fig. 1- Noise spectra: drill steel effects

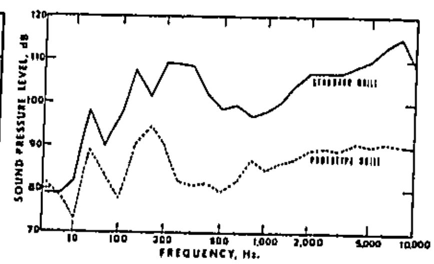

Fig. 2- Drill noise: prototype vs standard

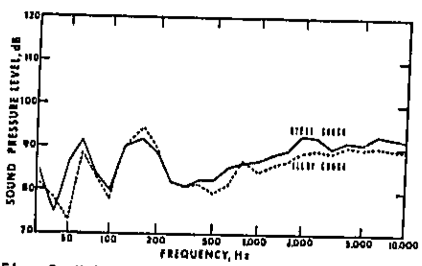

Fig. 3- Noise reduced by alloy rotation chuck

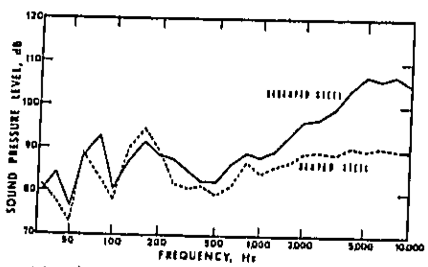

Fig. 4- Noise reduced by damping drill steel

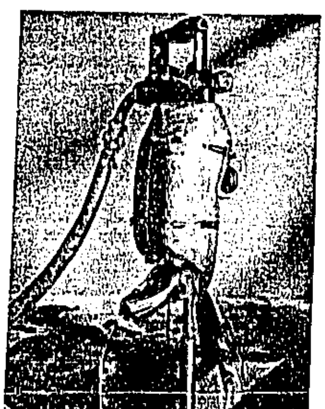

Fig. 5- Prototype drill showing case and muffler outlets and constrained layer damping on steel. End view of steel at right.

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A PRACTICAL APPROACH TO THE EXHAUST SILENCING OF THE PNEUMATIC ROCK DRILL

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INTRODUCTION

Pneumatic rock drills provide the most widely used method of percussion impact used in mining operations. They have also been long recognized as one of the major sources of high intensity noise associated with the mining industry. With the greater concern in recent years over noise as one of the several causes of deafness in the employees of the mining industry, the demand for quieter drilling operations has increased. With this increased demand, a study into the practical exhaust silencing of a pneumatic rock drill becomes essential.

In this research program sponsored by the U.S. Bureau of Mines, Rolla Metallurgy Research Center the exhaust noise of a medium sized pneumatic rock drill was measured and analyzed. A practical muffler was then developed which would reduce the exhaust noise to an acceptable level with a minimum increase in exhaust back pressure.

LITERATURE REVIEW

As a part of the United States Bureau of Mines Mineral Industry Health Program, a survey of forty metal, non-metal, and coal mining facilities across the nation was conducted by Derzay and Goodwin (1)* to determine the noise levels that miners were subjected to in their working environments. A total of 62 different sources of noise creating 90 dBA or more, at a position corresponding to that of the operator's ear, were measured. Of these, 37 created noise levels greater than 100 dBA, 14 greater than 110 dBA, and 4 had values in excess of 120 dBA. All observations of noise levels above 115 dBA were from pneumatic rock drills.

Miller (2) determined that there were three predominant sources of noise produced by the rock drill; air exhaust noise, drill steel vibrations, and noise produced by mechanisms within the drill.

Walker (3) and Holdo(4) each state that the two main sources of noise are the air exhaust noise and the impact or drill steel noise. Holdo calculated, on an energy basis from measurements with an Atlas Copco BBD 45 drill, that 87.5% of the rock drill noise comes from the exhaust air and 12.5% from the impact of the piston in the drill against the drill steel, which causes vibrations in the drill steel and in various parts of the drill. He further concluded that, in relation to the useful energy produced by the rock drill, the noise energy was only 0.08%.

A detailed investigation into the sources of noise in pneumatic rock drills was conducted by Beiers (5) to determine the nature and approximate quantitative value of each important noise source in a particular test drill. He concluded that, at the operator's ear position, the order of importance of the individual noise sources contributing to the overall intensity were; exhaust process, mounting rattle between the drill and the pusher leg, movement of the rifle bar, rock penetration, valve movement, and the impact between the piston and the drill steel shank. Using a drill producing approximately 123 dB at the operator's ear, he was able to arrive at the following values for the three most offending noise sources: exhaust noise - 122.5 dB, mounting rattle - 113.5 dB, rifle bar noise - 109.5 dB.

*Numbers in parenthesis denote references listed at end of paper.

In summary, investigators have distinguished the various noise sources in the pneumatic rock drill and have concluded that the unmuffled exhaust noise is the most offending source. Although mufflers have been investigated that are relatively effective under non-flow conditions, little has been published which illustrates how a practical muffler could be constructed to reduce the actual exhaust noise to acceptable levels.

EQUIPMENT AND PROCEDURE

In order that sound pressure levels quoted in this work could be compared to measurements taken by other researchers, sound measurement positions were selected according to the CAGI-PNEUROP (9) test code. Since the objective of this research program was to develop a muffler to suppress exhaust noise, the measurement procedure was modified slightly in that rather than drilling vertically into granite while the noise measurements were being taken, the drill was suspended from ropes and springs and allowed to reciprocate freely.

A Chicago Pneumatic CP-59 Sinker Drill was made available for the test program. This is a medium sized machine and is considered to be fairly representative of those commercially available. Tests confirmed that the exhaust spectrum produced by the CP-59 drill was quite similar to that produced by two other medium sized drills obtained from other manufacturers. Therefore, a muffler designed for the test drill should be applicable to most other rock drill models with only slight modifications and should produce approximately the same reduction in the level of the exhaust noise.

The exhaust noise produced during each run was recorded with the aid of a magnetic tape recorder. A portion of that recorded tape, approximately 26 inches long, was then made into a loop and played continuously through a 1/3 octave analyzer and recorded on chart paper with the aid of a graphic level recorder. The result was a permanent graphic recording of the sound pressure level as a function of frequency.

PARAMETERS SELECTED TO EVALUATE MUFFLER PERFORMANCE

In order that the mufflers developed for the suppression of exhaust noise during this research project could be utilized by the mining industry, four parameters were selected to evaluate the prototype muffler performance. These were the insertion loss, back pressure developed, adaptability to pneumatic drills, and icing characteristics.

Insertion loss is defined as the difference between the SPL at one point in space before and after the muffler is attached to the noise source. One-third octave analyses of the exhaust spectrum were utilized to identify the most troublesome bands of exhaust noise so that noise reduction techniques could be directed toward them. Additional 1/3 octave spectrograms then indicated if these techniques were successful. The overall reading on the "A" weighted network of a sound survey meter was obtained to give an indication of subjective response to the noise.

The Walsh-Healey Act was used as a guideline for the establishment of a design objective for the attenuation produced by the exhaust muffler. It is realized that the operation of a pneumatic rock drill is intermittent with noise being produced during approximately six hours of an eight hour shift. This would mean that the overall sound pressure level produced during operation should be no more than about 95 dBA.

Since the exhaust process is only one of the contributors to the overall noise level, many of which exceed 90 dBA, 90 dBA was chosen as the maximum allowable SPL to which the prototype muffler would reduce the exhaust noise. Although a reduction of the exhaust noise below 109 dBA to 107 dBA would not be detectable in present drilling operations due to the high intensity noise produced by drill steel vibrations, research presently being conducted by the Rolla Metallurgy Research Center is making progress in reducing this source of noise. Drill steel noise, during a drilling operation, has already been reduced below 100 dBA. Continuing research will (hopefully) further reduce this noise source to a level such that the reduction of the exhaust noise to 90 dBA will be necessary to obtain a quiet drilling operation.

Using the penetration rate, while drilling vertically into granite, as a measure of drill performance it was determined that the back pressure produced by a prototype muffler should be no greater than 2.0 psi. An air gage placed at the inlet to the muffler allowed the back pressure to be measured without interfering with either the operation of the drill or the muffler attenuation characteristics.

In order that the mufflers developed could be adaptable to new and existing pneumatic drills, it was decided that they must be small enough to be incorporated into the cases of new drills and practical enough to install on existing drills as an add-on feature. Since most drills of the medium classification studied in this research are about the same size, a length of from 8 to 10 inches toward the fronthead from the exhaust port was considered acceptable. It was felt that the cross sectional shape of the muffler need not be circular but could be as wide as the body of the drill, which is approximately 5 inches, and extend outward from the body of the drill by as much as 4 inches. The muffler could be extended from the exhaust inlet toward the backhead in order that air escaping from vents in the valve case of certain drills could be channeled through the muffler.

Muffler development progressed under the assumption that icing due to water in the supply air would not be a problem and that in wet drills* the water could be injected into the drill steel in such a manner that it would not enter the cylinder when the drilling operation ceased. This would prevent icing within the muffler due to the discharge of the accumulated water through the exhaust port when the drilling operation is resumed. After prototype mufflers had been developed which were considered acceptable by the first three parameters, their icing characteristics were studied by the introduction of water into the exhaust air.

ISOLATION OF EXHAUST NOISE AND TESTING OF PROTOTYPE MUFFLERS

A chamber was constructed in which the drill could be suspended and allowed to reciprocate freely. The exhaust air was vented to the outside of the enclosure with a short piece of one inch inside diameter rubber hose in order that spectrum analyses of exhaust noise could be obtained without the affects of body-radiated noises and air leaks from the drill. The outlet for the exhaust air was located 92 cm above a concrete floor and provided a very directional source of noise. The microphone position selected was one meter from the exhaust outlet and 50 cm above the center line of the exhaust air flow.

With the drill operating within the enclosure and the exhaust vented toward the microphone, the overall SPL was 116 dB (114.5 dBA). With the same microphone position, but using the exhaust hose to carry the exhaust noise away from the measuring location, the overall SPL was reduced to 77 dB. The high transmission loss through the enclosure allowed analyses of the exhaust noise and prototype mufflers to be performed without the effects of air leaks and body-radiated noise from the drill. The background noise varied from 62 dB to a maximum of 70 dB and did not affect the exhaust analyses.

For ease of construction, prototype muffler bodies were fabricated from polyvinyl chloride (PVC) tubing with an inside diameter of 2 in., and PVC sheet 1/8 in. thick. A number of simple expansion chambers and resonators of various dimensions were then evaluated. However, none of these designs reduced the unmuffled exhaust noise by more than 5 dBA. Further testing was concentrated on various expansion chambers filled with steel wool, except for a perforated central air passage.

RESULTS AND CONCLUSIONS

The results of performance testing of six mufflers, considered representative of those investigated, are shown in Table I. Those mufflers listed in this table are:

1) Simple expansion chamber muffler, 8 1/2 inches long.

2) Single chamber resonator muffler, 8 1/2 inches long.

3) Expansion chamber muffler, 8 1/2 inches long, steel wool liner.

4) Expansion chamber muffler with an extension toward the backhead, steel wool liner.

*Wet drills refer to those pneumatic rock drills in which water, rather than air, is used to remove dust and rock chips from the hole.

5) Elliptical expansion chamber muffler, 8 1/2 inches long, steel wool liner.

6) Elliptical expansion chamber muffler with an extension toward the backhead, steel wool liner

Each of those mufflers with a steel wool liner also utilized an air passage fabricated from PVC tubing with 3/8 inch diameter connecting holes. The hole spacing was such that each tube had approximately a 40% open area. All of the expansion chamber mufflers had a 3 inch tail pipe except muffler #6, which had a tail pipe 1 inch long.

Table I

Performance Tests of Six Prototype Mufflers

Muffler Configuration	Overall Sound Pressure Level			
	Frontal Position		Operator's Ear Position	
	(dB)	(dBA)	(dB)	(dBA)
1	112	109.5		
2	117.5	116.5		
3	102	95.5	99.5	94.5
4	97	90	96.5	88
5	102	98.5	100	95.5
6	94	88	93	87

From the results obtained, the prototype muffler shown in Figure I was developed, and the following conclusions were drawn.

1. Slight amounts of exhaust back pressure developed by a muffler are not detrimental to the drill performance and may, in some cases, cause an increased penetration rate.
2. In order that a muffler be small enough to be incorporated into a pneumatic rock drill and still reduce the exhaust noise to an acceptable level, it must combine reactive and dissipative elements.
3. Care should be taken in muffler development so as not to introduce materials or elements into the exhaust stream which will cause excessive back pressure and possible icing problems.

The prototype muffler developed during this research had an insertion loss of 22 dB and reduced the exhaust noise, at the operator's ear position, from 115 dBA to 87 dBA. Muffled and unmuffled exhaust spectra are shown in Figure 2. The back pressure developed by this muffler is 0.5 psi, which causes approximately a 4% increase in the drilling efficiency for the test drill.

The prototype muffler combined a reactive element (an expansion chamber) with a dissipative element composed of a steel wool liner. The muffler can be utilized without modification in mining operations where the supply air is free of moisture. Preliminary tests indicate that icing will not occur even when normal amounts of moisture are present.

Research presently being conducted by the Rolla Metallurgy Research Center is making progress in reducing the noise produced by drill steel vibrations and the internal mechanisms of the drill to produce a quiet pneumatic rock drill. The large reduction in the exhaust sound pressure level obtained during this study is necessary in order that the overall noise level produced by the quiet pneumatic rock drill will meet those standards established by the Walsh-Healey Act.

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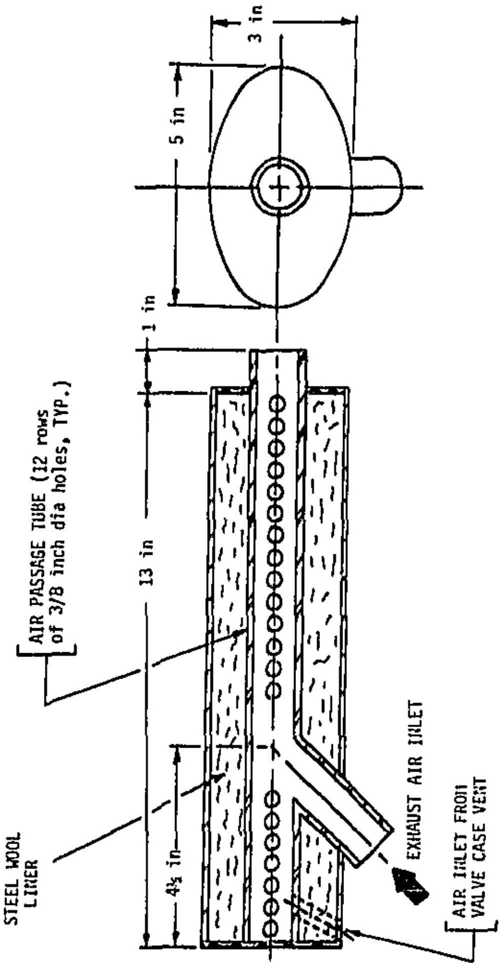
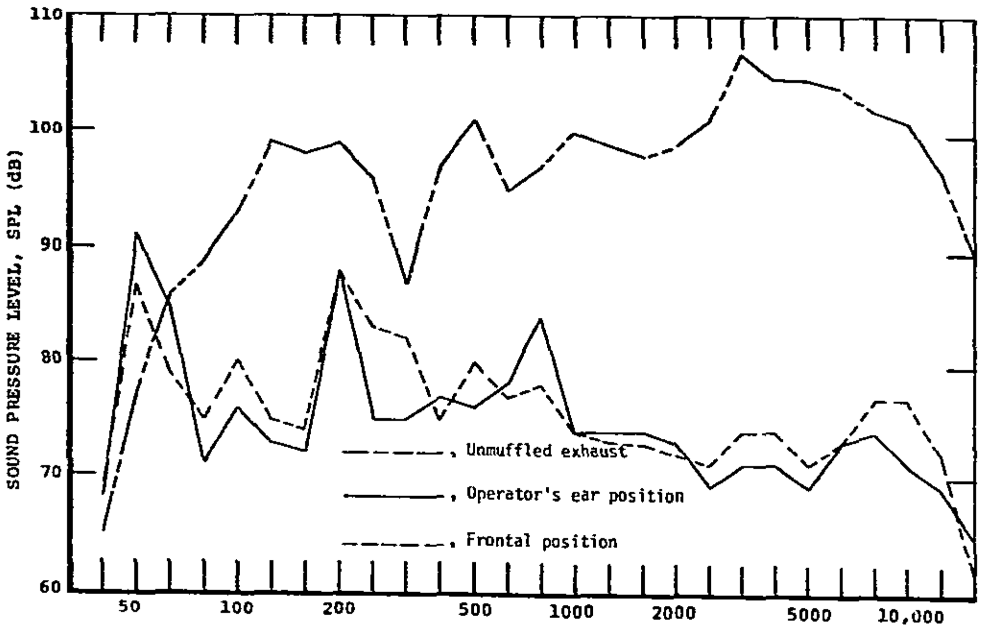


Figure I. Prototype Muffler Configuration No. 6



PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

Figure 2

One-third Octave Spectrogram for Prototype Muffler Configuration No. 6

PREDICTION SCHEME FOR THE SELF-GENERATED NOISE OF SILENCERS

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ABSTRACT

Analysis of a large volume of self-noise data of various types of duct silencers of different manufacturers showed that the octave band sound power level of the self noise collapses if normalized in respect to the 5.5th power of the flow speed in the air passage and in respect to the face area of the silencer. There is evidence that the derived empirical formula can also be used to predict the self noise of unobstructed tailpipes for subsonic velocities.

INTRODUCTION

While it is common knowledge that silencers attenuate noise, it is less known that under certain circumstances they may also generate noise. In the majority of cases, a silencer not only needs to attenuate noise but also is required to pass a certain gas flow with a minimum of pressure drop. The aerodynamic noise, which is a direct consequence of the gas flow, is created by vortex shedding and by wakes at the terminal end of the silencer. It is often referred to as the self-generated noise or self noise of the silencer.

When a silencer of high sound attenuation is used in ducts with a high velocity gas flow, it may frequently be the case that the self-generated noise of the silencer is higher than the noise attributable to the source. Accordingly, the designer of the noise control treatment needs to predict the level of the self-generated noise in terms of the known system parameters. The purpose of this paper is to provide the designer such a prediction scheme.

BASIS OF THE PREDICTION SCHEME

The system parameters always available to the designer of the silencer treatment are (1) face velocity, (2) face area, and (3) percentage of open area of the silencer. Sometimes, additional parameters such as pressure drop and the sound power level of the self noise can be found for packaged duct silencers of certain manufacturers. However, these parameters are not available for any new, yet untested designs. Accordingly, an empirical prediction scheme, which is based on the first listed three system parameters, would be the most practical to use in predicting the self noise of new silencer designs.

The leading manufacturers of duct silencers have published octave band sound power level data as a function of the face velocity for their line of silencers. The prediction scheme reported in this paper is based on the analysis of these published data [1]. The silencers considered varied significantly in geometry and in percentage of open area.*

Velocity Dependence

As expected from a noise of aerodynamic nature, the power level of the self-generated noise of silencers increases rapidly with the velocity of the gas stream through the silencer.

* The silencers investigated included various types of rectangular and circular cross section. The percentage of open area varied from 30% to 78%.

Figure 1 shows the measured octave band sound power level vs frequency curve of a rectangular duct silencer of 4 sq ft face area (30% open) with the face velocity as parameter. Observing the three curves, one notes that a higher face velocity shifts the spectrum upwards but practically does not affect its shape. This observation was found to be generally true also for all other silencer types analyzed. Accordingly, there was no need to attempt a Strouhal scaling.

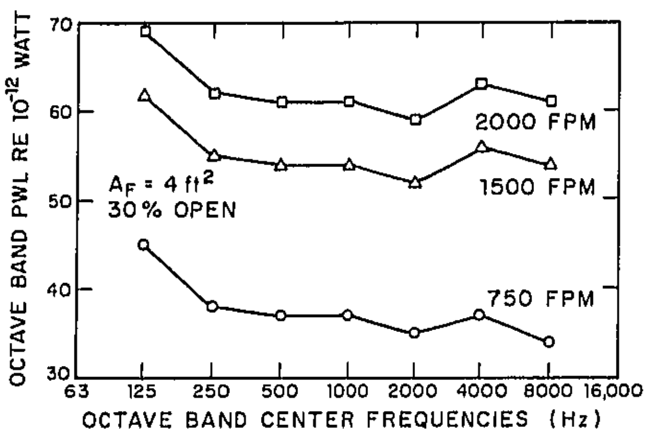

Fig. 1. Octave Band Sound Power Level of Self Noise of a Typical Silencer With Face Velocity as Parameter

Figure 2 shows the octave band spectra of Fig. 1 normalized by the 5.5th power of the face velocity. With this scaling, the data points obtained for the different face velocities collapse; a behavior characteristic also for all other silencers.

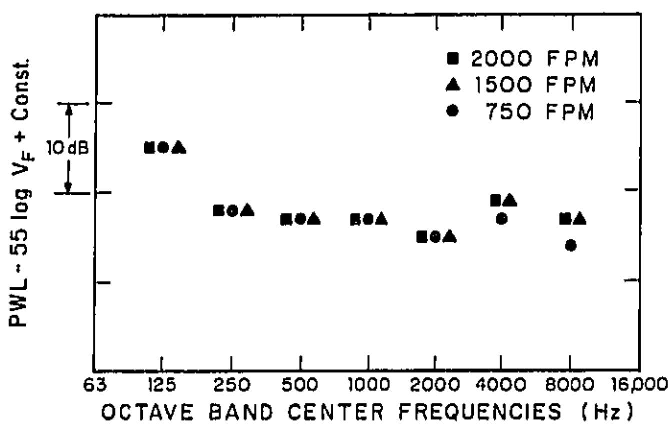

Fig. 2. Self-Noise Spectrum of Fig. 1 Normalized by 55 log V_F

Area Dependence

Since the sound power level of the self-generated noise is determined by the actual velocity in the air passage of the silencer and by the open area of the silencer rather than by the face velocity and by the face area, one has to apply correction terms in addition to the simple face velocity scaling to obtain data collapse for silencers with different percentages of open area. It was found that two additional terms were needed to accomplish this. The first term, 10 log A_F, with A_F being the face area, takes into account the total exit area, while the second term -45 log P/100, with P being the percentage of open area, adjusts for the increased velocity in the open passages of the silencer.

PREDICTION SCHEME

Normalizing the large body of available self-noise data of various silencers yielded a satisfactory data collapse, as shown in Fig. 3 where the normalized octave band sound power level data of silencers of different geometry and percentage of open area are plotted.

$$PWL_N = PWL_{oct} - 55 \log V_F + 10 \log A_F - 45 \log P/100 \quad , \tag{1}$$

The symbol PWL_{oct} represents the measured octave band sound power level of the self-generated noise in dB re 10^{-12} watt, V_F the face velocity in ft/min, A_F the face area in sq ft, and P the percentage of open area.

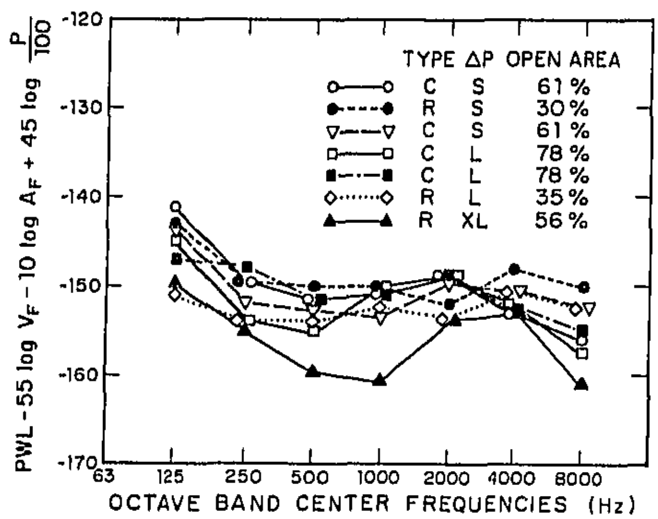

Fig. 3. Normalized Octave Band Self-Noise Sound Power Levels of Various Silencers

Figure 4 compares the range of the normalized self-noise sound power data of the products of two duct silencer manufacturers. The product lines of both manufacturers yield essentially the same range. Considering the similarity of the products, which is characteristic in this highly competitive field, this agreement is of little surprise. More surprising is the relatively narrow range of the normalized sound power level data of silencers of widely varying geometry and percentage of open area in such a wide velocity range.

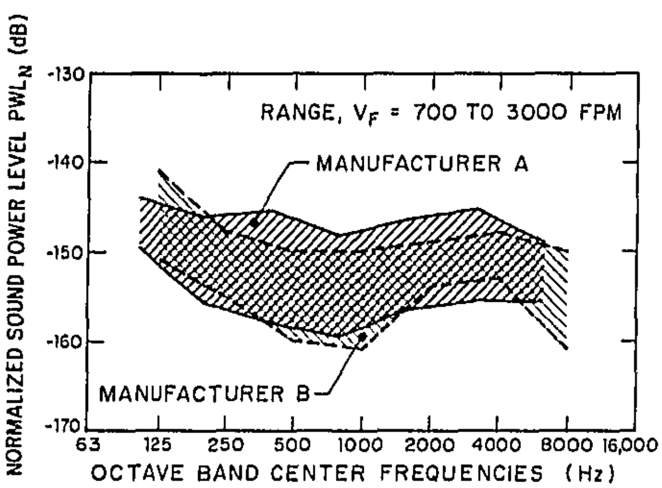

Fig. 4. Range of Normalized Octave Band Self Noise Levels for Two Silencers

Based on the results of our analyses, as summarized in Fig. 4, the following prediction scheme for the octave band sound power level of the self noise is offered:

$$PWL_{oct} = -145 + 55 \log V_F + 10 \log A_F - 45 \log \frac{P}{100} - 20 \log \frac{460+T}{530} \quad , \qquad (2)$$

where PWL_{oct} is the octave band sound power level re 10^{-12} watt, V_F is the face velocity in ft/min, A_F is the face area in sq ft, P is the percentage of open area of the cross section, and T is the temperature of the gas (air) in °F.

Introducing the face Mach number, $M_F = V_F/c$, modifies Eq. 2 to

$$PWL_{oct} = 121 + 55 \log M_F + 10 \log A_F - 45 \log \frac{P}{100} + 7.5 \log \frac{460+T}{530} \quad . \qquad (3)$$

Note that the predicted octave band spectrum is flat over the entire audio frequency range. Also note that the sound power level estimated by Eqs. 2 and 3 may be considered as an upper limit obtained with no particular care in aerodynamic design of the silencer. Streamlining the downstream end of the silencer baffles may reduce the power level of the self-generated noise by as much as 5 to 10 dB.

Note also that the octave band sound pressure level in dB re 0.0002 μ bar in the exit plane of the silencer attributable to self noise may be approximated by adding 10 dB to the power level predicted by Eqs. 2 and 3.

USE OF THE PREDICTION SCHEME

Equations 2 and 3 are based on data obtained in test facilities where the silencer terminates into a reverberant room. This is an accepted method of determining the self-noise characteristics of duct silencers [2]. Since terminal conditions are not much different in the cases of gas turbine exhaust silencers or discharge silencers, it is believed that, until more research is done on the subject, Eqs. 2 and 3 may be useful tools for the designer to estimate the sound power level of the self noise of discharge silencers. We note that Eqs. 2 and 3 are in line with schemes offered by other investigators [3] to predict the A-weighted sound power level of the self noise of silencers. In addition, they yield results which are in reasonable agreement with methods offered [4] for predicting the aerodynamically generated noise in plane empty stacks (P=100) discharging a highly turbulent subsonic gas.

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MECHANISMS OF NOISE GENERATION BY FLUID FLOW THROUGH CONTROL VALVES

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ABSTRACT

This paper presents a comprehensive analysis of the mechanisms of noise generation by fluid flow through control valves. The analysis, well supported by data obtained from actual tests of a large variety of valve styles in sizes 1" through 12", includes both aerodynamic and hydrodynamic noise sources. The information as presented has practical application in the design of control valves that generate noise at substantially lower levels than conventional valves. It is also shown that prediction of control valve noise for a given installation requires both a quantitative knowledge of the noise characteristic of the valve and an understanding of the noise transmission loss thru pipe.

INTRODUCTION

Control valves are predominant noise sources in many fluid process and/or transmission systems common to gas regulator stations, chemical plants, pulp and paper mills, oil refineries, and power stations. Prediction of the noise for a given valve installation necessitates both a quantitative knowledge of the noise characteristic of the valve and an understanding of sound transmission loss through the solid boundaries that contain the flow stream.

SOURCES OF VALVE NOISE

The major sources of control valve noise are:

(1) Mechanical vibration of valve components
(2) Hydrodynamic noise
(3) Aerodynamic noise

Mechanical Noise

Vibration of valve components is a result of random pressure fluctuations within the valve body and/or fluid impingement upon the movable or flexible parts. Noise that is a by-product of the vibration of valve components is usually of secondary concern and may even be beneficial since it warns that conditions exist which could produce valve failure. Mechanical vibration has for the most part been eliminated by improved valve design and is generally considered a structural problem rather than a noise problem.

Hydrodynamic Noise

The major source of hydrodynamic noise (noise resulting from liquid flow) is cavitation. This noise is caused by the implosion of vapor bubbles that are formed in the cavitation process. Cavitation occurs in valves controlling liquids when the service conditions are such that the static pressure downstream of the valve is greater than the vapor pressure and at some point within the valve the local static pressure, either because of high velocity and/or intense turbulence, is less than or equal to the liquid vapor pressure.

Figure 1 depicts the pressure profile of a cavitating flow stream as a function of distance along the stream. Vapor bubbles are formed in the region of minimum static pressure and subsequently are collapsed or imploded as they pass downstream into the pressure recovery region. Noise produced by cavitation has a broad frequency range and is frequently described as a rattling sound similar to that which would be anticipated if gravel were in the fluid stream.

Test results supported with field experience, indicate that noise levels from non-cavitating liquid applications are quite low and generally would not be considered a noise problem. Figure 2 depicts the typical characteristic of hydrodynamic noise as a function of the ratio of differential pressure across the valve (ΔP) to the static pressure at the inlet (P_1 - psia) minus the vapor pressure (P_V - psia).

Aerodynamic Noise

The major source of valve noise is aerodynamic noise. "Aerodynamic noise" is noise generated as a by-product of a turbulent gas stream, or noise produced without the interaction of the fluid with vibrating boundaries or other external energy sources. Lighthill's paper "On Sound Generated Aerodynamically" provides the general theory concerning production of noise created without the aid of vibrating bodies.[1]

The three fundamental sources of sound are defined here to facilitate comprehension of the subject matter.

(1) The monopole or simple source forces the mass in a fixed region of space to fluctuate; i.e., a pulsating sphere or a loud-speaker diaphragm.

(2) The dipole source forces the momentum in a fixed region of space to fluctuate; i.e., vibration of a solid object after being struck or two simple and equal sound sources a small distance apart pulsating 180° out of phase with each other.

(3) The quadrupole source forces the momentum flux to vary. Physically the quadrupole can be equated to two parallel and equal forces oppositely directed.

Aerodynamic noise is a result of the Reynolds stresses or shear forces created in the flow stream as a result of deceleration, expansion, or impingement. The principal area of noise generation in a control valve is the recovery region immediately downstream of the vena contracta where the flow field characterized by intense turbulence and mixing is of chaotic quality with phase being completely random and discontinuous. The concept of spectral density or amplitude as a function of frequency for the fluctuating physical quantities (pressure, density, and velocity) is without meaning.

The classification of the noise generating mechanism relative to the three fundamental sources of sound (monopole, dipole, and quadrupole) is not imperative to the development of a valve noise prediction technique. In the interest of space we must forego any brief not beneficial to the practical application of information presented, hence arguments concerning description of the noise mechanism will be deferred. However, from the definitions listed above it would seem obvious the subject sound field (turbulent mixing noise) would conceivably be either dipole or quadrupole in nature.

VALVE NOISE PREDICTION

The greatest limitation in development of a mathematical model that maps the noise characteristic of a control valve to the known flow parameters is lack of either quantitative or qualitative knowledge concerning the aerodynamics of a bounded flow stream. The noise generators are random functions of space and time, thus only statistical properties (such as autocorrelation functions) of these quantities can be measured or predicted.

Solution of many problems that defy classical analytical techniques, for practicality depend on elegantly simple models fashioned from dimensional reasoning. This is the category applicable to the problem of valve noise prediction.

AERODYNAMIC DIMENSIONAL ANALYSIS

In C. B. Schuder's "Coping with Control-Valve Noise", a comprehensive dimensional analysis of aerodynamic valve noise, it is hypothesized that the acoustic power generated by compressible flow thru a control valve obeys the following relationship:[2]

$$W \propto C_g^2 (\Delta P)^2 \; f(\Delta P/P_1) \qquad (1)$$

where: W = acoustic power
C_g = gas sizing coefficient (directly proportional to area of restriction)[3]
ΔP = pressure differential across valve
P_1 = absolute inlet pressure

The rationale used in development of the above relationship assumed a single port or noise generator. In the following equation an extension is made to the basic relationship to include the effects of multiple ports.

[1]Superior numbers refer to similarly-numbered references at the end of this paper.

$$W \propto 1/N\ C_g^2 (\Delta P)^2\ f(\Delta P/P_1) \qquad (2)$$

where: N = number of ports open to the flow stream

Proportionalities obtained in dimensional formulae are frequently inexact and must be verified by experimental findings before they can be regarded as acceptable.

TESTING PROGRAM

Test work to verify Relationship (2) was conducted in Fisher Controls modern Flow Laboratory using a controlled sound environment (mobile soft room) to provide isolation of the test valve and/or piping from other noise sources.

Figure 3 shows a photograph of the facilities used in the testing program. The mobile soft room (large rectangular-shaped structure in the foreground) can be parted longitudinally to facilitate piping changes and movement from one test line to another. The room can be sealed around 4" thru 12" piping and provides 50 decibels (dB) attenuation in ambient sound pressure level (SPL) from outside to inside the room in the frequency range of interest.

The flow facilities provide metered flow up to 10,700 gpm of water and 1.25×10^4 scfh of air. Maximum flow capabilities for tests of short duration are 16,000 gpm of water or 1.8×10^4 scfh of air. Absorption-type inline silencers were used upstream and downstream of the test section for isolation from fluid-borne noise generated at other points in the system.

Sound measurement was made with a Hewlett-Packard "Real Time Audio Spectrum Analyzer" that provided a CRT display and digital printout of 1/3-octave band analysis of the audio spectrum for test runs with time duration as short as 28 milliseconds.

Noise levels were measured for a large variety of valve styles in sizes 1" through 12" as a function of valve travel, $\Delta P/P_1$ ratio, and adjacent piping configuration.

WORKING EQUATION

Space does not permit a detailed presentation of all the valve noise measurements obtained, thus only a small sampling of over 1,000,000 accumulated data points is presented. Figures 4-8 show the excellent correlation of test data with Relationship (2) and serve to demonstrate the basis of a graphical technique for prediction of aerodynamic valve noise. The base SPL is taken from the ΔP curve with additive corrections then made for C_g and for valve geometry as a function of $\Delta P/P_1$ ratio. The result based on standard weight pipe gives the contribution to the ambient SPL at a distance of 48 in. downstream from the valve and 29 in. from the pipe surface. Since most noise measurements of valves are made on complete installations, quantitative information on the acoustic behavior of the valve itself would be of questionable value.

HYDRODYNAMIC DIMENSIONAL ANALYSIS

Using a similar approach to that delineated above for aerodynamic noise it can be shown that the following relationship exists for hydrodynamic valve noise.[4]

$$W \propto C_v^2\ f\left(\Delta P,\ \frac{\Delta P}{P_1 - P_v}\right)\ f\left(\frac{\Delta P}{P_1 - P_v}\right)$$

where: C_v = liquid sizing coefficient
P_v = vapor pressure

An expeditious graphical solution of the same format described for aerodynamic noise is utilized to predict hydrodynamic noise.

NOISE TRANSMISSION

The primary concern with valve noise is its contribution to the overall ambient noise level. In closed systems (not vented to atmosphere) any noise produced in the process becomes airborne only by vibration of the solid boundaries that contain the flow stream. Thus, an understanding of the relative transmission loss as a function of the physical properties of the solid boundaries is imperative to accurate noise prediction.

The transmission loss characteristic for cylindrical piping is shown in Figure 9. Transmission loss is stiffness controlled at frequencies below the ring frequency and mass controlled at higher frequencies. Ring frequency (f_r) can be calculated using the following equation:

$$f_r = \frac{C_L}{\pi D_p}$$

where: D_p = nominal diameter of pipe
C_L = longitudinal speed of sound in metal

The problem of predicting pipe transmission loss has not yet been solved. The multiplicity of response modes and broad band frequency excitation force (noise field) enhance the complexity of calculating the response of the pipe to the noise within. The theory of vibration of cylindrical shells can be used to calculate the various resonant frequencies. At high frequencies the resonances are grouped sufficiently close together to approach a condition of continuous response; see Figure 9. Even with the enormous computational capacity of computers, the large number of resonant frequencies and modes that exist impose a practical limitation on the utility of a theoretical approach to calculation of transmission loss.

Table I presents a tabulation of empirically determined values for the change in pipe transmission loss, relative to standard weight, as a function of pipe schedule.

The dissipation of acoustic energy into heat, by viscosity and heat conduction is a slow process; i.e., at a frequency of 4K Hz only half the energy is dissipated in the first 5000 ft of propagation thru the atmosphere. Thus the noise generated within a closed transmission system is frequently propagated for long distances in the fluid stream with little attenuation. Consequently, any change in piping schedule at relatively large distances from the valve may effect a change in the ambient noise level.

DESIGN OF QUIET VALVES*

Material presented herein provides basic information relevant to design of quiet valves. From the preceding sections the parameters that determine the level of noise generated by compressible flow thru a control valve for a given application are: the number of ports or restrictions exposed to the flow stream, the total C_g, differential pressure across the valve, and the ratio of differential pressure to absolute inlet pressure.

From Relationship (2), the acoustic power of a single flow restriction increases as a function of C_g^2 or changing the area by a factor of 2 results in a corresponding 6 dB change of power level, whereas, the power level is changed only 3 dB when the number of equal noise sources is changed by a factor of two. Thus noise reduction to be derived from utilization of many small restrictions vs a single or few large restrictions is self-evident.

Figures 4-8 show that the noise characteristic or noise potential increases as a function of P^2 and $\Delta P/P_1$. Thus for high pressure ratio applications ($\Delta P/P_1 > 0.7$) an appreciable reduction in noise can be effected by staging the reduction thru a series of restrictions to produce the total pressure head loss required. This approach to quiet valves can easily be incorporated into cage style trim fabricated from stacks of discs machined to stage the total pressure drop thru a series of circumferential restrictions.

Pertaining to the design of quiet valves for liquid application, the problem resolves itself to one of designing to eliminate cavitation. Service conditions that will produce cavitation can readily be calculated.[5] The use of staged or series reductions provide a very viable solution to cavitation and hence hydrodynamic noise.

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*Comments made in this section are applicable to the design of valves that introduce turbulence into the flow stream in producing the permanent head loss required to fulfill the basic function of the valve.
It is conceivable to design a valve that utilizes viscous losses to produce the permanent head loss required. Such an approach would require valve trim with a very high $\ell f/d$ ratio which becomes impractical from the standpoint of both economics and physical size.

CONCLUSIONS

Control valve noise technology has made giant strides during the past four years. Among the most notable advancements are the ability to accurately predict the level and characteristic of the noise radiated to the atmosphere via the adjacent piping and a parametric understanding of the noise generators such that noise can be designed out of a valve for a given application.

The overall noise technology concerned with fluid transmission systems is in its infancy and it is anticipated that the near future will bring refinement of present technology and significant advancements both academic and practical in nature.

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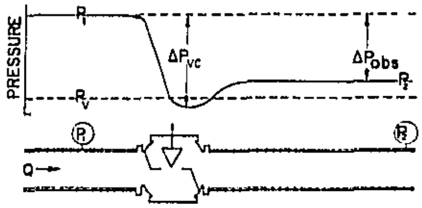

FIGURE 1. STATIC PRESSURE ALONG A STREAM LINE CAVITATING FLOW

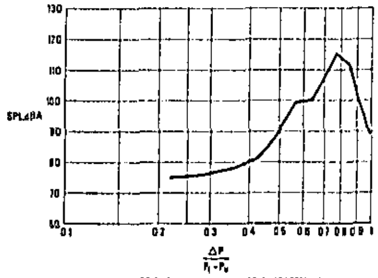

FIGURE 2. TYPICAL LIQUID NOISE CHARACTERISTIC

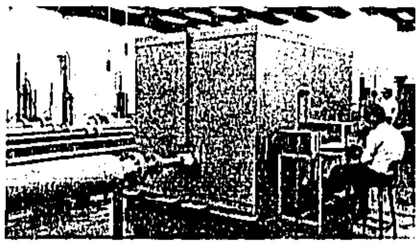

FIGURE 3. NOISE TEST FACILITIES FOR CONTROL VALVES

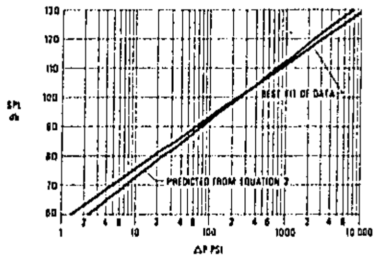

FIGURE 4. EFFECT OF PRESSURE DROP ON EXTERNAL SOUND PRESSURE LEVEL

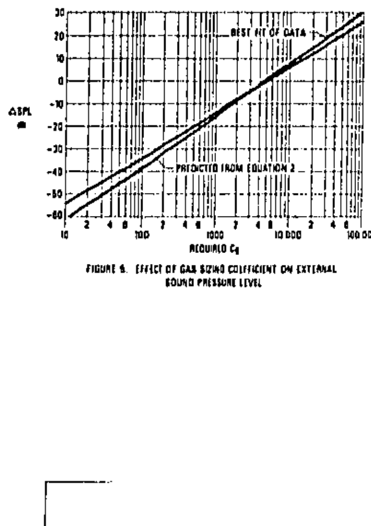

FIGURE 5. EFFECT OF GAS SIZING COEFFICIENT ON EXTERNAL SOUND PRESSURE LEVEL

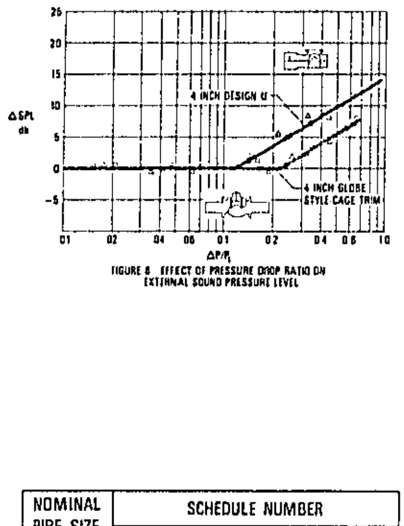

FIGURE 6. EFFECT OF PRESSURE DROP RATIO ON EXTERNAL SOUND PRESSURE LEVEL

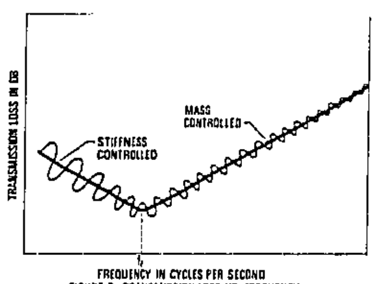

FIGURE 7. TRANSMISSION LOSS VS. FREQUENCY

NOMINAL PIPE SIZE, INCHES	SCHEDULE NUMBER							
	30	40	80	120	160	STD	XS	XXS
2	---	0	+6	---	+12	0	+6	+18
4	---	0	+7	+10	+13	0	+7	+16
6	---	0	+8	+12	+15	0	+8	+18
8	−3	0	+9	+14	+18	0	+9	+16
10	−3	0	+9	+14	+19	0	+6	---
12	−4	0	+10	+16	+20	0	+5	---

TABLE 1. CHANGE IN TRANSMISSION LOSS AS A FUNCTION OF PIPE WALL THICKNESS

NOISE REDUCTION OF MINIATURE FANS USING BLADE TREATMENT

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ABSTRACT

The method of noise reduction by providing porous surface [1,2] on the fan blade was experimentally investigated on a miniature axial-flow fan that is popularly used in cooling for a business machine. Different porous materials such as foam, fiberglass and felt metal were evaluated on the fan. Narrow-band analyses are presented to show the effectiveness on quieting the blade pressure noise and vortex shedding noise using their frequency spectrum signature. A simulation test with stationary blade is given to verify the identification of vortex shedding noise.

INTRODUCTION

Axial fans are used in machines and instruments for cooling. The noise from such fans could be annoying when the equipment is located in a quiet area. One of the methods to reduce fan noise is to mount a porous material on blade surfaces. This concept was discussed by Lowson.[1] The method was recently demonstrated by Chanaud[2] with significant noise reduction. In his paper, a "felt metal" material is used for the porous layer. The results indicate that a low density material gives better reduction and the most effective mounting area is the outer surface of the fan blade. However, his results were only expressed in dBA and gave no information of its effect on frequency spectrum. Also, results of using only one material is shown. In this investigation, three different commercially available materials were studied: metal fibers, fiberglass and polyurethane foam. The noise reduction was examined by constant bandwidth spectrum analysis in order to gain some insight on mechanisms of noise reduction.

EXPERIMENTAL EQUIPMENTS

In the experiments, a uniform thickness of the porous material is cemented to the retarding surface of the fan blade. A 3-blade miniature tube-axial fan of 2.470" Depth x 7" diameter was selected for the tests (Fig. 1). It has a volume flow of 270 CFM and rotating speed of 3400 rpm at free delivery. To facilitate the tests, the fan was mounted on the front of a chamber as indicated in Fig 2. The reflection due to the chamber end and the vibration of the fan transmitted to chamber was carefully minimized. During the tests, the fan was found to be operating at 0.15"static pressure, volume flow of 220 cu. ft./min., and 3,340 rpm. The sound measurements were made constantly at a distance of 2 ft. upstream from the fan and at an angle of 15° off the fan axis. The sound pressure was measured by a Hewlett Packard 8062A sound level meter (microphone H.P. 15119A) and was analyzed by a General Radio 1910A constant bandwidth analyzer. The air velocity at exit was measured by Alnor Velometer.

RESULTS OF VARIOUS SAMPLES

The tested results of different porous samples are listed in Table 1. The noise reduction achieved is far from substantial. However, it gives a comparison of their effectiveness. The metal fiber which is the closest material to the one used in Chanaud's paper, showed comparatively better results among the group. The only material which is comparable to it in the group is fiberglass. For fiberglass, the octive band levels show reduction except in the 250, 500 and 1000 bands.

SPECTRUM ANALYSIS

In order to understand the effectiveness of this method, further studies were made by using exclusively fiberglass for the porous surface.

A 10-Hz constant bandwidth analysis of the fan noise was made before and after the porous surface treatment (Fig. 3 and Fig. 4). The blade passing frequency is identified by the 3rd harmonic of the rotation frequency (55.7 Hz). Overtones of the blade passing frequency showed clearly in the spectrum. All these are generated due to blade pressure. In addition, a wide-band noise is seen centered at 1850 Hz. This was later identified as noise due to vortex shedding from the fan blade.

For convenience of comparison, the peak levels of the harmonics and the wide-band noise level of Fig. 3 are replotted in Fig. 4. The harmonic tones and the wide-band noise centered at 1850 Hz are both seen to be effectively reduced. The 12th harmonic is majorly contributed by the interaction of three blades and four mounting ribs that are located downstream of blades. It showed no reduction since the porous surface was mounted on the opposite surface of the blade with respect to the location of the ribs. However, the overall reduction is deteriated by the increase of wide-band noise below 1000 Hz. The mechanism to cause the noise increase will be discussed in the simulation test.

FAN BLADE SIMULATION TEST

The vortex is regenbrated by a 1-3/4" length x .0625" thickness plate in an air stream. The crossection of the plate is chosen to be the same as that of the blade. During the simulation, the plate was located across the center of the exit of the chamber and the air stream is generated by using a different fan. The sound pressure is measured at 5" above the plate. The spectrums of vortex-shedding noise from simulations are shown in Fig. 5. Patterns similar to the one at 1850 Hz in Fig. 2 are observed. From vortex frequencies indicated in Fig. 5, the Strouhol number of the plate (based on the plate thickness and free stream velocity) is found to be the same as that of a cylinder with diameter equal to the plate thickness.

To establish the validity of the test, a .0625"diameter cylinder rod is tested in the air stream. The Strouhol number is 0.22 at a Reynolds number of 2000. It is close to a well established Strouhol number of 0.21 for cylinder body[3]. This concludes the validity of the simulation test.

The vortex frequency of the blade in Fig. 3 is predicted using data from the simulation test. The free stream air velocity in formulation of Strouhol number is considered to be the component of velocity of air relative to the blade in the direction of the trailing edge of the blade. This is determined by

$$U_o = V_{air} \sin\theta + V_b \cos\theta ,$$

where U_o is the free stream velocity,

V_{air}, the air velocity in axial direction, V_b, the average blade rotating speed and θ, the angle between trailing edge of the blade and plane of the fan. Using the measured Strouhol number of 0.22 and computed free stream velocity, the frequency is determined to be 1800 Hz. In conclusion, the 1850 Hz in Fig. 3 is identified as vortex frequency.

The phenomenon of increasing noise at frequency below the vortex frequency, e.g., Fig. 4, is also studied in the simulation test. The 1/16" fiberglass is cemented to the top of the plate. The effect of the porous surface is shown in Fig. 6, the vortex shedding noise is reduced, however, the noise is increased at half of the vortex frequency. This indicates the low vortex frequency is actually introduced due to the increased overall thickness of the plate and the fiberglass. However, the question remains whether the vortex shedding noise is equally eliminated by only an increase of overall plate thickness. This is answered by comparing two sound spectrums(Fig.7): one due to a solid plate and the other, a plate cemented with fiberglass of the same overall thickness. The result disclosed that the low frequency noise is increased more when only the thickness of the plate is increased instead of adding on a porous surface. The porous material introduces less noise than solid material. In order to achieve overall noise reduction, it is important to keep the low frequency noise,introduced by porous blade,minimum.

CONCLUSIONS

1. For most fans of small number blades, the vortex shedding noise is at higher frequency than blade pressure. The method of mounting porous material on blades can effectively reduce fan noise if the vortex shedding noise is dominant.
2. The fiberglass material is found to be effective in porous surface treatment. The low cost of the fiberglass makes it an attractive material in this treatment. Future investigation should be made to improve the performance of fiberglass by determining an optimum selection of fiber diameter and density.
3. The porous material achieves the noise reduction by two mechanisms.
   a. It acts hydrodynamically as a material of high resistance to attenuate the fluctuation of blade pressure[2]. This is confirmed by reduction of harmonics in Fig. 3.
   b. It relieves the pressure built up around the blade to reduce the vortex strength. Some similar studies were made on the vibration created by the vortex shedding on cylinder body.[4] In this consideration, an effective porous material should not only reduce vortex shedding noise but also introduce the least low-frequency noise.
4. The simulation test using a stationary blade is useful in deriving the vortex frequency of blade and evaluating the porous material used for blade treatment.

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Porous Material	Volume Flow	dBA	Octave band levels (dB.) 62	125	250	500	1K	2K	4K	8K
			Fan Noise							
No material	100%	60	54	59.5	54.5	52.5	50.5	58	50.5	37.5
			Noise Reduction (dB.)							
1/8" thk. Polyurethane foam, 2 lb./ft.^{3}	92%	2.	0.5	2.5	2.5	0.	0.	2.5	0.5	0.
1/8" thk. Polyurethane foam, 12 lb./ft.^{3}	91%	2.	0.	3.5	0.	-5.5	-2.	8.0	2.5	0.
1/16" thk. Fiberglass, 3.5 lb./ft.^{3}, 4-10 micron fiber dia.	94%	4.0	0.	3.5	0.	-3	0	7.5	3	2.
0.012 thk. metal fiber, 0.070 lb./in.^{3}, 8 micron fiber dia.	95%	3.0	3.	4.	-1.5	-2.5	1	7.5	0.5	-1.5

Table 1 Noise reduction of the model fan using different porous materials.

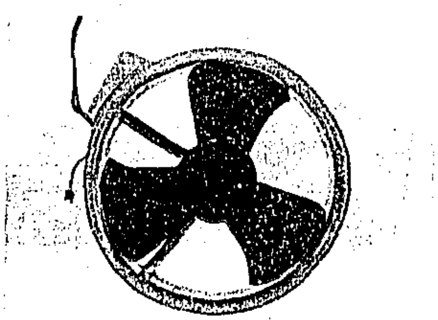

Fig. 1 Photograph of the model fan.

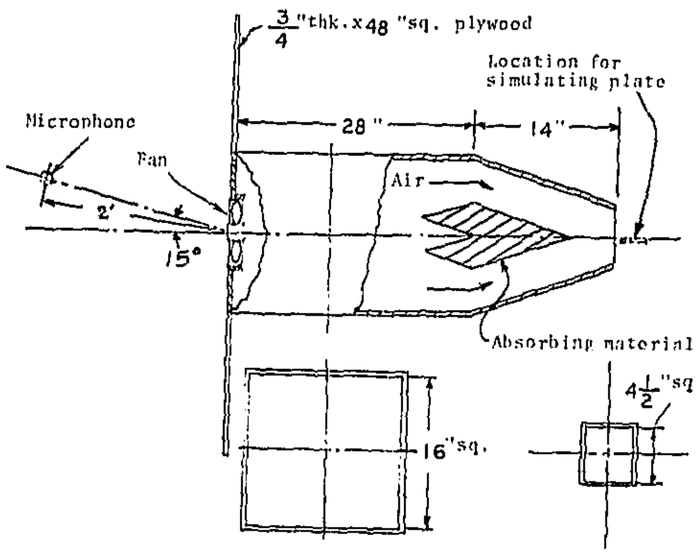

Fig. 2 Noise testing chamber for fan.

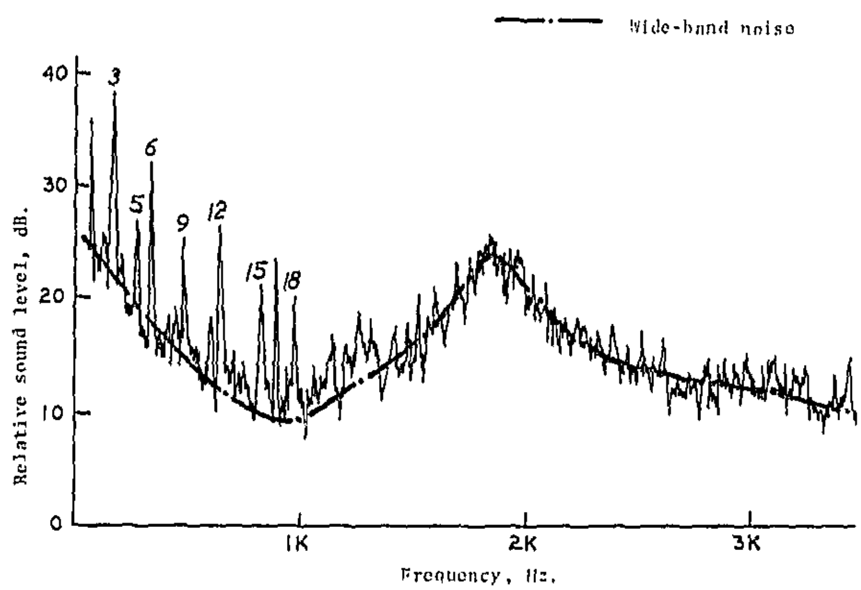

Fig. 3 10-Hz bandwidth sound spectrum of untreated fan.

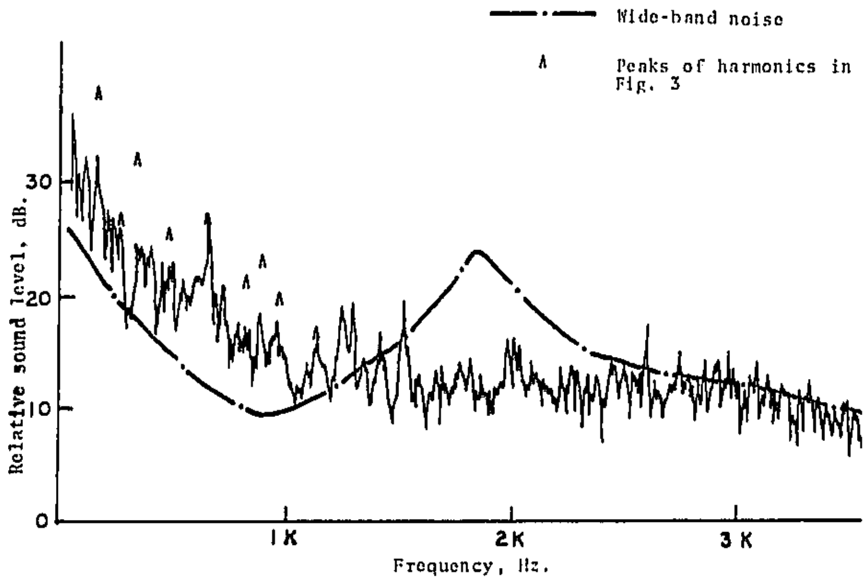

Fig. 4 10-Hz bandwidth sound spectrum of treated fan, porous material, 1/16"thk. fiberglass.

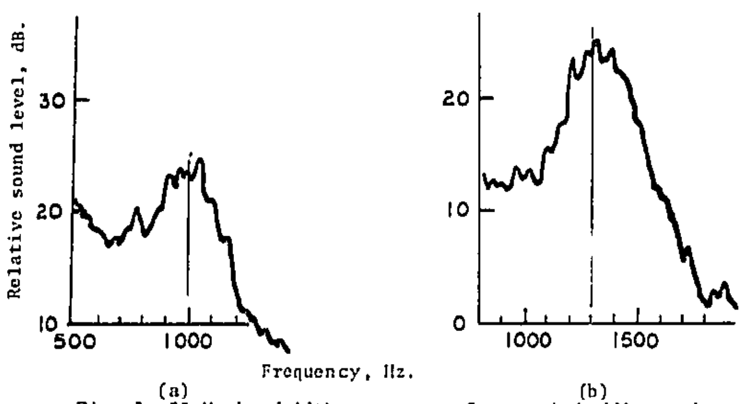

Fig. 5 50-Hz bandwidth spectrum of vortex shedding noise from a 1/16"thk.x1-3/4"span flat plate, (a) U_o=26. ft./sec. and (b) U_o=32. ft./sec.

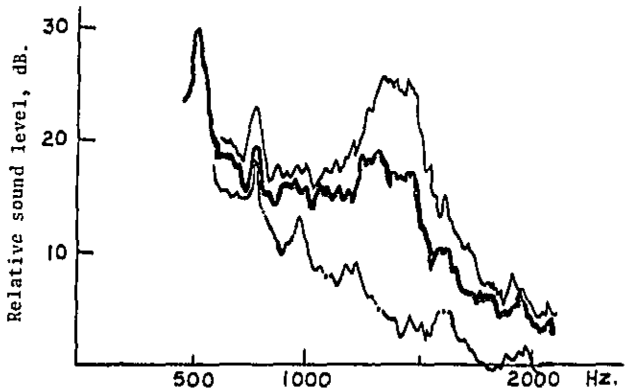

Fig. 7 50-Hz bandwidth sound spectrum of simulation tests, U_0=34. ft./sec. ——— .0625"thick plate. ━━━ .030"thk. plate with .03"thk fiberglass. —·— without plate.

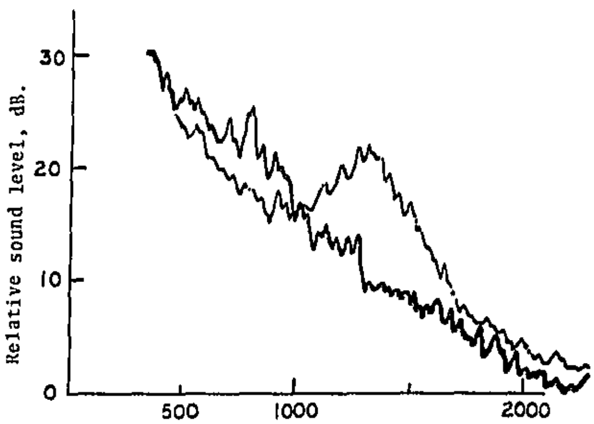

Fig. 6 50-Hz bandwidth sound spectrum of simulation tests, U_0=32. ft./sec. —·—·— .0625"thick plate. ━━━ .0625"thick plate with 1/16"thk. fiberglass.

REV-IT (A NOISE INVESTIGATION OF AN AMF HARLEY-DAVIDSON 61. CU. IN. SPORTSTER MOTORCYCLE)

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Subsidiary of AMF Incorporated
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This presentation will describe the techniques and results of a noise investigation of an AMF Harley-Davidson 61.cu.in. Sportster motorcycle. The initial sound level testing, by definition, was to determine objectively the noise levels created by the machine under actual test conditions and to further determine the degree of contribution by engine exhaust, engine intake, and mechanical sources of noise. The latter category encompasses all noise generated by other than the engine intake and exhaust. The individual test configurations will be described in a brief discussion and utilizing photographic slides. Slides showing the relative degree of contribution of the overall sound levels from the major noise sources during stationary testing are compared to the standard production machine during stationary testing. These values are then used to approximate the contribution of each individual major noise source during pass-by tests per SAE J331 test procedure. SAE J331 test procedure was used as the foundation for the present California Highway Patrol test code which restricts the bystander noise levels for new motorcycles offered for sale within the state. Sound pressure level spectra of the engine exhaust and induction noise will be presented describing the harmonic content. A brief discussion of the electronic instrumentation used for data acquisition and data reduction will also be included.

NOISE IN THE BREWING INDUSTRY - THE SOURCES, ITS CONTROL

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(1) INTRODUCTION

Noise in the brewing industry has received little or no attention in the open literature (1, 2); this is perhaps understandable since the word brewing is perhaps synonomous, at least in the layman's mind, with fermentation processes. However a very considerable portion of the industry is concerned with the packaging of the product, be it in cans, bottles or kegs. In the UK at least, the dominant forms of packaging are bottling and kegging. Both these forms of packaging have their characteristic noise spectra and indeed both produce significant continuous and impulsive noise levels.

This paper presents some typical noise levels resulting from both these forms of packaging and presents the resulting noise reductions obtained from various methods of noise control.

Whilst it is common for both keg and bottle filling to be carried out in the same factory space they can be conveniently dealt with separately.

(2) NOISE IN BOTTLING LINES

A modern bottling line consists of the following automated processes: depalletising, decrating, washing, filling, labelling, recrating and repalletising. Usually the depalletising and decrating, recrating and repalletising processes are carried out in an area external to the bottling hall. Consequently the machinery giving rise to noise generation within the bottling hall are washers, fillers, labellers and their associated conveyor system.

Bottling lines handle various sizes of bottle, the most common size being the half pint. Bottle handling rates on modern lines are typically in the region of 35,000 per hour and in some particular instances up to 80,000 per hour. The significant aspect of these numbers is that one particular processing machine on the line will be handling the total throughput of bottles hence conveyor speed will be high, so will bottle speed and consequently noise resulting from bottle impacts. There are, therefore, as might be anticipated, two significant sources of noise generation in any bottling installation, the actual processing plant and the bottle conveying system (between the plant items.)

Noise levels both from plant and that resulting from bottle pile-ups does therefore exhibit some variation as a result of different handling rates, however the values given in Table 1 are typical of those measured in practise.

Table 1. Measured 'A' weighted sound pressure levels (SPL) at operator location for particular plant items in a bottling hall (re: 2×10^{-5} N/m^2)

Location	SPL dBA
Bottle Washer (a) input side	94
(b) output side	95
Filler	100
Labeller	96
Bottle pile-up (1m from line)	93-97

Table 1 indicates that, for an acceptable operator exposure based on 90 dBA/8hr. per day exposure, a considerable hearing hazard exists. Further, the levels presented in Table 1 are the sum of both the direct and the reverberant field sound pressure levels, the latter of course being a function of the bottling hall geometry and surface finish and also machine density. In practice the reverberant level tends to range from the mid nineties to the upper eighties - 'A' weighted. Methods of affecting noise control obviously depend upon which component of the acoustic field is controlling the level at the operation location, that is, the relative levels of the direct and reverberant acoustic field, and indeed, the reduction required. By way of example, if the case of a single plant item in a room is considered, the methods available for reduction depend on conditions as follows:

(a) If both direct and reverberent field require reduction then enclosure (full or partial) may be the best practical means; alternatively, the introduction of absorption into the room and the use of screens (to control the direct field) may prove adequate depending on the spectrum.

(b) If only the direct field requires reduction then screens or partial enclosure may prove adequate.

Assuming for the present that it is impractical to introduce absorption in the bottling hall, then to economically reduce the reverberent field level necessitates the identification of the dominant sources. Further, since the acoustic power output of the machine is both a function of the radiated sound pressure level and its size, it is not necessarily obvious, from simple sound pressure level measurements, which are the dominant radiators of sound power and hence which plant items (if any) are controlling the reverberent field level.

Consequently, before embarking on a noise control programme it is desirable to establish the sound power level of the various plant items and also the areas of those plant items which are the controlling areas radiating the sound power. Using this information together with the direct field sound pressure level the most economical and practical means of noise control can be determined. In general these measures will be a combination of the introduction of absorption into the bottling hall, partial and total enclosure and screening techniques.

Table 2 gives some typical 'A' weighted power levels of the plant items cited in Table 1.

Table 2. 'A' weighted* sound power levels (PWL) of various plant items (re: 10^{-12} watt)

Plant Item	'A' Weighted Sound Power Level (PWL dBA)
Bottle Washer	112
Filler	114
Labeller	111
Bottle pile-up/metre of single line	104

The available methods of noise control are well understood and need not be entered into here.

Probably one of the most important factors evident from Table 2 is the sound power output resulting from bottle pile-ups which, in practice, is a common occurence; indeed, it may even be the dominant source. Modern machine characteristics are such that in the interests of efficiency, bottle accumulation to the machine is the norm rather than the exception. Consequently pile-up noise is significant. Accepting that bottle accumulation is necessary with concomitant noise production, it is suggested that the following techniques are employed in bottle handling in order to reduce (or eliminate) this source of noise. Bottle accumulation occurs in specific areas immediately adjacent to the input of the processing machine. These will be multitrack areas and hence will require convergence slats to feed the single line input into the machine. Usually the problem with

* 'A' weighted power levels may be employed since the majority of bottling halls finishes are such that the sound pressure level spectrum shape is not significantly affected as a result of frequency dependant absorption characteristics.

convergence slats is bottle jamming and hence the use of either electrodynamic or pneumatic bottle excitation systems in these areas to eliminate these problems. However, these devices in themselves cause considerable noise generation. Consequently, if pile-up noise and that resulting from anti-jam devices can be confined to a single area, then noise control enclosures may be employed, the control measure being confined to specific areas.

This has three obvious advantages:

(a) The bottle pile-up noise is controlled.

(b) The bottle anti-jamming device noise is controlled.

(c) Because of the use of exciters, bottle jamming does not occur and so the operator intervention is not necessary.

Since in general machine to bottle hall volume factors are constant, a typical maximum allowable total sound power level for an installation may be obtained. This may be subsequently divided between the various plant items and hence allowable power levels (as well as sound pressure levels) be specified as part of the bid requirement for various plant items. Such requirements are well established in the petro chemical field.

(3) KEGGING LINES

Racking lines are very similar in process terms to a bottling line, except of course kegs, rather than bottles, are being filled. The typical layout of a racking line is shown in figure 1. The noise sources can be described broadly as either pneumatic or mechanical in origin as distinct from being a result of the prime movers of the system.

Further, the nature of the sources is such that they are generally intermittent - apart from steam and pneumatic leaks - and therefore sound power levels are both difficult to obtain and of lesser application than for bottling lines. Rather than dwell on the detailed processes and their resultant noise levels, it is proposed to discuss only the broader aspects of the noise generating processes. Indeed reference to figure 1 clearly illustrates the noise levels before and after simple noise control measures have been taken.

The noise resulting from pneumatically operated processes is generally intermittent. However, the large number of pneumatic systems used throughout the line(s) result in a more or less continuous noise background from these sources. The principle of control is quite straight forward. In general the vents are choked and the noise output is dependent on the stream mechanical power. Two methods of noise control are available, either vent into an absorption silencer of suitable form or, fit a back pressure device. The principle of operation of the former is well appreciated; in the latter case noise reduction is obtained through unchoking the jet or vent and diffusing the flow over a larger area with the concomitant reduction in noise output. Reductions of 30dBA can be obtained in practice; the silencer being of the form of a porous plastic (or metal) sealed shroud which fits to the end of the vent. This form of silencing may prove troublesome with 'dirty' air supplies, blocking occuring with resulting malfunction of the pneumatic devices. In this case it may be more convenient to pipe all the discharges to a common absorption-type silencer.

The mechanical sources of noise result from either the mechanical excitation of the keg or the line structure. In the former case the noise emitted will, in general, be dependent on the induced vibration amplitude - as a result, for example, of the impact of the two kegs. The vibration amplitude will depend on the characteristics of the force and the vibrational response of the system; the emitted noise will also depend on how well the mechanical vibration system (in this case the keg), couples its vibration to the air, of prime importance in this case is the physical size of the vibrating object. The problem, therefore, is somewhat complex.

The noise emitted by empty containers is influenced by many factors. The kinetic energy or momentum of the moving container obviously plays a part, so that the more slowly the container is moving at the moment of impact the better. The material of construction of the keg also has significant effect. Further, as can be seen from figure 2, there is a measureable difference when the same kegs are impacted on a metal chain as compared to a plastic chain.

One of the most significant noise sources results from excitation of the keg by the conveyor system. The general processing procedure is such that waiting kegs are held stationary by arrestor arms and the conveyor runs continuously. In this configuration and with unlubricated conveyors, noise levels of up to 120dBA can be experienced. The provision of adequate lubrication will reduce the noise level to 78dBA for the same configuration; quite obviously adequate lubrication is of great importance. The material of construction of the kegs is also important: stainless steel containers emit less noise under these conditions.

(4) CONCLUDING REMARKS

Noise from bottling and racking installations is excessive. It is a result of designing without due regard to noise control both in general plant design and container handling techniques. Relatively simple but well understood techniques can be applied which will successfully reduce the noise levels below those which are considered to be hazardous to operators. The implementation of these techniques at the design stages of new installations should present no problems once the design engineer is made aware of the possibilities.

ACKNOWLEDGEMENT

The author would like to thank Whitbread & Co. Ltd., for making some of the enclosed information available for publication.

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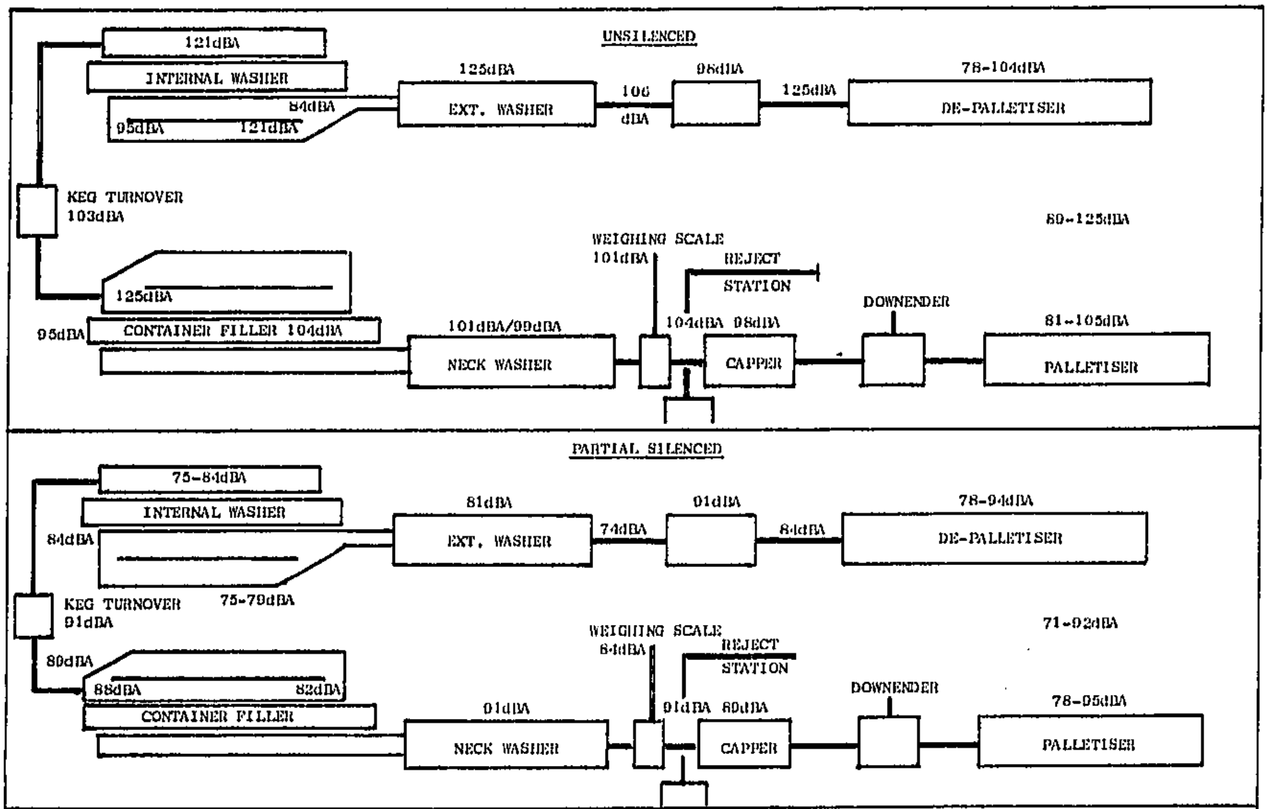

Fig. 1. Schematic of typical racking line showing measured 'A' weighted sound pressure levels before and after silencing.

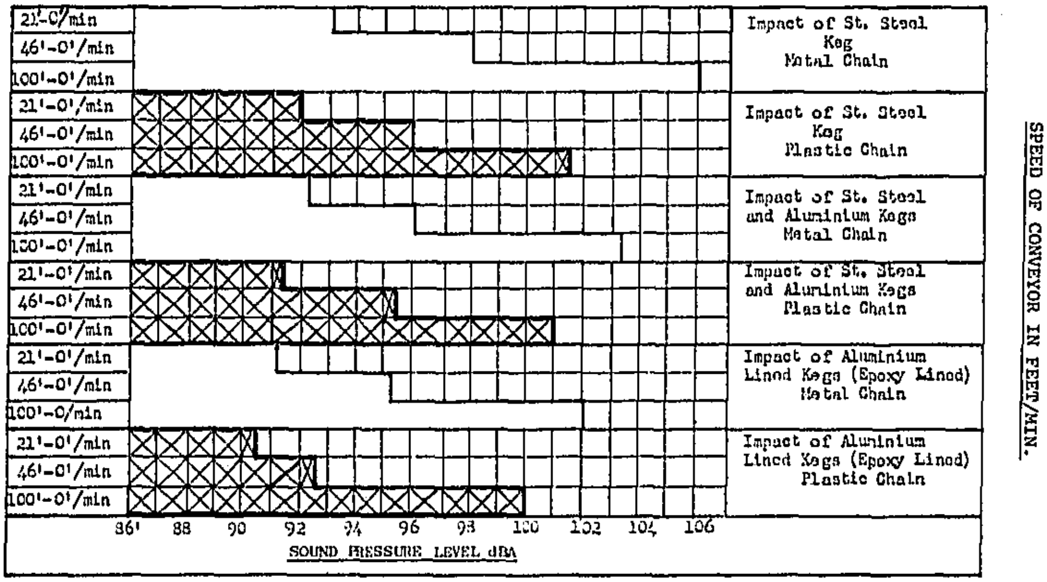

Fig. 2 'A' weighted sound pressure levels resulting from the impact of kegs.

AIRCRAFT AND AIRPORT NOISE

DESIGN TRENDS FOR NOISE CONTROL FOR AIRCRAFT POWER PLANTS -THE NASA QUIET ENGINE

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INTRODUCTION

In order to help relieve the airport community noise problem NASA initiated the Quiet Engine Program several years ago. The objective of the program was to develop engine noise reduction technology suitable for use on subsonic, conventional-takeoff and landing-type aircraft. Significant test results have recently been obtained with the two experimental Quiet Engines built in this program. The Quiet Engine Program and the recent test results are the subject of this paper.

DISCUSSION

Noise Sources

The turbofan engine, which is the type of engine commonly used in the current transport fleet, has two major noise sources. These noise sources, which are illustrated in figure 1, are jet noise and turbomachinery noise. Let's discuss jet noise first. The turbofan engine has two exhaust jets. One is the result of the air flowing through the engine core. The second is due to bypass air or air pumped by the fan around the engine core. The core jet noise is generally dominant because the core jet is usually of higher velocity and jet noise is primarily a function of jet velocity. Since the jet noise is caused by the turbulent mixing of the jet with the ambient air and this mixing takes place some distance downstream of the engine, it is very difficult and generally impractical to employ acoustic treatment to suppress this type of noise.

The turbomachinery noise is a result of the unsteady flow processes and/or shock waves that occur locally internal to the machinery. Because the fan is the largest machinery component in the engine, it is the largest noise source. Secondary sources are the engine core compressor and turbine. Since machinery noise is generated internally and has to escape through the engine inlet or exhaust nozzles, it is more amenable to acoustic suppression.

Quiet Engine Design Features

A number of noise reduction features were incorporated into the Quiet Engines as shown in figure 2. A high bypass ratio engine was chosen to reduce jet velocity and consequently jet noise and to obtain near optimum performance. Other features incorporated for fan noise reduction were as follows: A relatively large rotor-stator spacing of 2 rotor chords was employed. Reference 1 indicates that increasing the spacing from 0.15 to 2.0 rotor chords reduces sideline maximum noise by 5-6 PNdB. A choice of rotor tip speeds was available for the fan design. Low tip speeds have been found to produce less noise while high tip speed fans can improve airplane economics by reducing engine weight, but they require additional noise suppression to achieve equally low noise output. Both approaches were evaluated in the program. Finally, a noise-governed optimum ratio of fan stator to rotor blades was employed. This ratio was 2.25. In addition to design features aimed at low fan noise production, the fan noise can be reduced further by the addition of sound absorbing liners to the inlet and outlet ducts. This was also done for the Quiet Engines.

Overall Quiet Engine Program

In figure 3 the major elements of the Quiet Engine program are presented along with a schedule. Following several Quiet Engine design studies, a contract with the General

Electric Company was initiated in mid 1969 for the design, fabrication and testing of two Quiet Engines. A major part of the General Electric effort included the design, fabrication and testing of three full-scale fans. This was done in order to provide the best fans considering both aerodynamic and acoustic performance. Engine A contained the low speed "A" fan and Engine C incorporated the high speed "C" fan. Both engines are undergoing test programs that include aerodynamic and noise tests at General Electric and additional noise and altitude performance testing at the Lewis Research Center. In parallel with the engine program a contract with the Boeing Company, Wichita Division, was initiated to provide NASA with an acoustically treated, flight-type nacelle for Engine A. Initial tests have been conducted with this nacelle at Lewis.

Quiet Engine Test Results

In the following discussion some of the test results that have been obtained with the two Quiet Engines are presented. Data will be presented for both the baseline or unsuppressed engines and for Engine A with the acoustic nacelle added.

Baseline engines. - The two baseline Quiet Engines have been tested at the General Electric test facilities. A photograph of this facility with Engine A installed is shown in figure 4. The tall structures in the foreground and background are used to support the microphones used to measure engine noise and they are located on a 150 ft arc around the engine. A cross-section of the baseline Quiet Engine configuration is shown in figure 5. This configuration includes a simple bell-mouth type inlet. A small amount of acoustic treatment is built into the engine frame in the immediate vicinity of the fan and engine core compressor inlet. The treatment is of the resonator multiple-degree-of-freedom type as shown inset in the figure.

The aerodynamic characteristics of the two Quiet Engines are shown in figure 6. For comparison the JT3D engine is included. This is the engine that is used in the later versions of the Boeing 707 and DC-8 type transports. It can be seen that the thrust levels of the Quiet Engines are slightly higher but in the same class as the JT3D used in the Boeing 707 and DC-8 airplanes. A major difference is noted in bypass ratio where the Quiet Engines are in the range of 5 to 6 while that of the JT3D is about 1.4. The core jet velocities of the Quiet Engines are seen to be about 2/3 to 1/2 of that of the JT3D.

A comparison of the perceived noise directivity of the two baseline Quiet Engines is shown in figure 7 for the approach speed condition. As noted in the figure the noise levels are nearly identical. However, if we compare them at the takeoff engine speed, as shown in figure 8, there is a significant difference. The high speed engine (C) is front end noise dominated where as engine A is back-end noise dominated. The Engine C perceived noise level is greater by a maximum of 7 dB in the front end and about 3 dB overall. The reason for the higher front end noise at takeoff engine speed for engine C is the supersonic relative speed of fan rotor tips. The resulting shocks formed at the blade tips produce a "multiple pure tone" noise that adds significantly to the front end noise level. A more detailed discussion of multiple pure tone noise generation can be found in reference 2. The high speed engine, therefore, will require additional acoustic treatment in order to bring its noise level down to that of the low speed engine. Future testing with engine C will determine the extent of this treatment penalty.

Engine A with acoustic nacelle. - A cross-section showing engine A with the acoustic nacelle added is shown in figure 9. The nacelle has a flight-type inlet and acoustic treatment on the fan inlet and outlet duct walls. In addition, three acoustically treated splitters are located in the fan inlet and one in the outlet duct. The total weight added to the engine by the acoustic treatment is about 1500 pounds. However, in a flight-weight design the weight increment could be reduced by as much as 50 percent. A photograph of engine A with the Boeing acoustic nacelle is shown in figure 10. The engine and nacelle are shown mounted in the thrust stand of the Lewis engine noise test facility. The inlet rings and center body are observable in the photograph. A comparison of the perceived noise directivity of engine A in the baseline configuration and with the acoustic nacelle added is shown in figure 11 for the takeoff engine speed. The maximum noise level of the baseline configuration is 98 PNdB at the 120° angle. It can be seen that the acoustic treatment reduces the maximum noise levels by 6-7 PNdB. In the front end of the engine the reduction is greater and it amounts to 10 PNdB. Reductions in perceived noise for the approach speed condition were quite similar. A comparison of sound pressure level spectra is useful for making a more detailed evaluation of the performance of the acoustic treatment. This type of plot is shown in figure 12 for the 50°

angle to the inlet position. It can be seen that below a 500 hertz frequency the acoustic treatment does not reduce the noise level. This is as expected since this low frequency range is presumably controlled by jet noise. However, above 500 hertz where the fan noise is usually dominant the acoustic treatment is seen to significantly reduce the noise levels. For example, the blade-passage-frequency tone which occurs at 2000 hertz has been completely removed from the spectrum; this amounts to at least an 18 dB reduction in sound pressure level. The absence of the annoying fan tone is also apparent to listeners who observe the engine during test. Additionally a 5 to 10 dB reduction is noted in fan broadband noise. Therefore, the acoustic performance of the nacelle appears to be quite effective.

In addition to acoustic performance the affect that the treatment has on the engine aerodynamic performance is also of importance. In figure 13 the effect the acoustic treatment has on engine thrust is presented. The upper curve is the baseline or untreated configuration while the lower curve is for the acoustically treated nacelle. A reduction in engine thrust, which amounts to 5 percent at the takeoff speed of 3260 rpm, results when the acoustic treatment is added. Accordingly, airplane economics will be adversely affected due to the performance loss and also the weight increase associated with the use of large amounts of acoustic treatment. An estimate of the economic penalty was made for a 3-engine, medium range transport, and it was found that the acoustic treatment configuration used in these tests would increase airplane "direct operating costs" by about 5 percent. Lesser amounts of treatment would, of course, result in a smaller economic penalty.

Flyover Noise Comparison

It is interesting to estimate what impact the Quiet Engine technology would have if it were employed on typical current aircraft. Calculations of this nature were made and the results are shown in figure 14 and they are also compared to a typical four engine aircraft and the current FAA noise regulations. The noise levels are presented in terms of the standard FAA noise measuring unit. This noise unit is referred to as "effective perceived noise level" or EPNdB. The data are presented for the standard FAA noise measuring stations of takeoff and landing. It can be seen that the DC-8 type aircraft noise levels are substantially above the FAA's FAR-36 noise regulation for new aircraft of this weight. When the noise level for a DC-8 aircraft is calculated with flight weight untreated Quiet Engines, FAR-36 regulations are surpassed by 7 or 8 dB. These noise levels are also noted to be about 20 dB below the current DC-8 noise levels. Further if an acoustically treated nacelle is added to the Quiet Engines, the aircraft produces an additional 7 dB less noise. For the later case the noise levels are about 15 dB below FAA noise regulations.

CONCLUSIONS

The major conclusions that can be drawn from the Quiet Engine Program are as follows:

1. Most importantly we have developed and demonstrated engine noise reduction technology which, if applied to future aircraft, can bring about a substantial reduction in aircraft noise levels.

2. There will be an associated airplane economic penalty which, we will be studying in more detail in the future and hope to be able to reduce.

3. We are encountering new noise floors, as engine noise levels are lowered, towards which our future research can be directed in order to make further progress in aircraft noise reduction.

4. And finally, the information we have generated in the program will be useful in establishing future aircraft noise regulations.

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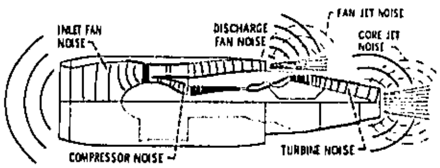

Figure 1. - Turbofan noise sources.

JET NOISE

HIGH BYPASS RATIO 5-6, GIVES LOW JET VELOCITY & LOW JET NOISE

FAN SOURCE NOISE

LARGE SPACING BETWEEN FAN ROTOR & STATOR; REDUCES INTERACTION NOISE

LOW TIP SPEED, 1160 FT/SEC, REDUCES FAN NOISE PRODUCTION

HIGH TIP SPEED FAN, 1550 FT/SEC, REQUIRES ADDITIONAL SUPPRESSION FOR LOW NOISE BUT IMPROVES ENGINE WEIGHT

OPTIMUM RATIO OF FAN STATOR TO ROTOR BLADES

FAN NOISE SUPPRESSION

SOUND ABSORBING LINERS IN FAN INLET & DISCHARGE DUCTS

CS-54046

Figure 2. - Quiet engine design features.

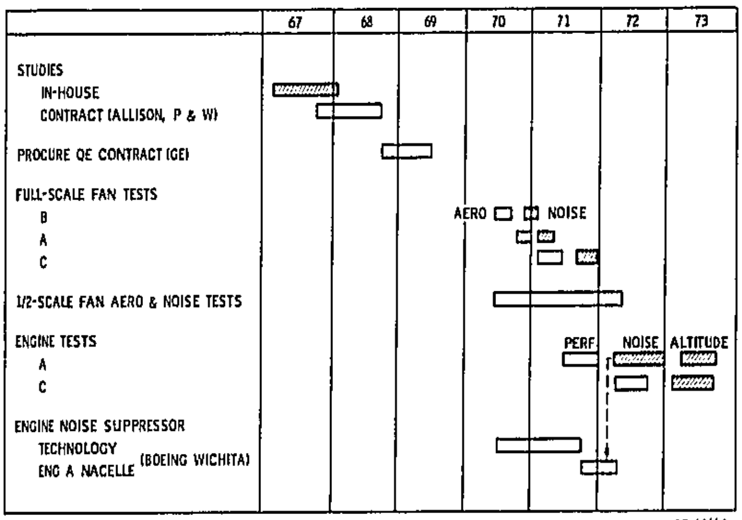

Figure 3. - Quiet engine program.

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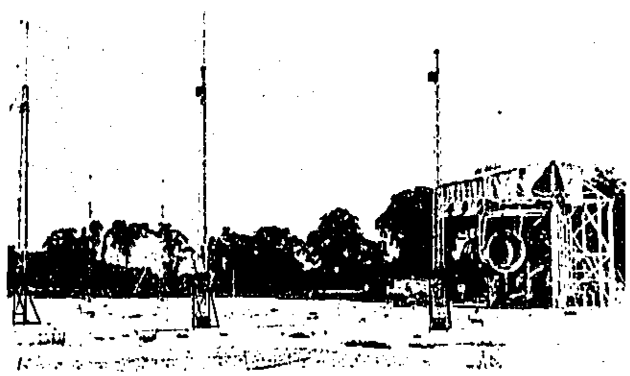

Figure 4. - Engine A on test.

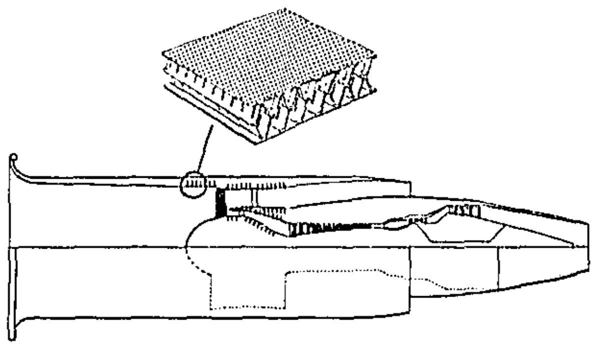

Figure 5. - Quiet engine A baseline configuration.

ENGINE	QE-A	QE-C	JT3D
CRUISE THRUST, LB	4 900	4 900	4 450
TAKEOFF THRUST, LB	22 000	22 000	18 000
FAN TIP SPEED, CRUISE	1 160	1 550	1 430
FAN PRESSURE RATIO, CRUISE	1.5	1.66	1.74
BYPASS RATIO, CRUISE, FT/SEC	6.1	5.1	1.42
CORE JET VELOCITY, FT/SEC	1 180	850	1 600

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Figure 6. - Engine characteristics.

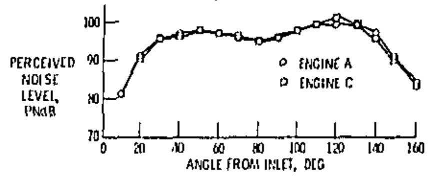


Figure 7. - Baseline engine perceived noise directivity.

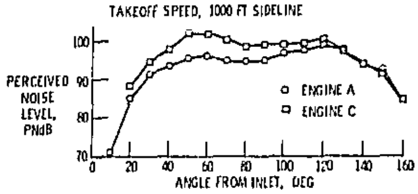
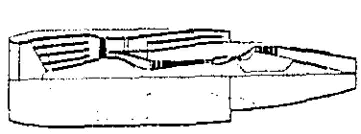


Figure 8. - Baseline engine perceived noise directivity.



NACELLE ACOUSTIC TREATMENT

	INLET DUCT	EXHAUST DUCT
AREA, SQ FT	353	362
WEIGHT, LB	846	666

Figure 9. - Quiet engine A with acoustic nacelle.

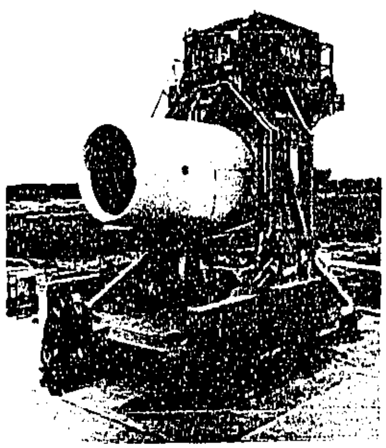


Figure 10. - Quiet engine "A" with acoustic nacelle.

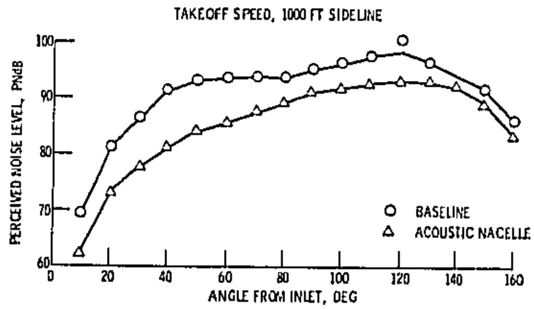


Figure 11. - Quiet engine perceived noise directivity.

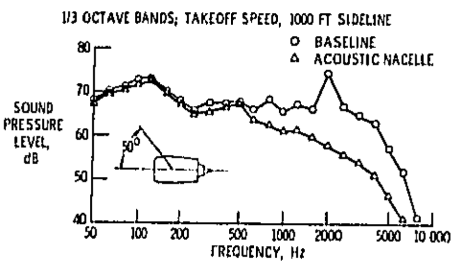


Figure 12. - Quiet engine sound spectra.

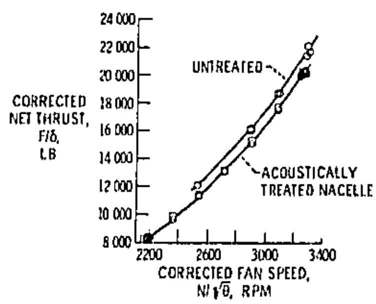


Figure 13. - Effect of acoustic treatment on engine "A" thrust.

	TAKEOFF	APPROACH
	EPNdB	
DC-8	116	118
FAR-36	104	106
BASELINE QUIET ENGINE A	97	98
QUIET ENGINE A WITH ACOUSTIC NACELLE	90	89

Figure 14. - Flyover noise comparison.

CS-63277

ON THE ROLE OF THE RADIATION DIRECTIVITY IN NOISE REDUCTION FOR STOL AIRCRAFT*)

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ABSTRACT

The radiation characteristics of distributed randomly fluctuating acoustic sources when shielded by finite surfaces are discussed briefly. A number of model tests using loudspeakers as artificial noise sources with a given broadband power density spectrum are used to demonstrate the effectiveness of reducing the radiated noise intensity in certain directions due to shielding. In the lateral direction of the source array noise reductions of 12 dB are observed with relatively small shields. The same shields reduce the backward radiation by approximately 20 dB. With the results obtained in these acoustic model tests the potentials of jet noise reduction of jet flap propulsion systems applicable in future STOL aircraft are discussed. The jet flap configuration as a complex aerodynamic noise source is described briefly.

INTRODUCTION

In the development of future STOL aircraft the noise produced by the aircraft deserves special attention since low noise levels will have to be achieved. In addition to its attractive aerodynamic features the jet flap principle bears the potential of sizable aircraft noise reductions, Ref. 2. These are due mainly to a modification of the jet noise radiation characteristics reducing the noise levels in areas at the side of and below the airplane. The desired noise radiation directivity is the result of the nonaxisymmetric jet flow and particularly the shielding property of the flap over which the plane jet is exhausted from a slot nozzle. Since the turbulent flow regions in the jet are producing the noise the jet flap as an aerodynamic noise source can be described as a randomly fluctuating acoustic source distribution moving with a certain convection velocity over a fixed surface of finite dimensions. While being convected downstream the correlation length between adjacent source elements increases continuously, whereas the source strength decreases rapidly downstream of the jet core region. The growth of a second intensive turbulent mixing region downstream of the flap trailing edge makes this aerodynamic noise source even more complex. The observed far-field radiation is the result of a superposition of the radiation from the different source regions. Provided most of the noise is produced in the flow over the flap a sizable noise reduction in the side line and downward direction can be expected. This is demonstrated by means of acoustic model tests described in the next section. In a following section real jet flap noise sources are compared with the acoustic model.

NOISE RADIATION FROM DISTRIBUTED RANDOM SOURCES

General Concepts. In practical noise reduction the sources encountered are quite often extended instead of being concentrated. That is, the overall noise power density spectrum involves acoustic wavelengths in the order of typical dimensions of the source or smaller. Consequently in general a

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certain directionality of the radiation pattern results.

The fluctuations of an extended source emitting a random signal may be
a) coherent (e.g. random vibrations of a piston without baffle),
b) noncoherent (e.g. thermal motion of a surface), or
c) partially coherent (e.g. turbulent wall jet).

The degree of coherence is determined by the correlation length of the source fluctuations along the surface. For a coherent random motion of a radiating surface it is immediately conceivable that the resulting radiation pattern will depend on the $k\ell$-values of the upper and lower limiting frequencies of the noise power density spectrum, if k is the wave number and ℓ is the characteristic dimension of the source. For partially coherent fluctuations, where the correlation length between adjacent source elements becomes very small, in general a non-uniform radiation pattern will result also, dependent on the relative size of the radiating surface. The radiation will become more directional as the degree of coherence increases.

A comprehensive analytical approach to this problem seems to be outstanding. In Ref. 1, Ch. 7, an approximate formula for the sound intensity in the far field of a randomly vibrating surface in an infinite baffle is derived. This, however, is not simply applicable to our present radiation problem of a "unilateral randomly vibrating surface without baffle of given geometry, overall power density spectrum, and correlation length". In the problem discussed in Ref. 1 the lateral and backward radiation is always zero.

Acoustic Model Tests. The radiation problem described above can be modelled fairly easily with an array of compact noise sources of known power density spectrum located on one side of a shield. In order to obtain an idea on the efficiency of shielding the noise emenating from a compact and distributed random source, a series of experiments has been conducted using a number of identical loudspeakers as source elements. Three different size square-shaped shields were used with sides ℓ = 40 cm, 80 cm, and 120 cm. The power density spectrum of the emitted noise signal had a lower and upper limiting frequency of f_1 = 280 Hz and f_2 = 18 kHz respectively. The resulting $k\ell$-values range from approximately $k\ell = 2$ to $k\ell = 350$. The geometry of the test setup is shown in Fig. 1, and the shape of the noise source power density spectrum and the corresponding 1/3-octave band spectrum are shown in Fig. 2a.

The following measurements were made: For a single (compact) source placed at position (1) on the shield and five sources placed at positions (1) to (5) (distributed source, coherent vibrations), the one third octave band sound spectra were observed in the far field in a plane vertical to the shield and recorded at $\theta = 0°$, 90° (lateral direction), 180° (backward direction). Further, the full directivity pattern was measured for broadband radiation ($\Delta f \underset{\sim}{} 18$ kHz) and 1/3 octave band radiation at the upper and lower band of the spectrum.

The change of the noise spectrum with varying direction caused by the shielding surface is demonstrated in Fig. 2b. This figure shows the noise spectra measured in the $\theta = 0°$, 90°, 180° direction for the simulated distributed source attached to the medium size shield. Similar spectrum changes were observed also for the other source configurations. The expected influence of the spectrum changes on variations of the D-weighted noise level as compared with the OSPL, however, could not be proved in these tests. An example is shown in Fig. 3b, $\theta = 90°$, where the observed linear OSPL noise reduction is compared with the D-weighted level of noise reduction. The absolute noise reduction as a result of the shielding is sizable for compact and distributed noise sources. The results are shown in Fig. 3a,b. For noise reduction in the lateral direction, the size of the shield shows less influence than expected, as long as $k\ell > 2$. Fig. 4 shows the observed directivity pattern for the compact and the distributed shielded noise sources.

AERODYNAMIC NOISE FROM JET FLAPS

The jet flap in the form of an internally blown flap (also known as jet-augmented-flap) represents an attractive propulsion concept in the development of STOL aircraft Refs. 2 and 3. A successful application, however, will critically depend on the aerodynamic noise generated in this type of exhaust flow.

The overall aerodynamic noise produced by a jet flap will generally originate in three different regions of the exhaust flow. These are, compare Fig. 5, the primary mixing region (1), the secondary mixing region (2), and the turbulent boundary layer on the flap surface (3). The flap may become effective for shielding the noise originating from the sources (1) and (3); noise generated in the secondary mixing region (2) will not be shielded by the flap. This complex noise source can be described as an "distributed random, partial coherent, convected source, shielded by a finite surface". The degree of coherence depends on the scale of turbulence in the flow. The convection of the source predominantly causes an inclination of the radiation characteristic towards the flow direction. Except for the coherence characteristics and the source convection, the acoustic model discussed above resembles the jet flap as noise source. Therefore, even with relatively small size flaps a sizable noise reduction could be expected in the side-line and downward direction for the noise from the sources (1) and (3). Since the radiation from region (2) will not be affected by the flap, the overall noise reduction performance of a jet flap due to radiation shielding will strongly depend on the relative strength of the source region downstream of the flap trailing edge.

The jet flow produced by an internally blown flap may assume three different configurations dependent on the slot nozzle height over flap length ratio. For short flaps the jet core will extend over the flap trailing edge, Fig. 5a. For medium size flaps the jet core will terminate in the vicinity of the flap tip, Fig. 5b, and for long flaps the core ends over the flap, Fig. 5c. One might expect that the second mixing region as noise source becomes less efficient as the flap length increases, because less undisturbed jet flow energy is available for the mixing process. Especially in the last case, Fig. 5c, the second turbulence layer is the result of the mixing of a highly turbulent flow with the surrounding air.

Since turbulence intensity and mean shear determine the strength of the aerodynamic source an estimate of the source strength per unit volume can be obtained from appropriate flow measurements. With data taken from Ref. 4, the product of maximum turbulence intensity and maximum mean velocity gradient measured in a model jet flap flow is plotted in Fig. 6 in arbitrary units versus the distance from the nozzle exit. This indicates that region (2) is a relative intensive aerodynamic noise source in all three jet configurations. This result is also supported by other observations, see Ref. 4.

As a result it appears that only if the noise from region (2) can be made a small portion of the noise from sources (1) and (3), the full noise reduction benefits of jet flap propulsion systems will be obtained.

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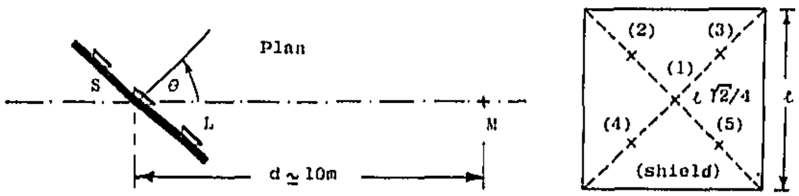

FIG. 1. Geometry of acoustic model test set up, shield dimensions (ℓ = .4,.8,1.2m) and location of noise sources (1) to (5). L = loudspeaker, S = shield, M = microphone.

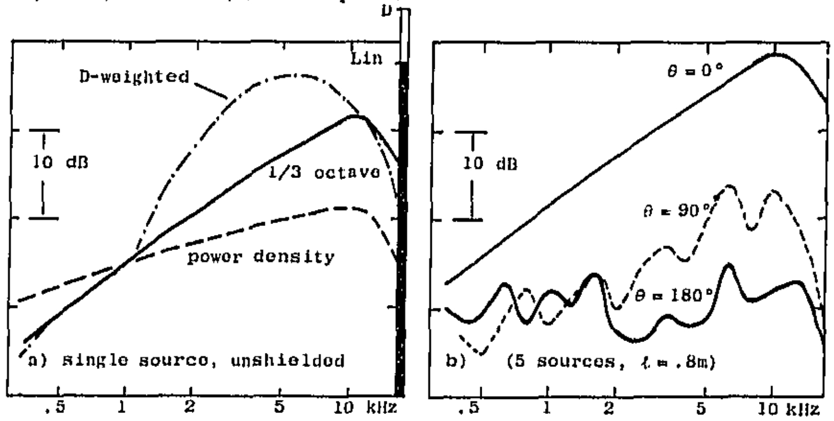

FIG. 2. a) Noise spectrum of compact unshielded source (single loudspeaker). b) Noise spectra from distributed shielded source observed in the forward (θ=0°), lateral (θ=90°), and backward (θ=180°) direction.

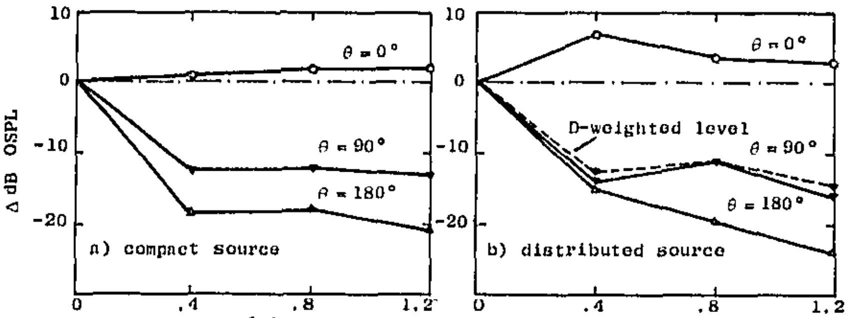

FIG. 3. Overall sound pressure level (OSPL) variations measured in three different directions as function of the shield size. The reference level (0 dB) originates from an unshielded source of identical strength. a) compact source at position (1), b) distributed source at positions (1) to (5).

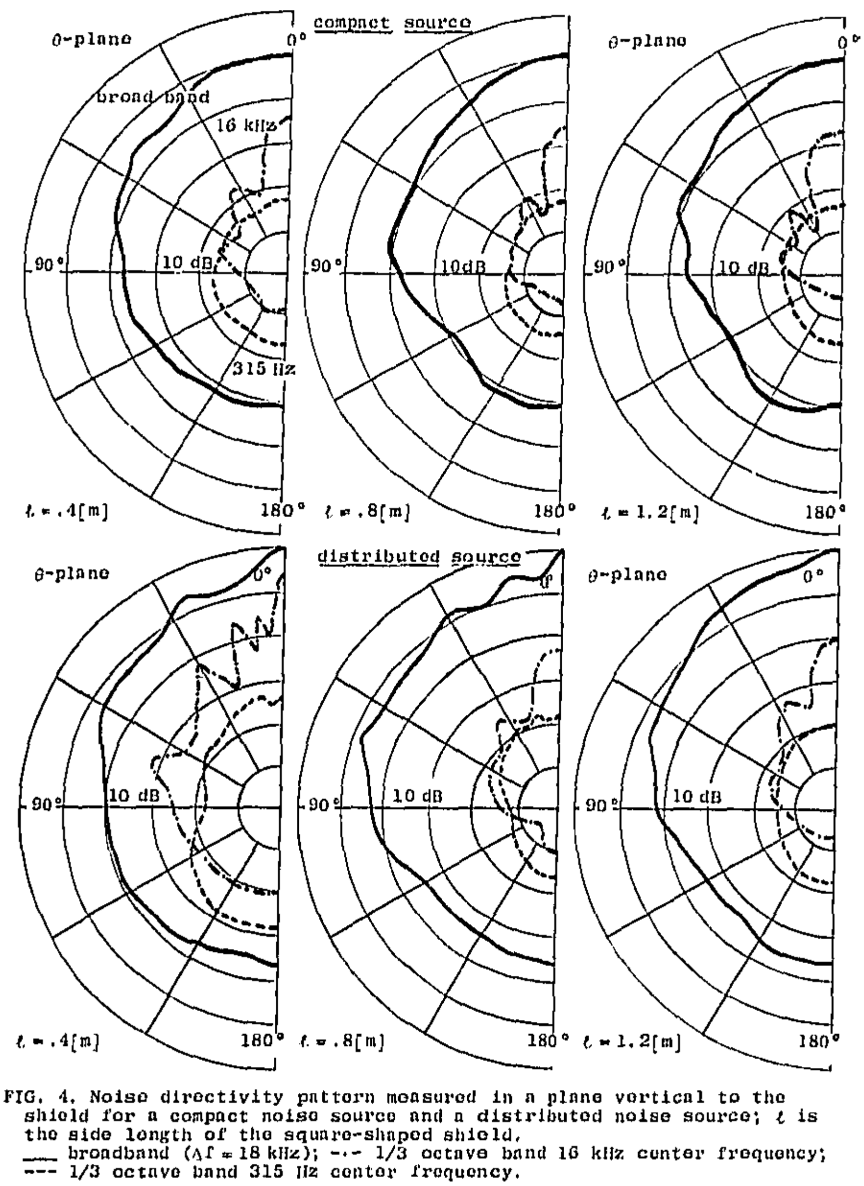

FIG. 4. Noise directivity pattern measured in a plane vertical to the shield for a compact noise source and a distributed noise source; ℓ is the side length of the square-shaped shield.
___ broadband ($\Delta f = 18$ kHz); -·- 1/3 octave band 16 kHz center frequency; --- 1/3 octave band 315 Hz center frequency.

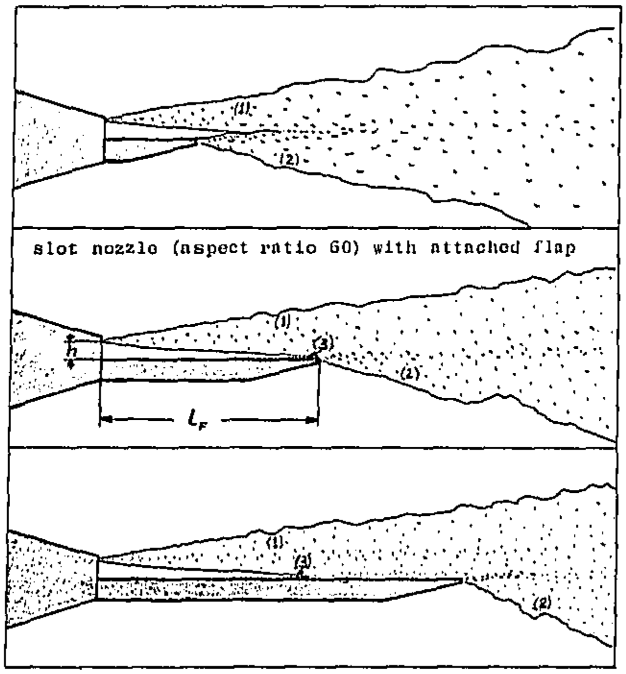

FIG. 5. The three typical flow configurations encountered in an internally blown flap system. The three distinct turbulence noise source regimes are: (1) primary mixing region, (2) secondary mixing region, (3) turbulent boundary layer.

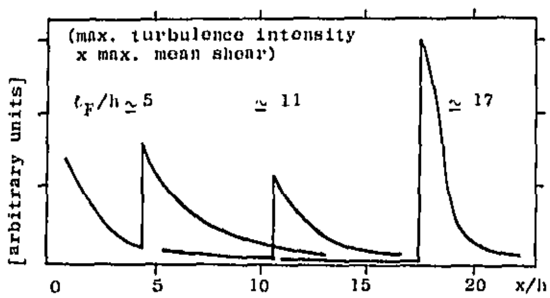

FIG. 6. Product of measured maximum turbulence intensity and maximum mean shear versus distance from nozzle exit for the three jet configurations shown in Fig. 5. The curves are a qualitative measure of the noise source intensity per unit volume, h height of slot nozzle, x downstream distance from nozzle exit.

THE ULTIMATE NOISE BARRIER -

FAR FIELD RADIATED AERODYNAMIC NOISE

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A NEW COMMERCIAL AIRCRAFT CONCERN

During our recent NASA sponsored Advanced Technology Transport (ATT) Study Contract, low noise engines and acoustically treated nacelles were studied extensively for use on the next generation of large, subsonic commercial transport aircraft. One of the objectives of these studies was to define future commercial transport aircraft capable of achieving noise levels equal to, minus 10, and minus 20 EPNdB below current Federal Aviation Regulation (FAR) 36 noise level requirements. Methods were found to reduce engine noise to nearly 20 EPNdB below the FAR 36 levels. Since this was a total system study, other potential noise sources were also evaluated. This brought to attention the subject of far field radiated aerodynamic noise, i.e., the noise produced by the airframe itself as it passes through the air. There are several possible aerodynamic noise sources involved here, such as boundary layer turbulence, fuselage wake, wing vortex shedding, etc. This problem had previously been of concern mainly in connection with aural detection of aircraft for quiet surveillance military uses. One of the more recent efforts along this line was a Lockheed-California, Navy sponsored program to measure the noise of several small gliding aircraft. The results of this program pointed to wing trailing edge vortex shedding as the primary noise source. This phenomena is of course similar to aeolian tone production due to a wire in an airstream, or more fundamentally, the whole class of noise problems related to vortex shedding from rigid bodies immersed in a fluid flow.

The results of the Lockheed-Navy program provided an empirical relationship for noise prediction based on several aircraft structural and aerodynamic parameters. When these empirical relationships are applied to new large ATT type subsonic aircraft weighing on the order of 600,000 pounds, the OASPL, spectrum shape, and peak frequency can be calculated. Figure 1 is the resulting calculated aerodynamic noise spectrum (for an aerodynamically clean wing) at the FAR 36 landing noise point which is 1.0 nautical mile from landing threshold. In terms of perceived noise, the calculated clean airplane aerodynamic noise level is 96.4 PNdB. To account for the noise effects of flaps down, landing gear down and wheel wells open, an estimated 5 PNdB is added giving 101.4 PNdB. To obtain noise levels in FAR 36 units of EPNdB, the empirical A.I.A. relationship for converting PNdB to EPNdB was used, due to the absence of data concerning the time history of aerodynamic noise. This procedure gives a net correction of -4.4 PNdB so that the new landing configured aircraft noise level is 97 EPNdB. In a similar fashion, takeoff sideline and flyover aerodynamic noise can be calculated in FAR 36 terms. Figure 2 is a table showing the calculated noise levels and the corresponding FAR 36 noise requirements. Examination of this table shows that landing noise is the most critical. If aerodynamic noise was the only source on landing, the lowest noise level that could be achieved would be 11 EPNdB under FAR 36. If engine noise were equal to aerodynamic noise, the total would be somewhere in the vicinity of 100 EPNdB, and 8 EPNdB under FAR 36 limits would be as low as could be achieved.

If these calculations prove to be accurate for large subsonic aircraft, then the objectives of the joint DOT/NASA CARD study (Ref 1) for future aircraft noise reductions would not be met unless some measures are taken regarding far field aerodynamic noise. It is not known exactly how accurate the current calculation procedure is for large aircraft since calculated aerodynamic noise levels are on the order of 10 EPNdB or more below current engine noise levels.

DEVELOPMENT OF BASIC AERODYNAMIC NOISE TECHNOLOGY

Since the state-of-the-art in this area is still rather limited and not generally well known, let's review briefly some of the highlights of previous work which is related to the suspected primary noise source, i.e., the unsteady aerodynamic forces (fluctuating lift and drag) resulting from wing trailing edge vortex shedding.

Throughout recorded history, various references are given to stringed musical instruments which produce sound as air, or wind, is blown across the strings. The sound produced has been referred to as aeolian tones since about the 17th century. The first quantitative investigation of this kind of sound was performed by Strouhal in 1878 (Ref. 2). His experiments involved the tone of sound from a moving cylindrical wire. His classic finding determined that the frequency of sound was independent of length or tension in the wire and was proportional to velocity and inversely proportional to diameter. He also noted that at certain speeds, tone production was increased by apparent wire vibration. Shortly after Strouhal's original experiments, Rayleigh (Ref. 3) speculated that aeolian sound production was related to the instability of vortex sheets and had noted that peak sound directivity was normal to the air flow.

Essentially no further work along these lines was done until 1914 when Kruger and Lauth (Ref. 2), who upon examining Strouhal's work and Benard's 1908 experiments concerning alternating vortices behind an object in running water, postulated that the production of parallel alternating vortex rows (which we now know as a Karman vortex street) was the source of sound. They further pointed out that if the frequency of vortex production matched the natural frequency of the body causing the vortex production, then the body would vibrate resonantly, amplifying the sound. Little did they know that in 26 years time (1940), this very vortex production and structural resonance phenomena would completely destroy the Tacoma Narrows suspension bridge in the state of Washington (Ref. 4). Work of a more modern nature began in 1924 (Ref. 2) when Richardson undertook a wind tunnel test program to determine the flow regimes where aeolian tone production from wires and cylinders occurred, and then in 1925 (Ref. 5), when his investigations turned to similar experiments of airfoil shapes. In a discussion of this work, after Richardson had presented it to the Physical Society of London, a Professor Hopewood stated that he had found that tying several short pieces of cord or streamers to a taut wire being pulled broadside through the water reduced the "thrumming" sound. Then a Dr. Tucker said he had observed "high frequency notes" caused by aircraft wing struts and wires on oscillograph records. Hopewood's comment is probably the first documented concerning reduction of vortex noise by physical devices and Tucker's comment is probably the first documenting aerodynamic noise produced by aircraft structure (both comments in Ref. 5).

In 1934 Graham (Ref. 6) began searching in the animal world for clues to airfoil noise reduction in regard to propellers. He studied the aerodynamic peculiarities of owls, known for quiet flight, as a possible source of information. One of Graham's postulations was that the trailing edge fringe noted on all primary wing feathers probably breaks up vortex formation at the wing trailing edge. In 1935, Stowell and Deming (Ref. 7) determined the basic vortex peaked broad-band noise spectrum for rotating cylindrical rods, and they found that sound produced was proportional to air velocity to the 5.5 power ($V^{5.5}$). In 1944 (Ref. 8), Yudin published his work on the properties of vortex noise of various shaped rotating rods. He generally found a V^6 relationship. Also in 1944 Krzywoblocki (Ref. 9), who was interested mainly in structural effects, studied the nature of vortex formation and Strouhal number variation in the trailing edge wakes of airplane wings. In the late 1940's and 1950's, considerable work was done by several investigators primarily at the Langley NACA (now NASA) Research Center concerning the definition of and prediction of propeller noise. Typical of this era was Hubbard's 1953 work (Ref. 10) on transport aircraft propeller rotational and vortex noise prediction. His vortex noise calculation procedure had blade area to the first power and V^6 terms.

During the 1950's and 1960's, numerous investigations provided better understandings of flow-structure interaction noise, and in particular fan noise and helicopter rotor noise. During most of this period, little work was done directly related to wing or aircraft far field radiated noise. However, it was in the late 60's when this subject became popular again. At Lockheed Missiles and Space Company in 1968, noise measurements were taken under a Schweizer SGS-232 sailplane (Ref. 11). The primary objectives were determination of velocity and gross weight effects. Consequently data were taken at 60 to 100 knots, and from 1250 to 1750 pounds gross weight. Altitude at the measurement points was 75 to

100 ft. Typical overall and octave band sound pressure levels varied with V^5 to V^7, averaging very nearly V^6. The effect of weight over the narrow range investigated was inconclusive.

In 1969, the USAF Flight Dynamic Laboratory conducted aural detection related noise measurements (Ref. 12) on three sailplanes, a Schweizer SGS 232, SGS 233, and a Libelle. These experiments covered a gross weight range from 552 to 1340 pounds. To determine the effects of sailplane velocity and gross weight were again major objectives. Measuring altitudes were generally from 50 to 150 feet and sailplane airspeeds of 50 to 130 feet per second. As in the case of the earlier Lockheed tests, gross weight effect tests were inconclusive. The airspeed effect however showed a strong velocity to the sixth power dependency. A generalized overall sound pressure level (OASPL) relationship was found to be:

$$\text{OASPL} = 10\ \text{LOG}_{10}V^6 - 10\ \text{LOG}_{10}R^2 + 10\ \text{LOG}_{10}A - K \quad \text{dB}$$

where: V = sailplane velocity in Ft/sec; R = altitude in feet; A = wing turbulent area in ft^2 (wing area less laminar flow area); K = constant = 42 (average for three sailplanes).

In 1970 the Lockheed-California Company undertook a Navy sponsored program to measure the noise of several gliding aircraft (Ref. 13) in order to get a better understanding of aero noise phenomena and to provide an empirical method of noise prediction. The five aircraft were a Prue-2 sailplane, Cessna 150, Aero Commander, Douglas DC-3, and a Convair 240, covering a gross weight range from 1,300 to 39,000 pounds. Measurement altitude ranged from 300 to 800 feet generally with some sailplane measurements as low as 180 feet, and airspeed varied from 58 to 192 knots. The noise measurements were analyzed and normalized in a form convenient for rapid noise prediction. The resulting empirical relationship for overall sound pressure level was:

$$\text{OASPL} = 10\ \text{LOG}_{10}\left(\frac{v^4}{h^2} \times \frac{W}{c_L} \times \frac{C}{b}\right) + K \quad \text{dB}$$

where: v = airspeed in knots; h = altitude in feet; W = gross weight in pounds; c_L = coefficient of lift; C = average wing chord in feet; b = wing span; K = constant which includes environmental variables, equal to 8.4 for standard day and for aerodynamically "clean" configurations.

Gross weight, W, is essentially equal to lift, which is proportional to v^2. Thus the overall SPL is related to v^6. The peak frequency relationship for the measured broad band spectra were found to be:

$$f = S\frac{v}{t}$$

where: f = peak frequency, Hz; S = Strouhal number equal to 1.85; v = airspeed in knots; t = mean wing thickness in feet; and the spectrum shape has previously been shown in Fig.1.

The noise phenomena observed showed a peak in amplitude when the aircraft was directly overhead, i.e. major aerodynamic forces perpendicular to microphone. The dipole like characteristics, frequency relationships, and other considerations, lead to the conclusion that the most predominant of the several aerodynamic noise sources is wing trailing edge vortex shedding. (This was the program that provided the basic methodology for the ATT noise calculations.)

CURRENT AND RECENT NOISE AND NOISE REDUCTION INVESTIGATIONS

In the area of airfoil noise reduction there are several recent investigations that are of interest. In 1970, at the University of Tennessee Space Institute under an Air Force contract, a combined aerodynamics, acoustics, and bionics study was conducted on birds in an attempt to find mechanisms to suppress noise from aircraft (Ref. 14). These studies were based on the silent flight of owls. In this program, owl wings were aerodynamically tested in a wind tunnel and noise measurements were made of a live, gliding owl. Noise reduction effects were stated to occur due to (1) large areas of attached flow due to vortex sheet generation caused by feather serrations near the leading edge, (2) downward frequency shift caused by compliant surfaces, and (3) reduction of trailing edge velocity gradients due to the distributed porosity of the wing.

Work sponsored by NASA Ames Research Center in 1970-71 has investigated owl wing-like leading edge serrations on airfoils and the resulting effects on noise and aerodynamics (Ref. 15). They found noise could be reduced and aerodynamic efficiency was actually improved due to the vortex sheet production by the serrations which reduced flow separation, and suppressed the formation of large trailing edge vortices. Also in 1971, a Pennsylvania State University program, under a Navy contract, has pursued the owl wing leading edge serration or comb idea in a study of airfoil noise reduction related primarily to helicopter rotors (Ref. 16). Reductions in rotational and vortex noise were found. In another program recently completed, a United Aircraft Research Laboratory study under an Army contract, investigated the vortex shedding noise of an isolated airfoil (Ref. 17). One of the conclusions from this program was that some of the broadband noise previously thought to be vortex noise may not be caused by vortices, but by some other mechanism.

Currently there is considerable effort in progress or planned by many organizations on the subject of airfoil noise and noise reduction. One of the research areas we recommended as a result of the ATT program was the measurement of aerodynamic noise from a large aircraft to better define the magnitude of the problem to new large aircraft and to give more insight into the effects of flaps, landing gear, etc., as well as the basic wing noise. Such a program is now underway at NASA Edwards Flight Research Center. Much of the current industry STOL aircraft noise work related to the externally blown flap concept is also applicable since blown flap noise appears to be a special case of airfoil noise. Likewise much of the work on helicopter blade and fan blade vortex noise reduction is also applicable to wing noise reduction. Within the Lockheed Corporation, several continuing and new investigations into a better definition of wing aerodynamic noise phenomena are underway. It is also thought that other related projects are in progress within the industry.

CONCLUSIONS

The ultimate noise barrier for new, large commercial aircraft appears to be the aircrafts' own self-generated aerodynamic noise. The predominant noise source has been tentatively identified as the unsteady aerodynamic forces associated with wing trailing edge vortex shedding. The basic technical problem is not new, however, the state-of-the-art in the particular area of wing airfoil noise is currently rather limited. Better understandings of the basic noise generation mechanisms are needed before the true magnitude of the noise problem is established and before an efficient noise reduction technology can be established.

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FIGURE 1

FAR FIELD AERODYNAMIC NOISE 1/3 OCTAVE SPECTRUM

FOR AN AERODYNAMICALLY CLEAN CASE AT LANDING SPEED, 370' ALTITUDE; 600,000 LB. AIRCRAFT

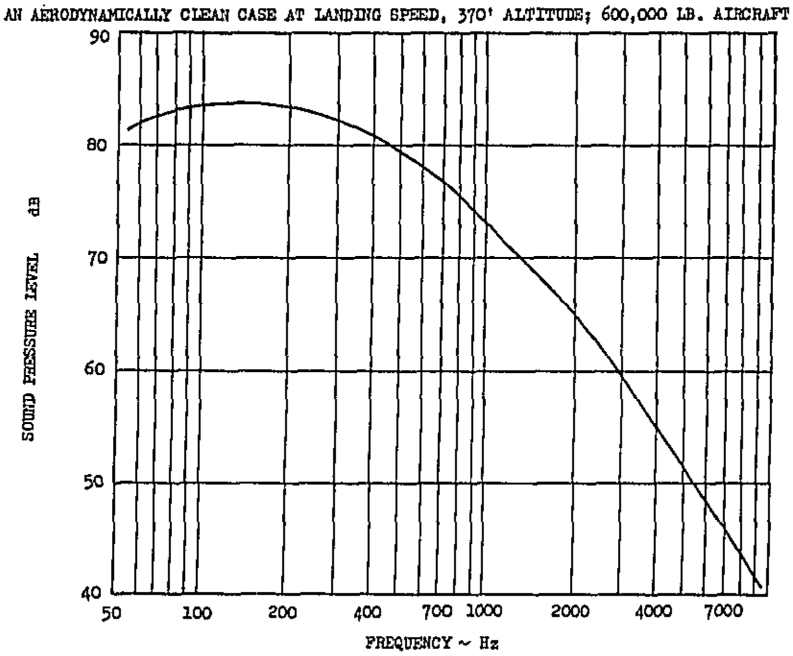

FIGURE 2

FAR FIELD AERODYNAMIC NOISE

CALCULATIONS IN FAR 36 TERMINOLOGY

FAR 36 LOCATION	FAR 36 LIMIT	CALCULATION
Takeoff Flyover (3.5 N.M.)	108 EPNdB	92 EPNdB
Takeoff Sideline (0.35 N.M.)	108 EPNdB	86 EPNdB
Landing Flyover (1.0 N.M.)	108 EPNdB	97 EPNdB

For a 4 engine 600,000 lb. aircraft in the appropriate T.O. and landing configurations.

NOISE CERTIFICATION OF A TRANSPORT AIRPLANE

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INTRODUCTION

Airplane flyover noise measurements are made for a variety of reasons -- to provide data in support of design development, to determine detectability of military aircraft, to monitor operations at an airport, to demonstrate noise guarantees; and a similar variety of instrumentation and procedural requirements exist. The introduction in late 1969 of Federal Aviation Regulation (FAR) Part 36, "Noise Standards: Aircraft Type Certification" (Ref. 1), followed in 1971 by the ICAO International Noise Standards (Ref. 2), added another important reason for conducting flyover noise measurements, and increased significantly the required sophistication of the instrumentation and the complexity of the procedures.

A round robin airplane noise measurement exercise held in February 1968 at Brown Field in Southern California, sponsored by SAE Committee A-21 (Ref. 3) and participated in by leading airframe and engine manufacturers, disclosed disturbing differences in measured flyover perceived noise levels reported by the participants for the same noise source under the same conditions. And yet each participating company used its best equipment, experienced personnel, and standard measurement and analysis procedures (Ref. 4). In an attempt to minimize such variability in results, FAR Part 36 delineated, in great detail, instrumentation specifications, airplane performance restrictions, and measurement and analysis requirements. However, experience now shows that there are still problems and uncertainties, and there is still ample need for improvement in the techniques of measuring airplane flyover noise.

It was during the time that the government was developing, discussing, and reviewing the ideas that culminated in FAR Part 36 that the wide-bodied jet transports came to the drawing boards. Among them was the Lockheed L-1011 TriStar, for which low flyover noise levels were established early as a basic design consideration. The L-1011-1 noise certification program will be presented here, covering instrumentation, test site, data reduction and analysis, FAR Part 36 demonstration for the Federal Aviation Administration (FAA), and results.

FAR PART 36

The need to establish aircraft community noise standards was expressed by the government in the requirements of the Supersonic Transport (SST) competition in the middle sixties, starting with a general requirement that the SST be no noisier than the current jet transports, and later introducing specific noise limits to be met at specific locations. In September 1966 a letter to Industry from the FAA's Associate Administrator for Development, the "Blatt" letter, announced the introduction in Congress of legislation to authorize regulation of aircraft noise by the FAA and proposed, for discussion, the Phase III SST competition noise limits. The authority was granted by Public Law 90-411, July 21, 1968, and on January 11, 1969, a Notice of Proposed Rule Making (Ref. 5) presaged Federal Aviation Regulation Chapter III Part 36, adopted November 3, 1969, and effective December 1, 1969.

FAR Part 36 establishes effective perceived noise level (EPNdB) limits to be met at the three reference locations shown on Figure 1: 3.5 nautical miles from brake release for takeoff; along a sideline parallel to the runway after lift-off, at 0.35 nautical miles for four-engine turbojet aircraft, and at 0.25 nautical miles for aircraft with fewer than four engines; and at one nautical mile from threshold for approach along a three degree glide slope, corresponding to a 370 foot height. For specified operating conditions, on an ISA (International Standard Atmosphere) plus 10°C (77°F)/70% relative humidity day, at sea level, an airplane must meet the noise limits of Figure 2, the limits being determined by the maximum takeoff gross weight of the vehicle.

Since the demonstration measurements to show compliance are not likely to occur on a reference day at reference conditions, Part 36 spells out the allowable test conditions. For example: relative humidity not higher than 90% or lower than 30%; ambient temperatures not above 86°F and not below 41°F; wind not above 10 knots; no temperature inversion or anomalous wind conditions that would significantly affect the noise level. Measurement and analysis systems requirements include: high frequency preemphasis must be added if limitations of dynamic range make it necessary; a windscreen must be used if wind is in excess of 6 knots; dynamic range capability must be at least 55 dB; and so on. In addition, FAR Part 36 spells out in great detail the measurement, EPNL calculation, and the correction and normalization procedures required to determine the three (or more) noise levels that show compliance with Noise Standards: Aircraft Type Certification.

The L-1011 TriStar noise certification program is typical of such programs, and a review of the plans, the instrumentation, the experiences, and the problems should be of interest and, hopefully, of value to anyone concerned with aircraft flyover noise.

L-1011 NOISE CERTIFICATION PROGRAM

The final planning and the equipment procurement for the L-1011 TriStar acoustical measurement program occurred after publication of FAR Part 36, so its provisions and requirements were given full consideration, in addition to the normal flight test noise measurement requirements.

Test Site

The Lockheed-California Company Commercial Engineering Flight Test Center is located at Palmdale, California, about 50 miles northeast of Los Angeles. In evaluating this location, as well as other less convenient locations, for noise certification, it was found, of course, that no site met all requirements. Ideally a site should be at sea level, flat, unobstructed, with temperatures generally about 77°F, relative humidities about 70 percent, winds below ten knots, with low acoustical ambient, and with no excessive periods of precipitation. Although Palmdale is at about 2500 foot elevation, it satisfactorily meets all the other requirements during at least part of the day most days of the year. The area is flat and unobstructed (Figure 3), the elevation difference between the runway and the 3.5 nautical mile point is only one foot, and the runway is clearly visible from this point. Temperatures and humidities in the early morning hours are well within the allowed range; prior to 0900 it is not unusual to have relative humidities of 60 percent or more. Winds at this same time of day are not only, usually, below the allowed 10 knots, but are so calm that it is possible to make takeoff and landing tests off the same end of the runway, eliminating the need to move microphones and instrumentation from one side of the airport to the other. The airfield, surrounded largely by alfalfa fields and uncultivated desert, is characterized by as low an acoustical ambient as one is ever likely to find in the vicinity of an airport. Rain or snow are seldom encountered, and the field is never closed for visibility problems.

Palmdale, during the early morning, does usually experience a temperature inversion. However, an analysis of the effects of temperature gradients on flyover noise propagation (Ref. 6) defined the limits that would "significantly" affect the noise level, and inversions more extreme than those encountered during certification would be needed.

Actually, the greatest site-associated problem encountered at Palmdale was animal damage to the electrical cables from the gnawing teeth of desert rodents. This problem was solved to a large extent by a changeover from cable with gold-colored insulation to cable with black insulation.

The flat, unobstructed character of the test area facilitated the laying of cable and allowed driving from one point to another for set-up, calibration, etc. Combined with the usual use of the area at one end of one runway permitted full advantage to be taken of the convenience of recording all data at one central "command" location.

Equipment and Instrumentation

The acoustical measurement and analysis equipment for L-1011 flyover noise was selected to meet the specifications of FAR Part 36 and to satisfy the general needs of an airplane noise program. The data acquisition set-up, illustrated in Figures 3, 4, and 5, consists of:

a. Microphone systems -- including tripods and remote signal conditioning
b. Central signal conditioning, monitoring, and recording equipment
c. Calibration sources for frequency and amplitude.

The portable microphone system used at each measurement location includes one-half inch microphone, windscreen, high-frequency pre-emphasis filter, cable loss compensation, and support tripod. This equipment, when the microphone is used at 90 degree incidence, conforms with applicable sections of IEC 179 (Ref. 7) for acoustical and electrical specifications for "Precision Sound Level Meters." The signals are transmitted from the microphone units to the central acoustics van via 200 ohm, balanced, twisted, and shielded cables.

The central recording system includes a 14-channel FM tape recorder with low-level, differential inputs, oscilloscope monitoring, and Lockheed-built attenuators. The microphone lines are terminated at the van and the signals are attenuated as required for the low-level tape recorder inputs. Data signals may be monitored at the record input and at the playback output for waveform. Acoustical monitoring is done on simultaneous playback with an amplifier/speaker unit. One-third octave-band spectrum monitoring is accomplished with a real time analyzer in the van. Synchronized IRIG (Inter Range Instrumentation Group) B time code from the airplane is recorded on a direct mode channel of the tape recorder.

The dynamic range of the system is optimized by the vernier adjustment of the remote preamplifier gain and by use of the pre-emphasis filter for recording, with de-emphasis on playback or analysis. The one-third octave-band signal-to-noise ratios of the system vary from 60 to 70 dB with reference to the sine clipping level, plus that achieved with the use of pre-emphasis.

Laboratory calibration of the microphone systems to determine frequency response is performed at regular intervals, utilizing an electrostatic actuator. In the field a pistonphone calibration before and after each test series establishes the amplitude sensitivity of the system at a single frequency (250 Hz). The system electrical response calibration is also carried out in the field, after each test series, with a pink noise insert voltage. An insert-voltage microphone adapter allows the microphone cartridge to remain in the circuit, maintaining electrical characteristics as they are during noise measurement.

During any noise measurement exercise meteorological data are obtained in the vicinity of the van -- wet and dry bulb temperatures at four feet above the ground and wet and dry bulb temperatures and wind speed and direction at ten meters. This information is printed on a paper tape along with a time record. A complete scan through all channels is repeated about once each minute. Variation of temperature with altitude over the flight range is recorded on the test airplane.

A variety of tracking schemes have been used for determining airplane position with respect to a microphone. For the basic FAA certification demonstration, radar tracking was used. Simpler photographic tracking has been found satisfactory, using either ground cameras photographing the airplane at overhead flyby or on-board cameras photographing ground markers.

Briefly reviewing the data analysis, the data in analog form on the FM tapes are processed using a real time spectrum analyzer controlled by a small computer, and are converted to one-third octave-band sound pressure levels in digital form on a tape compatible with the digital computer of the Flight Test Data Center. The pink noise calibrations on the analog tapes, recorded in the field, are also digitized for use in correcting the noise data. The acoustical response of the microphone and windscreen are handled separately as tabular input.

Tracking data (in the form of x, y, z, and t values) and weather data (temperature and relative humidity) are also entered into the computer, by means of punched cards. This information is used to normalize the measured noise data to FAR Part 36 reference conditions or to any other reference condition that may be specified. The digital computer, working with reference and test day weather, reference and test day flight paths, and the measured noise data, then makes all corrections and calculations necessary to show compliance with FAR Part 36 or to provide other noise analysis. The computer program has been written to calculate the subjective noise measures, perceived noise level (PNL), tone-corrected perceived noise level (PNLT), effective perceived noise level (EPNL), and A weighted noise level (dBA).

Certification Demonstrations

Although FAR Part 36 only requires actual takeoffs and landings for demonstrating noise, for L-1011 certification a series of level flyovers were also conducted, at two heights above the runway and at a series of thrusts and at several flap angles, to confirm the noise versus thrust, or other engine parameter, relationship developed in the course of the flight test program. The takeoff and the landing tests were conducted in two separate series, in order to stay within the weight window appropriate to each operation. Numerous meetings had been held with FAA personnel prior to the actual tests to discuss and agree on the test, measurement, and analysis procedures. During the tests airplane performance and configuration and noise calibration and measurement were monitored by the FAA. After the tests several meetings with the FAA were required to review the calculations, corrections, and normalizations and to approve the final results.

Problems and Solutions

Difficulties encountered during the noise certification program included "pseudo" high-frequency tones, ground reflection tones, inadequate dynamic range (signal to noise ratio), and transient ambient signals.

Disposing of the last item first, unanticipated tone corrections suddenly appearing in the 4 to 8 KHz range on some records were discovered to be caused by birds which, when only several feet from the microphone, were considerably louder than the L-1011 at one to two thousand feet. By watching a playback through a real time analyzer while listening to the tape output at the same time, these bird tones were readily identified and the times of occurrence noted. The computer was then instructed to ignore these tones and eliminate the tone corrections from the calculations.

Other tone problems were associated with the sharp spectrum roll-off at high frequencies and with cancellation/reenforcement effects resulting from noise path-length differences between direct and reflected rays reaching the microphone. The tone correction procedure of FAR Part 36 calculates a pseudo tone when large sound pressure level changes occur in successive one-third octave-bands (over about 10 dB per band). As there is often a very rapid drop in level at high frequencies, for the longer propagation distances at takeoff and sideline, a large tone correction penalty is assessed. A narrow band analysis will show that no tone actually exists in the frequency range under consideration, and such analyses were employed to justify eliminating these pseudo tones from the L-1011 EPNL's.

The low-frequency tones caused by cancellation/reenforcement do appear in a narrow band analysis, however, so other means were necessary to prove that they were not aircraft-generated but were due to microphone-ground spacing. The simplest method involved a flush mounted microphone at ground level. Comparing the spectrum from such a microphone with that recorded at the same time for a microphone at the conventional four feet clearly showed the nature of the tone. In addition, knowing the geometry, the frequencies for this reflection phenomenon can be calculated. First, second, and third harmonics of reenforcement and cancellation frequencies were computed and shown to match exactly each depression and peak of the measured spectra. The FAA then permitted discarding any tone correction at one of these frequencies.

The most serious difficulty encountered was that of inadequate dynamic range, even with the 80 to 90 dB with pre-emphasis, particularly at the more distant sideline microphones. Because of the relatively low acoustical output of the source and the substantial atmospheric absorption, the high frequency portion of the airplane noise signature received at a distant microphone would be below the noise floor of the measurement system, and this portion of recorded spectrum would be incorrect. Even for relative humidities of 60 to 70 percent, a dynamic range of over 100 dB would be required to measure the complete spectrum at 0.25 nautical mile. At lower humidities, but still within FAR 36 limits, this requirement approaches 150 dB. When the incorrect portion of test day spectrum is corrected to reference day conditions, levels that are recorded to be 50 dB or more above the true noise signature are corrected to levels that are the same amount above the true reference day spectrum. Computed reference day PNL's can be 30 to 40 PNdB above what would be measured if the airplane flew on a reference day.

This problem is not unique to the L-1011 and has been encountered before. Among the solutions that have been suggested are:

- Elimination of frequencies above 5 KHz from PNL calculations. Generally these bands have little impact on the noisiness.
- Elimination of the sideline noise measurement requirement. Sideline noise is relatively less important, and excessive effort required for its determination may not be warranted.
- Extrapolation to sideline distance, and possibly to 3.5 nautical mile flyover distances, from close-in measurements. There is precedent in the present corrections from measured flyover distances to reference profiles.

The solution to the problem evolved for the L-1011 certification was to take data at the sideline as well as at a close-in centerline microphone. Then, using test day conditions, the close-in spectrum was extrapolated to the sideline microphone distances, and the resultant shape of the high frequency portion of this spectrum was applied to determine the shape of the sideline-measured spectrum where it dropped below the instrument noise floor. This still required that sideline noise measurements be made, with the additional problem of picking peak noise from an extremely flat noise versus position characteristic.

RESULTS

Many years of research, development, and design, followed by over a year of flight test noise measurements, culminated in three mornings of FAR Part 36 demonstration flights and a month of data analysis and report preparation to give the results of Figure 6. The noise goals established early in the L-1011 program had been achieved. The FAA approved Airplane Flight Manual noise levels for the RB.211-22C engine, and the anticipated levels for the RB.211-22B engine, to be available in 1973, are significantly lower than the FAR Part 36 limits and are substantially lower than the noise levels from earlier large jet transports.

Experience in noise certifying the Lockheed L-1011 TriStar has shown that improvements are needed in the techniques and procedures for measuring airplane flyover noise and for noise-certifying aircraft under Federal Aviation Regulation Part 36. However, the FAA's noise certification regulation has forced a standardization and distinct improvement in flyover noise measurement technology and has established improved, but achievable, noise standards for aircraft.

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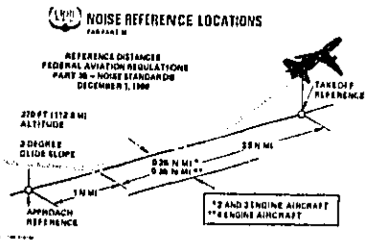

FIGURE 1

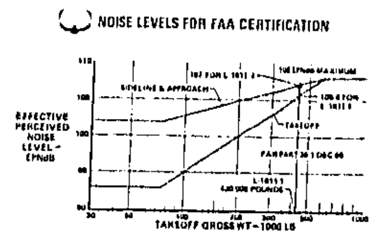

FIGURE 2

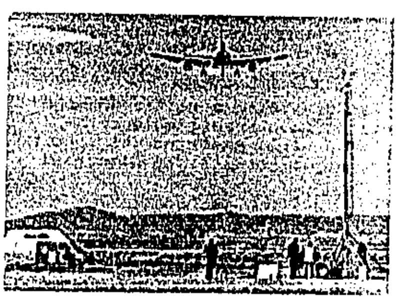

FIGURE 3. Central Recording Station - Instrument Van and Weather Tower

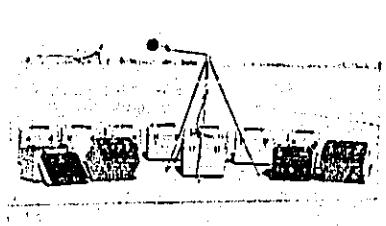

FIGURE 4. Microphone Systems

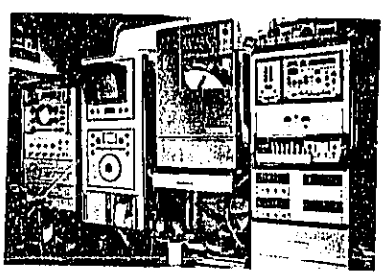

FIGURE 5. Recording and Monitoring Equipment in Acoustics Van

L-1011-1 TRISTAR FAR PART 36 NOISE LEVEL

CONDITIONS	FAR PART 36 LIMITS (EPNdB)	L-1011-1 TRISTAR RB 211-22C (UNTIL MARCH 1973) (EPNdB)	L-1011-1 TRISTAR RB 211-22B (AFTER MARCH 1973) (EPNdB)
TAKEOFF 430,000 LBS.	106.8	97	96
SIDELINE	107.0	96	96
APPROACH 368,000 LBS.	107.0	103	103

FIGURE 6

A PROPOSED LITTORAL AIRPORT

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GENERAL

For years, offshore airports have held glittering promise of vast areas with unobstructed flight patterns and no complaining residents below the approach and departure paths of the large commercial jetliners - hundreds of sea acres free for the asking.

Such seadromes are possible in over sixty American and European cities, as in Los Angeles, San Francisco, Seattle, Boston, New York, and even Chicago. But in Los Angeles, study after study has not led to such a port, including a 1967 proposal to build an airfield at Santa Monica Bay, a 1969 design of an offshore landing facility at Long Beach, and the subject plan for an intercontinental airport in San Pedro Bay initiated in 1971, whose acceptability however is still pending.

The major reason for the non-construction was generally a matter of high initial cost, while the minor reasons consisted in unpredictable maintenance expense, undeveloped technology in respect to instrumental landing systems for two- and three-segmental approaches, uncertain fog dispersal methods, and others.

But two new elements have entered the arena of interest to make offshore airports even more desirable if not necessary - class action law suits on grounds of nuisance, and added emphasis on ecology or environmental protection. Thus, on April 28, 1972, the California Supreme Court ruled that residents under the approach and departure pattern of jet aircraft may use the grounds of nuisance to sue for compensation for physical injury and emotional disturbances caused by the operation of city-owned airports. The ruling reversed a Los Angeles County trial court's dismissal of a 12.5 million class action suit brought by 700 persons against the city of Santa Monica, California, on grounds that its airport constituted a nuisance. For 1972, the total amount of suits against LAX has been estimated as 4.2 billion dollars - triple the cost of an offshore airport at San Pedro Bay.

And in respect to whatever use may be made of land for a mainland airport, public paranoia regarding such construction has never been higher. The action of environmentalists virtually precludes the building of mainland airports anywhere at this time. The location of new airports at great distances from the city, on farm land or the desert, besides facing injunction suits by ecologists, is made still more difficult by the expense of mass transit to and from the airport, which generally depends on voter approval of municipal bonds. For the suggested airport at Palmdale, 40 miles from Los Angeles, the cost of the required rapid transit system has been estimated from 600 to 900 million dollars. Nor is it likely that sufficient jet aircraft noise reduction can be effected by new aero engines, placement of the propulsion machinery above the wings to achieve shielding, the use of new fuels, etc. The driving powers involved in aircraft are simply too large and the acoustic-mechanic efficiency of the engines is too unfavorable to permit appreciable noise reduction by design changes.

The solution to the problem of aircraft noise abatement must lie in a combination of retrofitted engines, slightly steeper glide slopes and climb rates, and above all the utilization of distance as a buffer without encroaching on residential land and requiring huge new transportation systems for airport travel.

The following represents an acoustic analysis of offshore airports, and considers such factors as the variation of aircraft noise level with distance, the noise reduction achieved by a two-segment approach pattern, and the resulting NEF and CNEL contours, particularly as they pertain to the proposed offshore airport at San Pedro Bay, California.

Fig. 1 illustrates the four principal methods of constructing "wetports."

The top figure shows the so-called polder-and-dike system, developed by the Dutch, by which an impervious perimeter dam or causeway is built from the sea bottom up and to above the ocean, after which the water is pumped out of the basin so that the reclaimed land (the polder) can be used as a landing field. Generally, a breakwater is required for such a port, since it must be built in relatively shallow water.

The second diagram from the top represents the fill method of construction, by which a man-made island is formed, as by dumping dirt into the ocean and holding it in place by means of an encircling pyramidical barrier of quarry rock.

The third diagram from the top depicts a caisson-carried airport, whereby vertical watertight pilings are embedded in the ocean floor to support the landing field from twenty to sixty feet above the water.

The bottom diagram displays an airport held afloat by means of huge air-chambers below the water. Because such a floating island may be located at a great distance from shore, the noise level on the mainland due to aircraft flyovers is relatively small; however, the cost of construction and of maintaining such a structure is extreme.

Fig. 2 shows plan of the proposed airport at San Pedro Bay, made by the fill method, and estimated at 1.4 billion dollars.

EFFECT OF AIRCRAFT DISTANCE ON GROUND-MEASURED NOISE LEVEL

Numerous investigators in the field of aircraft noise propagation have reported that the change of noise level with distance of a plane with constant glide slope or constant climb rate does not follow the rule that the noise level is proportional to 20 $\log(D/d_0)$, 20 $\log(H/h_0)$, and 20 $\log(S/s_0)$, where D represents the distance from the airport, H the altitude of the plane, and S the slant distance to the plane, and where d_0, h_0, and s_0 are the corresponding reference distances, but that the change in noise level is greater than 6 dB for double the distance. This is undoubtedly due to temperature inversion with its sound-refracting phenomena, air turbulency caused by wind, temperature gradients with their sound-scattering effects, viscous and molecular absorption of sound by air, and other micrometeriological conditions. These atmospheric inhomogeneities are also responsible for temporal noise level fluctuations when the point of observation and the position of the outdoor sound source remain constant. It is therefore best to describe the change of noise level with distance for an approaching or departing plane in the form of $NL = NL_0 - f(Y) \log D$ where NL_0 = reference noise level at 1 mile from touch-down or lift-off; D = distance from airport, statute miles; f(Y) = function of distance, temperature, temperature gradient, barometric pressure, humidity, wind, engine design, etc. = 20 to 30

Thus, when f(Y) = 30, and the resident nearest to the aircraft is 7.8 miles from the point of touch-down on the runway, as in the case of the proposed offshore airport at San Pedro Bay, the noise level reduction comes to 30 log 7.8 = 26.8 dB, compared to the noise level at 1 mile. This is the noise level reduction directly under the flight path, at a given time, when $f(Y) = 30$.

Because the altitude of a jet aircraft landing with a 3 deg. glide slope is 277 ft at a distance of 1 mile from touchdown, it may be estimated that an elevated airport within the city, to give a noise level reduction of 27 dB to the nearest resident, would have to have an elevation of 7.8x277 = 2160 ft* - a very lofty enterprise which could well constitute an eyesore to many lower neighbors.

The variation of aircraft noise level with plane altitude, for the subject project, shall be taken as

$$NL = NL_0 - 30 \log \frac{H}{h_0} = NL_0 - 30 \log \frac{s}{h_0}$$

$$= NL_0 - 30 \log \frac{\sqrt{D^2 \tan^2 a + d^2}}{\tan a}$$

where s = slant distance to aircraft, statute miles
NL_0 = reference noise level at 1 statute mile from point of touchdown for a landing aircraft and 1 statute mile from point of lift-off for a departing aircraft
H = altitude of aircraft at point of observation, statute miles
h_0 = altitude at 1 statute mile from touchdown or 1 statute mile from lift-off
a = glide angle of approaching plane or ascent angle for departing plane
d = lateral distance from vertically projected flight path, statute miles
D = distance from point of touchdown or lift-off, under flight path, statute miles

Contours of equal noise level may be derived from the above equation; they are ellipses. When the glide slope of the landing aircraft is 3^o, and $\Delta DB = NL_0 - NL$, the equation is

$$\frac{D^2}{10^{.067\Delta dB}} + \frac{d^2}{0.00274 \times 10^{.067\Delta dB}} = 1$$

It can be demonstrated that when $a = 6^o$, the ellipses become twice as wide for the same abscissa value (distance from the airport) and the same delta dB (noise level reduction), because the tangent of small angles is linearly proportional to the angle.

Fig. 3 shows the noise level of a jetliner landing with a 3^o glide slope as a function of distance from the airport, points of observation directly below the flight path ($d = 0$), and for points laterally removed from the projected flight path. Note that at a distance of 1 mile from the airport, the noise level is 38 dB lower at a lateral distance of 1 mile than directly below the flight path. This explains why persons near an airport, but not under any flight path, are often far less annoyed by aircraft noise than more distant residents directly below landing or departing planes.

NOISE LEVEL ON THE GROUND AS A FUNCTION OF AIRCRAFT GLIDE SLOPE

It was previously calculated that at 7.8 miles from the point of touchdown on the runway the noise level of an aircraft landing with a

* The height of the Eiffel Tower in Paris is 986 ft; that of the Ostankino Tower, Greater Moscow, U.S.S.R. is 1749 feet.

constant glide slope was 26.8 dB lower than at 1 mile from touch-down. This is so for all glide slopes, because at 7.8 miles from touch-down the altitude of the plane will be 7.8 times the altitude at 1 mile, and it is thirty times the logarithm of the ratio of the two altitudes which determines the noise level reduction. Thus, when the glide slope angle is 3°, taken as a reference, and we assume the noise level of a landing Boeing 707 to be 107 dB-A at 1 mile from touch-down, then the reference noise level at 7.8 miles will be 107 - 26.8 = 80.2 dB-A.

It is now of interest to learn how this reference noise level of 80.2 dB-A changes with aircraft altitude, or what amounts to the same thing, how it changes with the glide slope angle. The table below gives the results of the pertinent calculation (see also Fig. 4):

Glide Slope	Noise Level Reduction*	Noise Level at 7.8 Miles**
3°	26.8 dB	80.2 dB-A
4°	30.5 dB	76.5 dB-A
5°	33.5 dB	73.5 dB-A
6°	35.8 dB	71.2 dB-A

*Noise level reduction at 7.8 miles from touch-down in reference to the noise level at 1 mile where plane altitude is 277 ft (altitude of plane approaching the field with a 3° glide slope).
**Noise level at 7.8 miles from touch-down when the noise level is 107 dB-A at 1 mile from touch-down for a landing plane with 3° glide slope.

It is seen that at 7.8 miles from touch-down the noise level reduction is in the order of 35 dB when the glide slope is 5.5°, compared to the noise level at 1 mile when the glide slope angle is 3°.

The noise level reduction with a change of glide slope at 1 mile laterally and at right angles to the flight path is not as much as it is directly under the path, because the relative change in distances to the aircraft is small. Thus, at 7.8 miles from touch-down and 1 mile laterally, the noise level of a plane landing with a 3° glide slope is 12.6 dB less than directly below the path, or 80.2 - 12.6 = 67.6 dB-A; at the same measurement position, the noise level of a plane landing with a 4° glide slope is 9.5 dB less than directly below the path, or 76.5 - 9.5 = 67.0 dB-A. Hence the difference in noise level at 7.8 miles from touch-down and 1 mile laterally is barely a half decibel when the glide slope increases from 3° to 4°, while at 7.8 miles directly below the path the difference in noise level comes to 80.2 - 76.5 = 3.7 dB.

In a two-segmental aircraft landing, where the plane approaches the field first with a 6° glide slope and then changes to a 3° glide slope, it is only the area under the first part of the approach pattern which experiences noise reduction, not the area under the second part, that is, the area closest to the airport, as shown on Fig. 5.

In Los Angeles, commercial airliners fly in four major approach (STAR) corridors. When a single-segment landing pattern is used, the aircraft assumes a 3 deg. glide slope at 5.5 miles from the field, at an altitude of close to 1800 ft. Constant stabilized descent to the west is at the rate of 800 ft/min under ILS (Instrument Landing System). When clear weather prevails, it is possible for some planes, such as PSA (Pacific Southwest Airlines), to employ a two-segment approach, whereby over the first half of the landing route the glide slope is 6 deg., after which the descent follows normal fashion. At 4 miles from the airport, the noise level reduction has been measured as 16 dB; 11 dB at 2.5 miles, and from 5 to 6 dB at 1.5 miles. Noise reduction is achieved not only by greater altitude, but also by reduced flap setting (25 deg. instead of 40 deg.) which allows a lower power setting for the engines. The approach is manual, because no instrument landing system exists for commercial aircraft operation employing the two-segmat approach.

In Fig. 6, the heavily marked upper range in every column shows the noise level and noise exposure level variation for landing commercial jetliners at 1 mile from the point of touch-down at the airport when the plane approaches the field with a constant glide slope of 3°. The heavily marked lower range in every column shows the noise level and noise exposure level variation for landing commercial jetliners at 7.8 miles from the point of touch-down when the plane approaches the field in a two-segment approach with reduced power setting and steeper flap position. As an example, the noise level at 1 mile from the field varies between 109 and 105 dB-A, while at 7.8 miles it varies from 74 to 70 dB-A.

In the figure, CNEL (Community Noise Equivalent Level) is calculated on the basis of 107 dB-A at 1 mile and 72 dB-A at 7.8 miles from the field, and 1000 daily flights, of which 700 arrive during the day, 230 during the evening, and 70 during the night, as forecast for LAX in 1975. On the assumption that all planes generate the same noise level, CNEL may be calculated by $CNEL = NL_o + 10 \log(N_d + 3N_e + 10N_n) - 38.4$ where N_d, N_e, and N_n are the number of daytime, evening, and night flights, and NL_o is the reference noise level at the point of observation.

In the same figure, NEF (Noise Exposure Forecast) is calculated on the basis of 120 PNdB at 1 mile and 85 PNdB at 7.8 miles, and 1000 daily flights, of which 700 arrive during the day and 300 nightly.

It is further expected that thn take-off disturbances of the very noisy SST's, if such will fly the skies in the near future, will not annoy residents on the California mainland.

CONCLUSION

A 10,000-acre offshore airport, 7.8 miles from California mainland, to which port airliners travel by stabilized two-segment landing pattern under ILS offers the nearest mainland residents a noise level reduction of at least 35 dB relative to the noise level at 1 mile from presently existing and conventionally operated mainland ports. Assuming that a noise level reduction of 10 dB corresponds to half the loudness of the signal, an elevenfold reduction in loudness can be achieved by such a littoral landing field. This corresponds to a 99.9% reduction of the power of the unwanted sound called aircraft noise. Maintenance cost reduction will be achieved by the exploitation of submarine oil deposits in San Pedro Bay and the remuneration obtained from a 4000-acre pleasure boat harbor and commercial docking facilities as part of the project.

Access to the island is to be by causeway, hydrofoil, monorail, and subterranean passage.

The estimated cost of the seadrome is 1.4 billion dollars. The airport, 25 ft above the ocean, will extend seaward from the 8.26 mile long federal breakwater in San Pedro Bay. The harbor berthing area, deep enough to accommodate supertankers, will include a liquid natural gas tank farm which will provide cheaper natural gas than that which can be pipelined to the island. Seawater conversion and waste disposal facilities will be part of the airdrome. Depth of the continental shelf for the proposed airport averages 75 ft. at the chosen site. An average cut of 37 ft. below the sea floor from the adjoining area will supply 1.6 billion cu. yds. of material for the 3.5x5 mile island. The four runways will each be 17,500 ft long, inclusive of a 2500 ft safety strip at each runway end. Fog conditions in the bay are less severe than at the present LAX location, and will be made non-critical by thermal and dispersal techniques. Should the aero engine manufacturers be able to lower the noise levels of their engines in the foreseeable future, additional noise abatement, particularly about the airport, will be reaped by their devices; but their achievements are not imperative for the immediate securement of quiet coastal areas about San Pedro Bay. The project will put 20,000 men to work within the first year of construction; employment will rise to 45,000 jobs in the second year, and to 56,000 jobs in the third and last year of the undertaking.

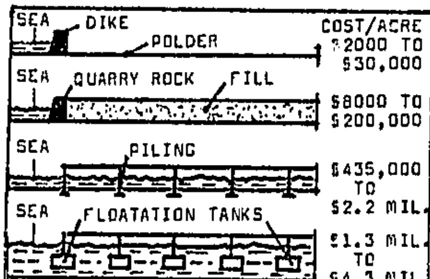

Fig.1 Offshore airport constructions

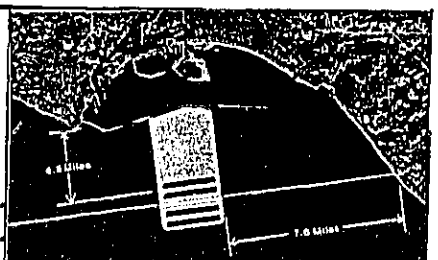
Fig.2 LOCATION OF PROPOSED OFFSHORE AIRPORT AT SAN PEDRO BAY, CALIFORNIA

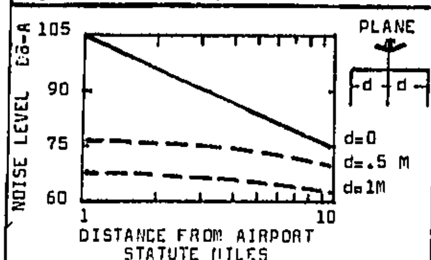

Fig.3 Change of noise level of landing plane with distance from airport

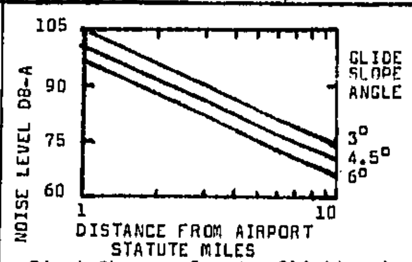

Fig.4 Change of under-flight noise level with glide slope of plane

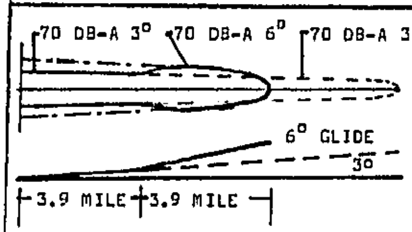

Fig.5 Noise level contour for two-segmental approach of jetliner

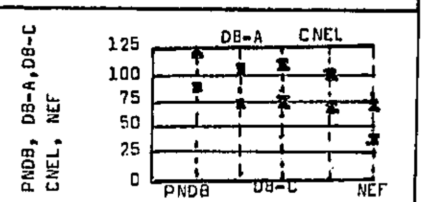

Fig.6 Noise level and noise exposure level range at 1 mile from airport (upper dark sections), and at 7.8 miles (lower dark sections)

ARLANDA AIRPORT, THE NOISE SITUATION NOW AND IN THE FUTURE

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When the Arlanda airport, 27 miles north of Stockholm, was opened in 1960 the surroundings were sparely populated. Since then an extensive urban development has taken place closer and closer to the airport. As late as last year a new densely populated area was decided upon to be built only 3 miles from the 01-19 runway. This will require certain restrictions, due to noise, on the departure routes for runway 19.

NOISE CRITERIA

The Swedish criteria for airport noise was presented in 1961 by a governmental investigation named "Flygbuller som samhällsproblem" (Aircraft noise as a social problem).

The criteria given in this investigation is called "critical noise limit", CNL, according to the following table,

No. of eq. flights/year	Crit. noise limit dB(A)
150- 500	95
500- 1500	90
1500- 5000	85
5000-15000	80
15000-50000	75

where the number of equivalent flights is calculated as the sum

$$N = 1 \times \text{day flights} + 3 \times \text{evening flights} + 10 \times \text{night flights}$$

This table has later been modified to a continuous function according to the principal of equal energy giving

$$\text{Const.} = \sum_i N_i \text{ antilog } \frac{dB(A)_i}{10} \qquad (1)$$

where i denotes different types of aircraft, N_i the number of equivalent flights and, $dB(A)_i$ the max. dB(A) value for the corresponding type of aircraft. The CNL is reached when

$$\text{Const.} = 2800 \text{ antilog } \frac{85}{10} \text{ per year or}$$

$$8 \text{ antilog } \frac{85}{10} \text{ per day (24 hours).}$$

In the investigations published 1961 the number of disturbed people was also studied and the result is shown in fig. 1.

FUTURE DEVELOPMENT AT ARLANDA

The airport authorities are planning a new parallel runway to 01-19, fig. 2a and 2b in order to increase the capacity of the airport. The surrounding communities have put very strong opposition against this. It has also resulted in a lot of public discussions and that the Regional Planning Office, which is the authority for the development of the Stockholm region, has conducted a survey of the airport problems. This survey has resulted in a proposition for a new airport to be built immediately north of the present one, with two parallel runways east-west, 09-27, fig. 2c.

The Aeronautical Research Institute of Sweden (FFA) has been the main Consultant regarding airport noise problems for the latter part of this survey.

AIRPORT NOISE STUDIES

At FFA a computer program has been developed for calculation of the critical noise limit and levels 5 dB(A) below and 5, 10 and 15 dB(A) above it.

Noise contours for different types of aircraft and flight profiles are calculated separately and stored on magnetic tape.

The number of flights for each type of aircraft and flight profile together with the appropriate flight tracks are fed into the computer and the accumulated noise immission according to equation 1 is calculated in a grid system consisting of 130 x 130 dots, 500 metres (1650 ft) apart giving a covered area of 65 x 65 kilometres (40 x 40 miles).

For each dot the number of people living in the surrounding area is also fed into the computer and the number of severely disturbed people is calculated according to the curve given in fig. 1.

Computer time for calculation of one case with 10 different types of aircraft and 24 different flight tracks is around 4-5 minutes using a CDC 6600 computer. For this study 200k core memory cells are needed.

The extension of the airport with a new N-S runway gives a total of 12010 severely disturbed people compared to 974 for the best E-W system, fig. 3 and 4. The new E-W runway system consists of two close parallel runways 09-27 and the flight tracks are chosen to give the least noise disturbancies.

However, since this study was done, more material on retrofit possibilities has come out. If a retrofit program can reduce the noise emission from DC-8 and DC-9 or similar powered aircraft with 10 dB(A), which seems to be a probable figure, the noise situation for the N-S runway system together with a slight decrease in the usage of the cross runway 08-24 will show a much better result with only 1213 severely disturbed people, fig. 5.

In my opinion a reasonable retrofit program can be a better solution to airport noise problems than building new runways further out from the cities. In the Arlanda case, the east-west runways means an almost new airport, as one of the existing runways will be used less than 5% and the other one will be completely closed.

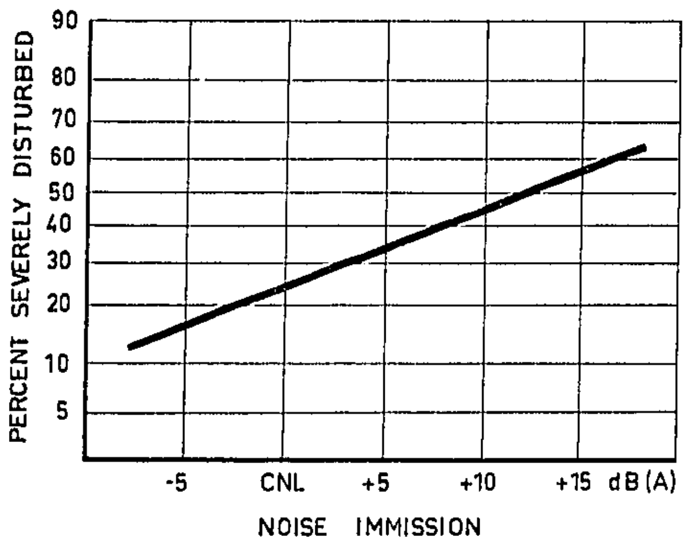

Fig. 1. Percent severely disturbed people vs. noise immission.

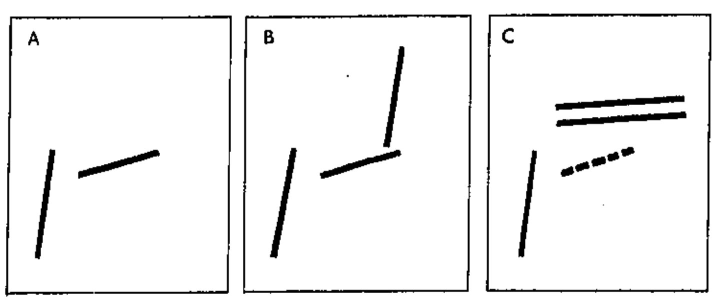

Fig. 2. Runway configuration.
A. existing.
B. extension proposed by Airport Authorities.
C. extension proposed by Regional Planning Office.

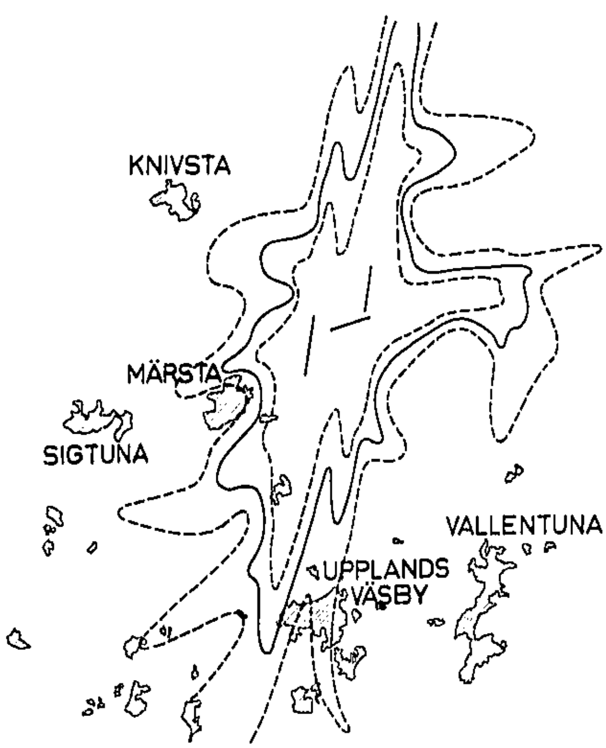

Fig. 3. Critical noise limit, CNL, and ±5 dB(A) contours for 1985 operations. New E-W runways.

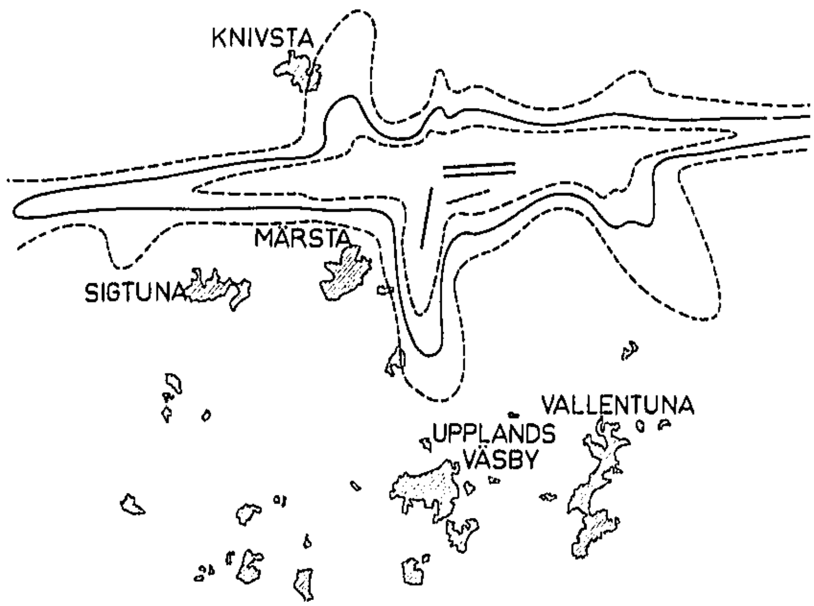

Fig. 4. Critical noise limit, CNL, and ±5 dB(A) contours for 1985 operations. New E-W runways.

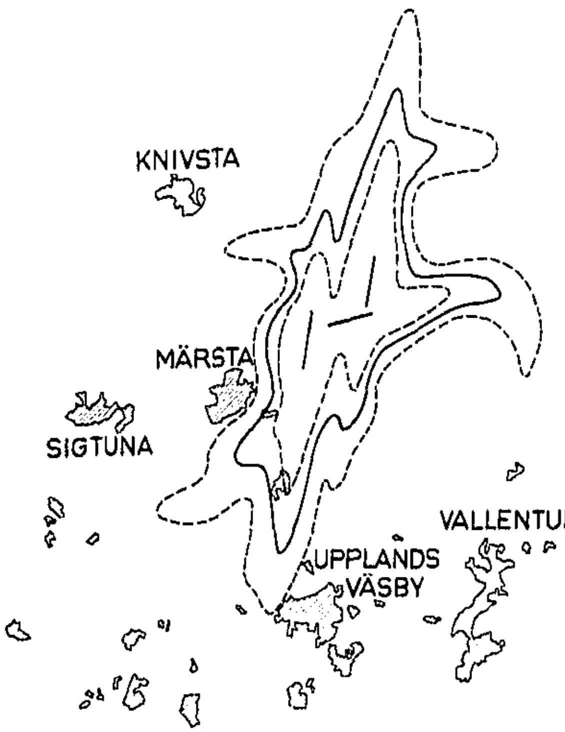

Fig. 5. Critical noise limit, CNL, and ±5 dB(A) contours for 1985 operations with retrofitted aircraft. New N-S runway.

PREDICTING THE REDUCTION IN NOISE EXPOSURE AROUND AIRPORTS

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INTRODUCTION

The use of airport noise models to identify areas of noise exposure around airports has become an accepted tool in many countries. These models generally consider the noise characteristics of the various types of aircraft operating at the airport, individual runway and flight path utilizations, day/night operations, and the total number of each aircraft type operating in some time period, usually a 24-hour day. The effect of altered flight procedures, changes in aircraft mix, or the impact of a new runway are usefully evaluated, for planning purposes at a specific airport, by exercising such models.

The present emphasis on means for reducing airport noise exposure has brought forth many proposals for changes in air carrier fleet composition as well as for uniform adoption of noise abatement flight procedures. To assess the impact of these proposals, if applied fleetwide to the 2000 jet aircraft U.S. system at all airports served by air carrier operations, would involve analyzing the many aircraft alternatives at more than 500 airports. It would be highly useful to have a simplified model that would allow planning analyses of alternate fleet configurations to be made without detailed knowledge of operations at each airport. We describe an approach to such a model.

GENERALIZED AIRPORT MODEL

The first step in the analysis consisted of computing NEF contours for 9 airports, ranging in size from Indianapolis (≈300 operations per day) to Chicago-O'Hare (≈2000 operations per day), for the forecast operations of 1985, if the attrition of older aircraft and introduction of current design aircraft were allowed to take place in a normal growth pattern. The areas enclosed by NEF contours from 30 to 45 were then plotted on a semi-logarithmic basis as shown in Figure 1. The striking similarity in the slopes of area as a function of NEF value, irrespective of the particular runway layouts and flight path utilizations at the individual airports, suggested that a simple model might represent NEF area for many airports, the primary variation between airports being number of operations.

Many airport noise models, including NEF, assume exposure to vary as $10 \log_{10}$ of the number of operations. The empirical slope of area with NEF (at least from NEF 30 to 40) shown in Figure 1 suggested that a $15 \log_{10}$ (area) relationship might be assumed. Using these assumptions a forced least squares fit of area and NEF produced the following relationship:

$$\text{Area (sq. mi.)} = A_o \cdot 10^{\left[\frac{10 \log N + 24 - NEF}{15}\right]}$$

where N is daily operations and A_o an arbitrary constant. Comparing this equation with the actual areas for the 9 airports, assuming A_o equal to

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- P.24-28 AIRCRAFT NOISE MODELS

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unity, overpredicted NEF 30 by 15%, with a standard deviation of 21%, and NEF 40 by 2%, with a standard deviation of 18%. The value of A_{30} for NEF 30 is thus chosen as 0.86 and that for NEF 40, A_{40}, as 0.98.

Closer examination of the differences between airports revealed that the most deviant predictions of actual area from the general model occurred at airports having substantial deviations from the average in three quantities: 1) ratio of day/night operations, 2) percent of 4-engine low bypass fan aircraft (707, DC-8), 3) proportions of short range to long range operations. Empirical corrections were derived for these factors, resulting in the following improved model:

$$A = A_1 \text{ (NEF) } 10^{\left[\frac{10 \log N + 24 - NEF + C}{15}\right]}$$

Where A = area in square miles

A_1 (NEF) = 1.01 for NEF 30 for fleet 1

= 0.94 for NEF 40 for fleet 1

N = number of operations in 24 hours

NEF = NEF value

$$C = C_1 + C_2 + C_3$$

$$C_1 = 10 \log \frac{N_d + 16.7 N_n}{N_d + N_n} - 4.2$$

N_d = number of operations between 0700 - 2200

N_n = number of operations between 2200 - 0700

C_2 = 4-engine LBPF correction

= -2 if 4-engine LBPF <5% of operations

= -1 if 4-engine LBPF are between 5% and less than 10% of operations

= 0 if 4-engine LBPF are between 10% and less than 15% of operations

= 0.5 if 4-engine LBPF are equal to or greater than 15% of operations

C_3 = short range correction

= 1.0 if 80% of aircraft have range of $\leq$ 1000 miles

= -0.5 if 50% of aircraft have range of $\leq$ 1000 miles

This model predicts the area of NEF 30 with a standard deviation of 3% and the NEF 40 area with a standard deviation of 6%.

SPECIALIZED MODELS

The second part of the problem is to find a mechanism for evaluating the effect of operating procedure changes (e.g. noise abatement takeoffs, two-segment approaches) and aircraft fleet configuration changes. The relative effect of many possible combinations can be evaluated simply by assuming a single runway airport having straight-in and straight-out flight paths. One can define a series of "base case" conditions as that made up of various mixes of current aircraft, the mixes being typical of those found at short range, long range domestic, international, etc., airports.

The relative effect of procedures, engine changes, new aircraft, or any other set of conditions, can be calculated by determining the ratios of specific NEF areas of the particular assumed case to the appropriate NEF areas for the base cases. These area ratios may then be used as multipliers to the appropriate A constants in the preceding area equations to estimate the area change at specific airports. We have used this approach to evaluate a large number of alternate fleet assumptions. When applied to specific airports where the full NEF analysis has also been performed, the simplified model approach predicts the areas for NEF 30 and 40 contours within a few percent of the areas determined by the detailed analysis.

APPLICATION TO A NATIONAL SYSTEM

As an example of the use of these techniques in a "global" planning problem we examined the simple model for 1985 applied to all U.S. air carrier airports having more than 10 flights a day. It was assumed that operations at each airport would be from a fleet having aircraft types proportioned to the relative population of that type in the total U.S. air carrier fleet inventory. Area coefficients were derived for different 1985 fleet mixes, operational procedures, and for the 1972 fleet as presently constituted. The gross assessment of these effects is provided in Tables I and II.

CONCLUSION

A model is suggested for evaluating the gross effects of various aircraft fleet operations on airport noise exposure. The model is primarily directed at assessing policy implications and not for detailed planning at specific airports. The use of area coefficients to examine alternate fleet configuration effects at specific airports may be used for individual airport planning, however, if an initial detailed analysis is made for a "base case" condition at that airport.

TABLE I

AREA ENCOMPASSED BY NEF 30
U.S. AIR CARRIER AIRPORTS HAVING
10 OR MORE FLIGHTS PER DAY

Number of Annual Operations (in 1,000's)	Number of Airports 1972	Number of Airports 1985	NEF 30 Area Sq.Mi.	Total NEF 30 Area-Sq.Mi. 1972	Total NEF 30 Area-Sq.Mi. 1985	Assumed Airport Area Sq. Mi.	Total Airport Area-Sq.Mi. 1972	Total Airport Area-Sq.Mi. 1985
0-50	148	206	5.2	770	1070	2	296	412
51-100	22	35	10.8	238	378	2	44	70
101-150	10	11	15.4	154	169	3	30	33
151-200	6	8	19.3	116	154	3	18	24
201-250	2	4	22.6	45	91	4	8	16
251-300	3	3	25.8	77	77	4	12	12
301-350	2	2	29.2	58	58	5	10	10
351-400	1	2	31.9	32	64	5	5	10
401-450	1	2	35.0	35	70	7	7	14
451-500	0	1	37.7		38	9	9	9
501-550	0	1	40.0		40	10		10
551-600	1	0	42.5	43		10	10	
> 600		1	43.0		43	10		10
Total	196	276		1568	2252		440	630

1972 area:

Effect of fleet composition and procedures shows 1985 to be 0.43 of 1972 area. Thus $\frac{1568}{0.43}$ = 3650 sq. mi. total NEF 30 area. Area outside airports is 3650 - 440 = 3210 sq. mi.

TABLE II

EFFECT OF VARIOUS FLEET COMPOSITIONS
ON NEF 30 AREAS

1. Area coefficients derived from single runway tradeoff analysis relative to 1985 base case.

Case A	-	Nacelle retrofit	0.55
B	-	Replace 707/DC-8 with 747, DC-10, L-1011, retrofit nacelle on 727, 737, DC-9	0.47
C	-	Retrofit with new front fans	0.47
D	-	Replace all earlier aircraft with 747, DC-10, L-1011 and new 2 engine	0.27
E	-	Re-engine with "quiet" engine	0.05

2. 1985 area - total NEF 30

Case A	-	2252 (0.55) = 1240 sq. mi.
B	-	2252 (0.47) = 1060 sq. mi.
C	-	2252 (0.47) = 1060 sq. mi.
D	-	2252 (0.27) = 610 sq. mi.
E	-	2252 (0.05) = 113 sq. mi.

Note: 1985 airport area to be subtracted is 630 sq. mi.

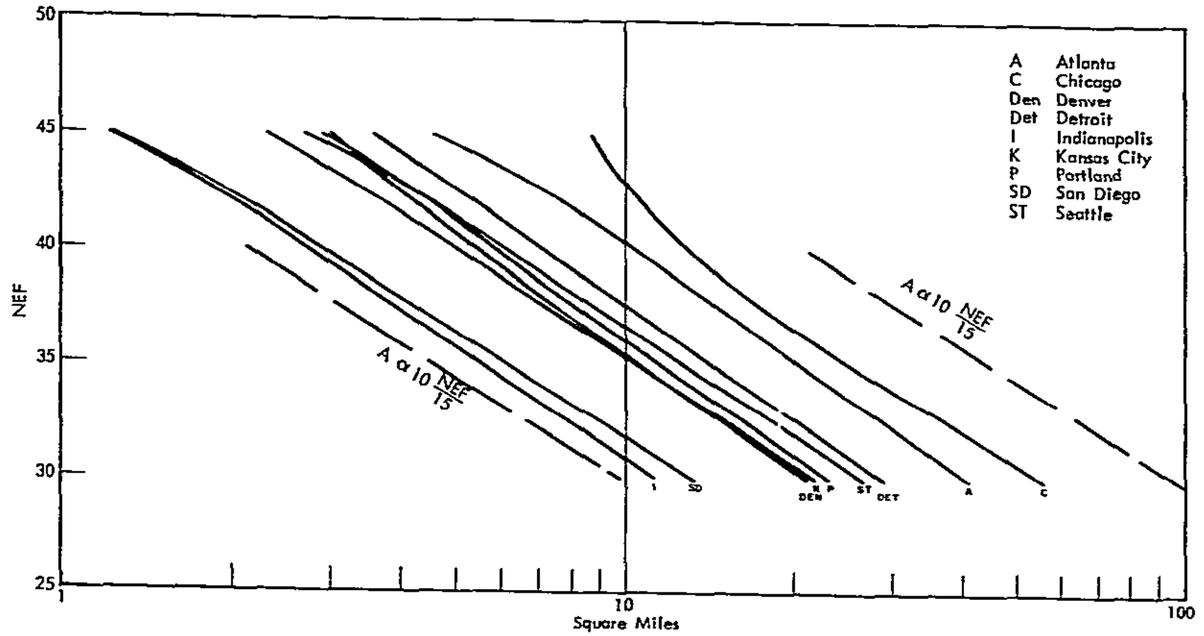

FIGURE 1. VARIATION IN TOTAL AREA WITH NEF - 1985 BASE CASE

AIRCRAFT NOISE DISRUPTION IN PUBLIC SCHOOLS: A DEFINITION OF AN IMPASSE

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Introduction

There appears to be two classifications of noise emitters: noise emitted by aircraft and noise emitted by other sources. While the latter problem is approached directly, the former becomes a complex of evolving technologies, governmental jurisdictions and airline economics that approach Kafkaesque fantasy. However, school systems cannot classify disruptive noises by emissions and we must explore this privileged status.

The problem is nation-wide and I have been consulted by a school district as remote from New York City as the Highline Public School District surrounding the Seattle-Tacoma International Airport. I hope the Kafkaesque analogy is not strictly accurate. I hope this is not a problem without a solution: a problem that must be lived with or at best contained, to use a Federal euphonism. My experience with the problem leads me to believe there is a sensible approach, an approach that would undeniably ruffle some feathers, but may well be the only practicable consideration of the problem.

The Problem

The protagonists are a heavily trafficked airport runway, being JFK International Airport Runway 13L and a heavily utilized school at the end of the runway, being Public School No. 124 in the Borough of Queens. The guardians are The Port of New York Authority with the Federal Aviation Administration responsible for ground and airborne aircraft respectively and the Board of Education with the Community School District responsible for the proper operation of the school. There is agreement on the problem: site instrumentation as well as extrapolation from many similar studies produce noise levels from 80 to 100 dBA. There are known remedies: by the school through closure of the building's curtain walls and by the runway through alteration of its plane movements. There are also expenses: $800 thousand for the building's closure and a substantially greater amount for appropriate plane movement alterations. Who is responsible?

Governmental Jurisdictions

The City of New York owns the airport and has leased it to The Port of New York Authority as tenant for fifty years (1) in consideration of a yearly rental of about $450 thousand. One could assume the landlord would have some control over its tenant in this matter, particularly when the lease specifies a capacity of one thousand plane movements per day (2), a figure that is exceeded daily. Not so. The City's Chief Legal Officer, its Corporation Counsel, not only interprets "capacity" as "minimum", but opinions operational authority to existing governmental regulations (3).

Let us consider an action by the Federal Aviation Administration: although trivial in itself, it may well be typical. The FAA applied for and received from the Board of Education permission for the installation of a so-called Compass Locator Facility on Grover Cleveland High School in the Borough of Queens (4), the school being on a line with the noted Runway 13L. I questioned the feasibility of defining an airport approach corridor directly over a high school with its attendant noise from low flying aircraft. The FAA by letter to the Board of Education then noted that they were reconsidering the location of the facility. There has been no further action.

There is the other side of the airport, adjacent to the Town of Hemstead near New York City. This Town's concern with the problem resulted in their pursuit of the FAA up to the United States Supreme Court (5). The pursuit lacked jurisdiction and the Congress amended the Federal Aviation Act by placing control and abatement of aircraft noise with the FAA (6). Like placing a fox in charge of a hen-house, ran a comment by an official of the Town. No doubt the comment was colored by the moment, but it is not without aptness.

Having an engineer's logic, I find the existence of the inconsistency startling. Effective aircraft noise abatement necessarily restricts the optimum performance of an airport. It should be evident that investing an agency with two conflicting premises encourages one and demeans the other. There should be no surprise in finding the demeaned premise couched on a tiresome bag of public relations gimmicks to maintain the status quo.

The essence of regulation is the lack of bias. That effective noise abatement should be practicable is evident: the outstanding example is the Federal Labor Department's Occupational Safety and Health Administration. I could muse on the extension of the Walsh-Healey criteria from Federal contractual employees to public school children, but that may be stretching a point. The Congress has stepped in the right direction by considering authorization to the Environmental Protection Agency to review and report on FAA regulations (7). Similar steps are needed until the aircraft noise abatement function is totally external to the Federal Aviation Administration.

<u>The Engineering Function</u>

The Institute sponsoring this Conference has defined their function laudatorily as applying noise control technology to the benefit of mankind (8). The definition covers a lot of ground and the pertinency of governmental jurisdictions and associate legislation could be explored. Engineers are prone to consider such matters a world apart to the point of deliberate exclusion. Yet jurisdictions and legislation must reflect practicable performance to be effective. I recall the classic example of impracticable legislation: the so-called Prohibition Amendment to the U.S. Constitution. The law may have prohibitions, definite and explicit, but these prohibitions would be useless when compliance is not possible in practicable and economically feasible ways.

A curious approach to the problem of Public School No. 124 with JFK International Airport was considered: amendment to The State of New York Education Law to define quantitatively tolerable noise levels in school classrooms. It was my considered opinion that the result not only would be nil, but would include contempt for the law. Neither the school nor the airport is going to move and the law will fix responsibility only when it defines practicable and economically feasible ways for compliance. It is the engineers' job to define these ways for compliance and the engineers have not done so.

Noise reduction is becoming a highly sophisticated technical discipline. The recent increase in acoustical consultants, in instrumentation manufacture, in the use of noise attenuating material and the like is explosive. Understandably, the engineers must live, particularly with the Federal defense expenditures as low as they are. But what is the point in selling a town or a school district a vast array of acoustical instrumentation, providing weighted technical evaluations and attenuating demonstrations to prove what has already been proven? Why must we reinvent the wheel? Why do we instigate enormously expensive public litigations only to have the matters thrown out of court for lack of jurisdiction?

I look forward to the performance by the members of the Institute of their laudatory function. However, we must also concern ourselves with the present.

There may be those here who consider the above function somewhat altruistic: the so-called "hard-headed, what's-in-it-for-us" businessmen. They too are against sin, but they have no time to sit contemplating their navels. There are payrolls to meet and mortgages to pay. Therefore, for the moment, let us neglect the compassionate needs of public school children and analyze the market potential.

The Market Potential

Consider the cost factors within New York City. There are at least forty primary and secondary schools within the Borough of Queens alone concerned with the problem of airports within its boundary. Let us add ten more from the remaining four Boroughs. The cost for a primary school was noted as $800 thousand and we may round this off at $1 million for the average of both primary and secondary schools. Since the public school system within New York City comprises about one thousand schools, our cost factors are $1 million per school for one-twentieth of the total number of schools.

These cost factors are very conservative. They are based on the present and do not include the increased noise levels and school populations to be expected. They do not consider the one-third addition of parochial and private schools. They neglect the higher education school system entirely, both public and private. All this is noted to confirm that these cost factors can be no lower than stated.

The problem is not confined to urban areas, if for no other reason than remotely located feeder airports. The Stewart Air Force Base, about sixty miles from New York City near Newburgh, comes to mind. The Metropolitan Transit Authority is planning conversion and expansion there for an air freight terminal to serve New York City. The extension of the noted cost factors to non-urban areas should not be disputed.

The approach to the problem at the Stewart AFB struck a familiar chord, if I may digress a moment, when the opposition to the MTA plan sought my advice. Their approach illustrates the paucity of legal definitions of the problem: initiating a series of Federal Court actions as successive stumbling blocks to produce delay. It was curious that the opposition employed the same group that similarly delayed the Consolidated Edison Company from their Storm King Project to the point of abandonment.

The nation-wide public school system may be documented (9) as follows:

Public Urban Elementary Schools	22,500
Public Non-Urban Elementary Schools	45,500
Public Junior High Schools	14,500
Public Urban High Schools	3,500
Public Non-Urban High Schools	12,000
Total Public Schools	98,000
Total Public School Districts	9,000

Using the noted cost factors, the market potential may be stated as $5 billion. According to your present approach, the best that could be expected from the public school sector is the installation of a sound level evaluation laboratory costing about $50 thousand in each concerned public school district, say one-twentieth of the total number of districts. Thus, we compare a very conservative $5 billion to a very extravagant $23 million. In this case, an altruistic approach to the problem does not mean poor business.

Summary
When the facts are known, they cannot be ignored. That present conditions produce an impasse to the solution of the problem may only be attributed to ignorance. We do not need public relations soothsayers to tell us that school children really do not mind aircraft noise or that the Federal Aviation Administration shall do everything possible to abate aircraft noise. We simply need realists.

There lies the impasse in present conditions.
Two changes are needed:

. . . vesting the Federal Environmental Protection Agency with all aircraft and airport noise abatement functions and

. . . providing practicable and eco nomically feasible technical means to complement aircraft and airport noise abatement legislation realistically.

Continuing the present conditions into the future can only continue the impasse.

FOOTNOTES

1. The City of New York and The Port of New York Authority "Agreement with Respect to Municipal Air Terminals" dated April 17, 1947 with eight supplements up to December 12, 1968. Sections 3. and 4.

2. Ibid, Section 9. (d).

3. The City of New York Law Department. Opinion No. 107,523 by J. Lee Rankin, Corporation Counsel, dated July 21, 1971.

4. Board of Education Calendar No. 49 of September 30, 1968. FAA Eastern Region Drawings NYC-D-0693, three sheets "New York, N.Y. Metropolitan Navigational Aids for Air Traffic Improvement".

5. 272-Federal Supplement 266 of June 30, 1967
398-Federal Supplement 369 of July 17, 1968
37-US Law Week 3274 of January 13, 1969
"US Supreme Court denied the Town of Hemstead Petition for a Writ of Certiorari".

6. Public Law 90-411, 90th Congress, H.R. 3400, July 21, 1968

7. House Report 92-892 by the 1972 Sub-Committee on Air and Water Pollution, H.R. 11021.

8. INCE Preamble to its By-Laws.

9. Patterson's American Education, 1971 Bulletin No. 24 Educational Directories, Inc., Mount Prospect, Illinois

NOISE IN AN AIRPORT COMMUNITY

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INTRODUCTION

The City of Inglewood lies directly under the approach paths to the four runways at Los Angeles International Airport. Noise from landing aircraft has affected the city to the point where Inglewood has instituted a full-time noise abatement program including technical staff and monitoring equipment. This paper will report briefly on five programs of current interest.

SAN DIEGO APPROACH ALTITUDES

Using our mobile monitoring laboratory, we measured the approach angles of aircraft landing at San Diego International Airport and compared them to the approach angles of aircraft landing at Los Angeles. The San Diego Airport was chosen for monitoring because high terrain causes aircraft to fly steeper than normal approaches.

The mobile monitoring unit is shown in Figure 1. The system includes a microphone, sound level meter, graphic recorder, tape recorder, aircraft band radio, and camera. The camera was the primary instrument used for measuring aircraft altitude: by knowing the dimensions of each aircraft the altitude could be directly determined from image size. The accuracy of this technique is approximately 1%.

A total of 88 flyovers were monitored at San Diego on three successive days during March, 1972. These were compared to a total of 506 flyovers monitored at Los Angeles during February, 1971. At each airport the study included 2, 3, and 4 engine jet transports. At each airport the measurement site was approximately three miles from the touchdown point on the runway.

The average approach altitude at San Diego was 1375 feet. The average altitude at Los Angeles was 864 feet. These altitudes correspond to approach angles of 5^{o} and 3.1^{o} respectively. These steep approaches were made at San Diego in spite of the fact that the runway length available for landing is only 7,590 feet compared to 11,395 feet at Los Angeles. The number of flyovers in each sample was large enough that the calculated difference in average approach altitudes was statistically significant at the 99% confidence level.

If 5^{o} approaches were flown at Los Angeles, aircraft noise would be reduced approximately 12 EPNdB, or 56%!

Figure 2 shows the basic results of this monitoring.

In April, 1972, the Federal Aviation Administration installed a $4\frac{1}{2}^{o}$ VASI (Visual Approach Slope Indicator) at San Diego. This system now provides visual guidance to pilots making steep approaches there. A $4\frac{1}{2}^{o}$ VASI system, if installed and used at Los Angeles, would reduce aircraft noise approximately 9 EPNdB, or 46%.

NOISE EXPOSURE IN INGLEWOOD

In addition to its mobile monitoring laboratory, the City of Inglewood operates a network of fixed monitoring stations. A map showing the locations of the four stations is shown as Figure 3. Figure 4 shows a typical microphone installation on top of a telephone pole. Figure 5 shows the central station, which is connected to the microphones by broadcast quality telephone lines.

The basic data recorded by this system are the total number of minutes that noise exceeds preset levels (usually 80, 90, and 100 dBA). Our current monitoring practice is to record these data weekly. The result, then, is week-by-week information on the total noise exposure to four locations in Inglewood. From this we can determine long term trends in noise exposure.

The data reported here were obtained during the period February through November, 1971. No significant trends were observed at stations 2, 3, or 4. Station 1 exhibited a significant trend toward reduced noise as shown in Figure 6. The data are reported in units of "exposure time level," defined by

$$ETL = 10 \log_{10} \frac{T}{T_o}$$

where T = exposure time per week above a given noise level

T_o = 0.1 minutes per week (smallest measurable time increment)

Over the measurement period, the arithmetic average of the exposure time levels above 80, 90, and 100 dBA at station 1 decreased approximately 6.5 dB. This was a statistically significant trend. Although the true cause of this decrease can not be known with certainty, we attribute the trend to the increasing acceptance and use of steep approaches by various pilots and airlines.

COMMUNITY NOISE MONITORING

Between August, 1971 and April, 1972 we used our mobile monitoring unit to take 24-hour samples of noise at 35 different locations in Inglewood. Of primary interest were ambient noise levels, peak noise levels, and Community Noise Equivalent Levels (CNEL).

Ambient noise levels varied from nighttime lows of 34-53 dBA to daytime highs of 45-60 dBA. The average day-night spread in ambient noise levels at a given site was 9.5 dBA, with a range of 3-15 dBA. Proximity of major streets seemed to be the dominant factor in determining ambient noise levels.

The highest noise peak ranged from 70 to 105 (depending on location) dBA at night and from 80 to 104 dBA in the daytime. The lower values occurred in quiet residential areas away from the flight patterns. The highest peaks occurred immediately under the flight paths. The number of peaks within 10 dBA of the highest peak ranged from less than one per hour far from the flight paths to 19 per hour in the evening directly under the southern flight path.

CNEL values ranged from 54 to 86 dB, dominated almost entirely by proximity to the flight patterns. A strip approximately 0.3 miles wide had CNEL values above 80 dB, and a strip approximately 2.5 miles wide had CNEL values above 65 dB. Figure 7 shows these CNEL trends schematically.

PROPERTY VALUES IN INGLEWOOD

The City of Inglewood conducted a study to see if there was any correlation between aircraft noise levels and residential land values and dwelling vacancy rates in Inglewood. The results were computed by a linear regression analysis based on Census Data and Assessor's Data.

The results showed that high noise levels were correlated with low land values and with high vacancy rates. On the average, residential land values were approximately 50% higher in areas where aircraft noise is less than 80 PNdB compared to areas where aircraft noise exceeds 110 PNdB. Similarly, rental dwelling vacancy rates are 50% higher where aircraft noise exceeds 110 PNdB as compared to areas where this noise is less than 80 PNdB. Figures 8 and 9 show these trends schematically.

SOUNDPROOFING ORDINANCE

Most complaints about aircraft noise relate to the disruption of indoor activities. Therefore the City of Inglewood is considering adoption of an ordinance that would require acoustical treatment of new residences under or near the flight paths.

The details of the ordinance were developed by Wyle Laboratories as part of Inglewood's Community Review Program. Two soundproofing zones are designated, one requiring an external to internal noise reduction of 28 dBA (Zone A), the other requiring 35 dBA (Zone B). The requirement of Zone A can be met with normal construction techniques if an air circulation system is installed allowing doors and windows to remain closed. The 35 dBA criteria of Zone B requires acoustical doors and windows (STC 36 and 39) in addition to an air circulation system.

The City asked various door, window, and air conditioner suppliers to quote on the costs involved in complying with the ordinance. The results were that construction costs would be increased by 3% and 10% ($0.35 and $1.45 per square foot of floor area) for Zones A and B, respectively. A rent differential of $3.50 and $14.50 per month in Zones A and B would be sufficient to bring a 15% rate of return on the investment in acoustical treatment.

CONCLUSIONS

The results of these and other studies in the City of Inglewood continue to point up two major conclusions:

1. Aircraft noise pollution is a matter of grave concern to the community.

2. Methods are readily available to significantly reduce aircraft noise pollution.

Figure 1. Inglewood's Mobile Noise Monitoring Laboratory

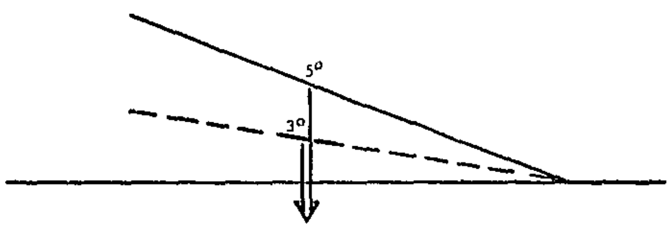

Figure 2. San Diego 5° Approach

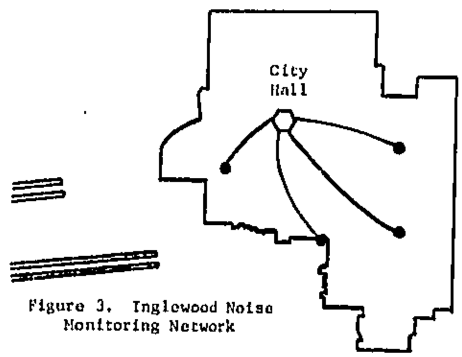

Figure 3. Inglewood Noise Monitoring Network

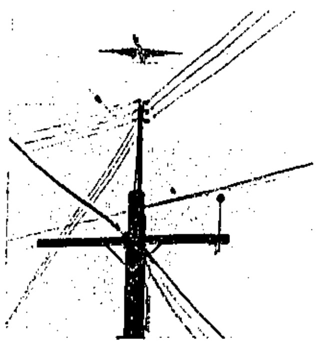

Figure 4. Remote Microphone Installation

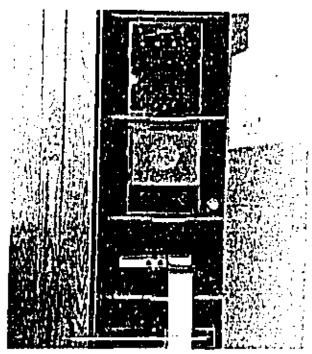

Figure 5. Central Recording Station

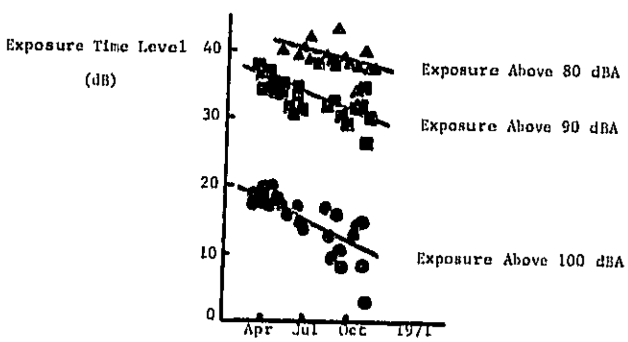

Figure 6. Noise Exposure Trends at Station 1

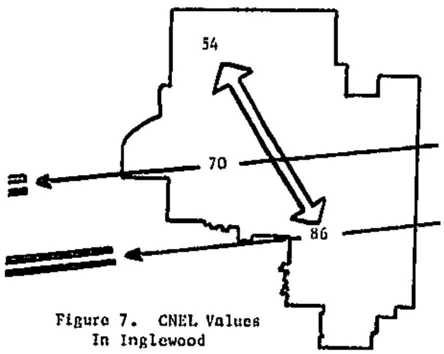

Figure 7. CNEL Values
In Inglewood

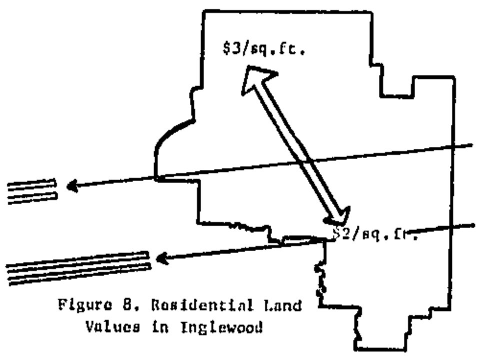

Figure 8. Residential Land
Values in Inglewood

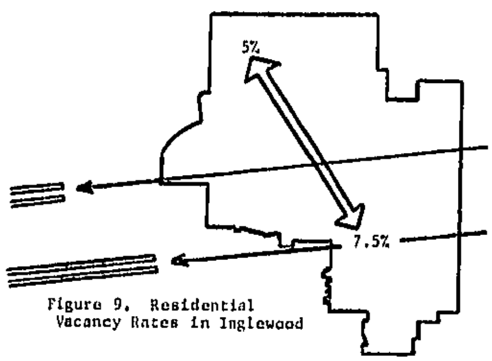

Figure 9. Residential
Vacancy Rates in Inglewood

NOISE INSTRUMENTATION AND MEASUREMENT

THE USE OF A REFERENCE SOUND SOURCE IN STUDYING INDUSTRIAL NOISE

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1 - INTRODUCTION

There is nothing novel in seeking to use a reference sound source as intermediary to establish, in a sound field perturbed by reflections, the acoustic characteristics of machines or devices, in particular their power level. First mention of such work is found in technical papers in the 1960'S with the publications of H.C. HARDY [1] and R.J. WELLS with F.M. WIENER [2]. At about the same time, the Laboratory of Acoustics of ELECTRICITE DE FRANCE was already using such a reference source for rapid control of its microphones.

What is novel, however, today, with the systemization of efforts undertaken to protect the environment, is the need to multiply acoustic determinations of an industrial nature, i.e. using expert knowledge and control "in-situ". To reach this objective, simplified procedures must be defined permitting not only to save time compared with standard laboratory procedures but also to operate under any circumstances, especially in the presence of parasite reflections more or less well defined and even more or less determinable. It was, moreover, with this idea of simplifying measurement methods that the experts of ISO/TC.43/SCI/WG6 introduced in the first basic documents now being examined by national committees, the possibility of using a reference sound source for determinations in reverberation rooms.

Alerted by the work of ISO in which they took an active part and faced with national needs increased by the birth of the French "Ministère de l'Environnement", the authors of the present publication have endeavored to make a personal contribution to solving the practical problems posed by acoustic measurements "in-situ". They have first proceeded with the development of a new type of reference sound source whose characteristics - more especially those relating to radiation directivity and the spectral distribution of energy - make it possible to expand the field of use they were limited to with pre-existing sources. They then carried out systematic experimentation of this source to establish, by methods of substitution and comparison, the acoustic characteristics of machines placed inside and outside buildings. They also verified the possibilities of extending use of this equipment to building acoustics in order to determine the absorption characteristics of the rooms and the insulation of partitions.

2 - FOR AN ALL-PURPOSE REFERENCE SOUND SOURCE

2.1 - Origin of the idea of the source presented ; description

The many uses planned for this reference source require it to produce a sufficiently high level of sound in the measurement space, that the distribution of acoustic energy be as even as possible in the frequency range involved, that the general line of the spectrum be within the limits formed by the spectrum of white noise (increase of 3 dB/octave going towards the high frequencies) and that of pink noise (horizontal spectrum), and lastly that the sound be of the wide-band type and that the radiation be practically omnidirectional.

The authors first systematically tested the two sources that had already long been available to them for special laboratory tests and each of which consists essentially of an electric motor with horizontal shaft at the extremities of which are fitted one or several fan turbines, one for the source ILG, four for the source EDF (CEM), (see figure 1).

It was then noted [5] on these two sources that a certain number of characteristics were not suitable for the planned general use, especially the presence of pure sounds in a quantity varying with the height of the source above the ground, resulting from the position of the reflecting plane at the side of and near the turbines, thus causing them to ...

operate unsymmetrically and creating a marked directivity in the horizontal plane, the usual one for operation. To remedy these two defects, we sought to merge as for as possible the acoustic center with the reflecting plane, to ensure a regular feed of the ventilator wheel used to create the sound, and to send upwards that part of the sound radiation likely to show a directivity index exceeding 3 dB.

For this purpose, the electric motor drive is on a vertical shaft and the turbine of the type used in air conditioning appliances, entirely covers the motor (see fig. 2). A perforated grid surrounds the turbine as a safety measure, and its characteristics were chosen to raise the spectrum in the high frequencies. The empty spaces left free between the base and the turbine in order to feed the latter, as well as between the grid and the base were subjected to a few summary tests for radiation optimization.

2.2 - Characteristics of the new source

The characteristics of the various models of this reference source were taken in anechoic room by continuous exploration on a series of 10 horizontal circles spaces at regular intervals and located at 1 m from the source. Paper [5] gives further indications on the calibration measurements. The acoustic power levels in third octave band are given in fig. 3 from 100 to 10.000 Hz compared with those of two other sources initially studied. The directivity characteristics are already adequate, at the present of development of the source, for its general use even outdoors. At the upper part alone, in fact, in a solid angle of 30° in relation to the vertical axis, we have a directivity index exceeding 3 dB above 3.000 Hz, this index remaining, moreover, below 2 dB whatever the direction for the overall values in dB (A) or in dB (C) as well as in practically all other directions and other third octave bands (see fig. 4).

Furthermore, various tests were made on the first model to make certain that we had effectively a wide-band sound in the frequency range chosen and that the influence of power supply voltage and frequency variations were negligible [6]. The change in slope of the spectrum thus obtained between the first model of this source and the second, used for the experimentation referred to further on, was obtained by reducing the space allowed between the grid and the wheel of the fan. Action on this distance and on the diameter and distribution grid perforations permits the shape of the spectrum to be regulated within the limits indicated at the beginning of this section.

3 - EVALUATION OF THE ACOUSTIC POWER OF MACHINES IN AN ENVIRONMENT PERTURBED BY REFLECTIONS

3.1 - Nature of the problem

The method to be used for implementing the reference sound source appears to depend essentially, at first sight at least, on the removability of the source to be tested and the directivity of the field of reflections. In actual practice the real problem lies in choosing positions for the reference source in relation to those of the source tested.

When the source is removable, a substitution of the unknown source by the reference source is effected. On the other hand, when the source is not removable, one has to proceed by comparison : the reference source is given a location equivalent to that of the machine to be tested, or by juxtaposition : the reference source is placed at best against or on the machine this according as the field of feflection is diffuse or, on the contrary, has directions of greater influence. The latter is the case nearly always met with when one is outside a building and when the parasite reflections are caused by a clearly located obstacle in the surrounding area.

3.2 - The substitution method

In the most complex case, we meet with a directive field of reflection - whether we are indoor or outdoor - and a source that can be considered as practically punctual has to be substituted for an extensive source which, moreover, is very often not placed on the ground. In such a situation, according to the diagram of fig. 5, four kinds of positions are possible for the reference source. The latter can, in fact be placed either in the center, or at the periphery of the outside enveloppe of the machine and in either of two parallel planes, one merging with the ground and the other containing the mechanical center of the machine. In order to preserve some degree of precision in the determinations, which are to be subject to corrections, one can evidently adopt the central position and establish a law of decrease while retaining the faculty of making any possible assumption respecting the exact location of the acoustic center of the machine. But this finally becomes a method for an expert in Acoustics. For routine measurements of an industrial nature, where a standardized test code is most often applied, the series of measurement points must be the same and at the same ...

location before and after substitution. This is the solution the authors have adopted to take the characteristics of a construction site compressor that was placed as well as the reference source successively in free field on a reflecting plane (unoccupied car park), facing the front of a building, and under an archway. For these two sources and in all the indicated positionings, measurements were made at 4 points at 1 m and 4 points at 7 m in accordance with the provisions of the ISO 2151 norm [7]. In every case, the height of the microphone was 1.5 m as prescribed by the norm. About 200 spectra were taken and enabled us to attain the results shown in the tables below.

TABLE I - Influence of the height above ground of the reference source (RSS) and of the measurement distance in free field on reflecting plane

Positionning adopted to determine the Lw of the RSS (positioning & measurement points after ISO 2151)	ΔLw re to calibration values in anechoic room on reflecting plane (Glasnl) Overall and octave bands								
	A	C	125	250	500	1000	2000	4000	8000
SOURCE ON GROUND									
. Points at 1 m compressor	+1	+1	+3	0	0	+1	0	+1	+2
SOURCE m. 0.85 ABOVE GROUND (Merging acoustic centers of SBR and compressor)									
. Points at 1 m compressor	+1.5	+2	+1.5	-0.5	+0,5	+0.5	0	+2.5	+2.5
. Points at 7 m compressor	+2.5	+3.5	+7	0	-5	+1	+0.5	+3	+3.5

TABLE II - Influence of the position of the reference source (RSS) on values of the acoustic power level determined on perturbed field

Positionning adopted to determine Lw of the RSS (positioning and measurement points after ISO 2151)	ΔLw re to calibration values, dB overall and octave bands								
	A	C	125	250	500	1000	2000	4000	8000
FRONT OF BUILDING (at 3m. from center of compressor)									
Lp_{1m} & centr. posit. RSS (1,2 fig.5)	+2	+0,5	0	+1,5	+2	+2,5	+2,5	+2,5	+2,5
Lp_{1m} & lat. high posit. RSS (4 fig. 5)	+3	+2,5	+0,5	+2,5	+4	+3	+4	+3.5	+3
Lp_{7m} & mean all posit. RSS (1,2,4 fig. 5)	+3.5	+2,5	+3.5	+3.5	+1	+3.5	+1.5	+2,5	+3.5
Dispersion of Lp_{7m} above	0,5	0,5	1	3	4	2	1.5	3.5	0
ARCHWAY OF BUILDING (lateral walls at 3 & 4 m from center of compressor)									
Lp_{1m} & mean all posit. RSS (1 to 4, fig. 5)	+3.5	+3.5	0	+3	+5	+5	+3.5	+2.5	0
Lp_{1m} : dispersion of Lp above	1,5	2	2	0,5	2	1	1.5	2	2
Lp_{7m} & mean all posit. RSS (1 to 4 fig. 5)	+11	+8	+6,5	+10	+11	+10	+9	+8,5	+5
Dispersion of Lp_{7m} above	2	1	4	1	0,5	2	2	0,5	0,5

TABLE III - Differences in estimation of acoustic power level between the reference source and the compressor for different degrees of reflection and different measurement distances

Principal measuring situations	ΔΔ Lw (reference source-compressor), dB for overall and octave bands								
	A	C	125	250	500	1000	2000	4000	8000
FRONT OF BUILDING									
. Series of measurement points at 1 m.	+0.5	-0.5	-0.5	+0.5	+1	+0.5	+1	+0.5	+0.5
. Series of measurement points at 7 m.	0	0	+0.5	0	-3	-0,5	-2,5	+2.5	+1.5
ARCHWAY OF BUILDING									
. Series of measurement points at 1 m.	0	+0.5	0	-0.5	+1.5	+1	-0.5	-2	-4
. Series of measurement points at 7 m.	+1	+1	0	+0.5	+0.5	-1	-2	-2.5	-3
Mean differences	+0.5	0	0	0	0	0	-1	-1.5	-1.5

...

The set of results given in the three tables finally shows that the position of the reference source in the space occupied by the substituted machine is all the less important when there is more reflection and when the measurement distance is shorter. The most difficult configuration appears to be the one with a single image source at such distance from real source that one may have an interferential system (source 0.85m above the reflecting plane and measurement at 7 m ; see last line of table I). Apart from 2 image sources of some importance because of the nearness of the reflecting planes (source near 2 walls at 2 or 3 m) and for industrial type tests, there would appear to be little need for concern regarding the location of the substitution source. Between these two cases, i.e. with two image sources (a wall at 2 or 3 m from the source), it should be noted that prudence is necessary for measurements in a near field (at 1 m) and at least 2 lateral positions (marked 4 in figure 1) should be used for the reference source.

Before closing this chapter on tests by substitution, outside buildings, and to conclude with the results of table III, it appears necessary to note that use of a substitution constant to correct the mean of points surrounding a source appears more realistic than making corrections point by point. It will also be noted that the use of an enveloping measurement surface, well centered and with points regularly spaced over the entire area - i.e. an arrangement affording the best conditions for evaluating the acoustic power level - would probably have led to a smaller dispersion of the differences in the upper octave bands.

So far, it has only been at matter of operating outside buildings because it is there that the most directive reflection fields are found and hence the greatest difficulties. Inside a building of moderate size, with little reverberation time, the substitution method leads to most satisfactory results. It is true that in that case we approach the more conventional methods of tests in reverberation rooms, in which there is seldom need to proceed with a substitution, und with which a test by comparison (location of reference source is not that of the machine tested) has the same value. Determination of the acoustic power of a dish-washing machine was made by the substitution method, first in a reverberation room of the E.D.F. acoustics laboratory (330 m3, TR (500 Hz) = 7 s), and then in a normal room with little reverberation time at L.C.I.E. (Laboratoire Central des Industries Electriques) (150 m3, TR 500 Hz = 0,5 s) where functional tests of household appliances are made. The following substitution formula was applied to measurements made at the six different microphone positions used in the room.

$$Lw_{(mach)} = Lp_{(mach)} + (Lw_{SBR} - Lp_{SBR})$$

with
$Lw_{(mach)}$ Power levels of source tested (values sought) in dB exceeding 10^{-12} W
$Lp_{(mach)}$ Mean pressure levels of the source tested in dB exceeding 2.10^{-5} Pa
$Lw_{(SBR)}$ Power levels of the reference source (calibration values) in dB
$Lp_{(SBR)}$ Mean pressure levels due to the reference source, in dB exceeding 2.10^{-5} Pa.

TABLEAU IV - Differences in value of the acoustic power level of a dish-washing machine determined in a laboratory (reverberation room) and "in-situ" (ordinary room)

Nature of differences indicated	Overall		Middle of octave bands, in Hz						
	A	C	125	250	500	1000	2000	4000	8000
ΔLw (labo "in-situ")									
. measured values	+1		+0.5	+1.5	-0.5	0	0	-2	-2
. recomposed values	0	+1							
Greatest Lp difference at 6 measurement points									
. reference source	2	2	0.5	3.5	2	2	2	2	3
. machine tested	3	2.5	6	3.5	2	3.5	3.5	2	4.5

3.4 - Juxtaposition method

Within the framework of experiments made to test the possibilities of the substitution method, complementary measurements were added by setting the reference sound source on the ground, under each end of the compressor, its axis of rotation being plumb with the outer wall of the machine. In building front tests, as in those under an archway, the power levels determined for the reference source were 3 dB below those estimated at the same locations but without the compressor. It can be assumed that radiation towards the top of the machine ...

was sufficiently attenuates for account to be taken of only half the energy radiated by the reference source.

In actual practice, it thus appears that, in order to exclude any uncertainty regarding the attenuation resulting from the structure of the machine on which a reference source is installed, an auxiliairy device has to be conceived, similar to the one described in fig. 8, ensuring radiation in a 1/4 of a sphere towards the free side and absorption of the radiated energy at the back of the second 1/4 of sphere.

3.5 - The comparison method

This method, in which the reference source is placed at a location other than that of the source to be tested, finally requires identification of the propagation curve on which one is in the room where the measurements are made. A few measurements (4 to 6) on one or both propagation axes usually suffice to establish from what distanse and at what value the acoustic pressure level becomes constant. From there, direct estimate can be made of the constant to add to the acoustic pressure read around the machine to be studied, either to know the level one would have at the same distance in free field, or to obtain the power level.

With this measurement as basis, and thus also for the needs of building acoustics, we can furthermore male direct estimate of reverberation time and the average absorption constantes $\Sigma S a$ of the room measurements made in different rooms successively with the reference source and the conventional method, show (see results, fig. 9) that the simplified method can permit linking with the standardized method by introducing a constant varying from -1dB for the octave band centered on 125 Hz, to +2.5 dB for the octave band centered on 4.000 Hz.

4 - USE OF THE REFERENCE SOUND SOURCE IN BUILDING ACOUSTICS

Everyday practive shows that there is often a great difference between the insulation values of partitions determined in a laboratory and those noted on the construction. A large proportion of acoustic energy can be transmitted indirectly and in this field the part played by the placement of materials is determinant, but it is difficult to assess accurately on the construction site the relationship of direct and indirect energy radiated by the walls. To attain this practical objective, J. PUJOLLE [8] recently proposed direct reading from a new type of insulation with the aid of a sound source emitting a pink noise. Taking up this idea, A. MILLET and D. de MONTUSSAINT [9] have tried to show what differences this method could lead to, in comparison with results obtained by conventional laboratory methods, when using the various reference sound sources that have at present been described in technical publications.

4.1 - Definition of a type of weighted insulation [8]

We know the defects of using a single value to characterize an average insulation in a more or less wide frequency band ; it is, however, necessary if we wish to multiply the checks. In this direction, French regulations on acoustic insulation recently introduced the notion of weighted insulation, by setting threshold limits to the weighted pressure level in dB(A) in far better relationship to the psycho-physiological properties of the ear. It is therefore of interest to calculate in emission and reception rooms the weighted acoustic intensity level by introducing the correction values of the standardized curve A. The value expressed in dB(A) will have no exact meaning unless the source produces a pink noise, i.e. a sound whose intensity by octave or third of an octave band is constant. Weighted overall insulation, that we designate by D_{GP} can be worked out by the following relation :

$$D_{GP} = 10 \log \Sigma I_1 \cdot a'_f - 10 \log \Sigma I_1 \cdot a_f^{-1} \cdot a'_f$$

with

- I_1 constant, sound intensity by band centered on frequency f (pink noise)
- a_f transmission loss of the partition
- a'_f correction terms corresponding to standardized response curve A

If we have a pink noise source, D_{GP} is directly measured. If a_f is known (a'_f being a well defined constant D_{GP} is calculated.

4.2 - Direct measurement of the weighted insulation with the aid of the reference source [9]

The spectra of different reference sound sources whose characteristics are given in fig. 4 were used to calculate the weighted insulation of the partitions, the transmission losses of which are given in fig. 6. The difference between the results obtained and those given by a true pink spectrum is indicated in fig. 7.

...

The set of results shows that while the presence of high frequencies in the spectrum of the source used is of little importance, the problem of low frequencies is critical. The accuracy of the method is based on that fact. It appears, however, that the EDF-AIRAP source leads to satisfactory results, since the dispersion of differences is below 1 dB. The result obtained by direct measurement will be systematically from 0.5 to 1 dB higher.

5 - CONCLUSIONS

The various experiments made with a reference sound source as intermediary lead to overall values in dB(A) which show a difference of not more than 1 dB in comparison with those obtained by more accurate procedures. Hence, and for measurements of an industrial type, we can consider that the great simplification and saving in time in no way affect the accuracy required for by this type of determination.

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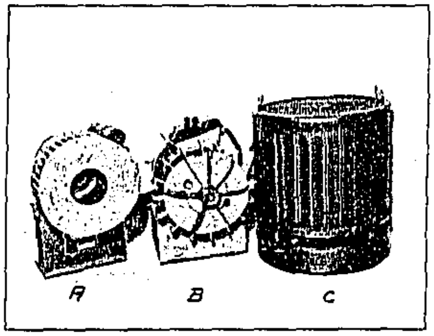


Fig.1 - View of different tested Reference Sound Sources

A . ILG (USA)
B . EDF.CEM (France)
C . EDF.AIRAP (France) (type2)

Fig.2 - Open View of the reference Sound Source

EDF. AIRAP
(type 1)

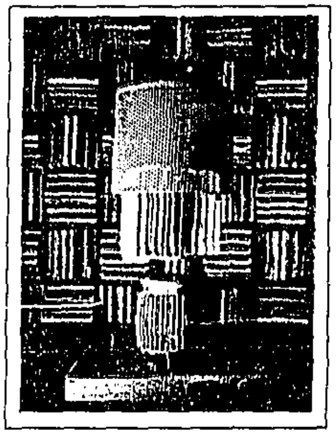


Fig.3 - Directivity index variations of reference source EDF - AIRAP

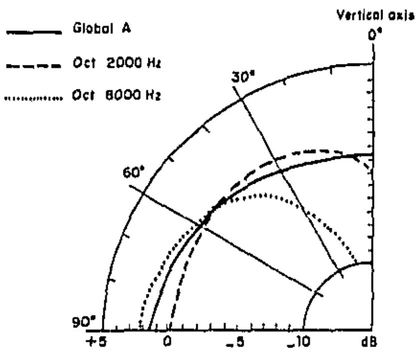


Fig.4 - Power levels of different tested Reference Sound Sources

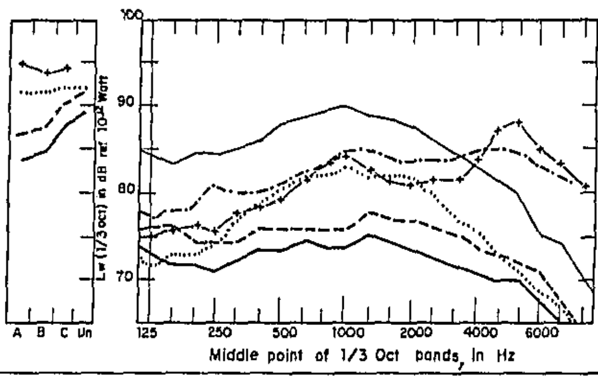


Fig.5_Possibilities for implantation of the Reference Sound Source in the volume defined by machine above the floor

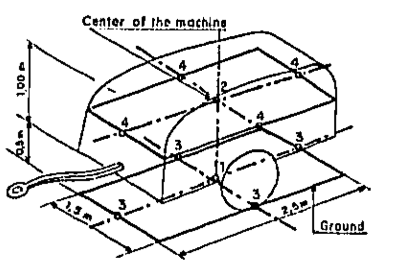


Fig.6-Insulation of the tested partitions

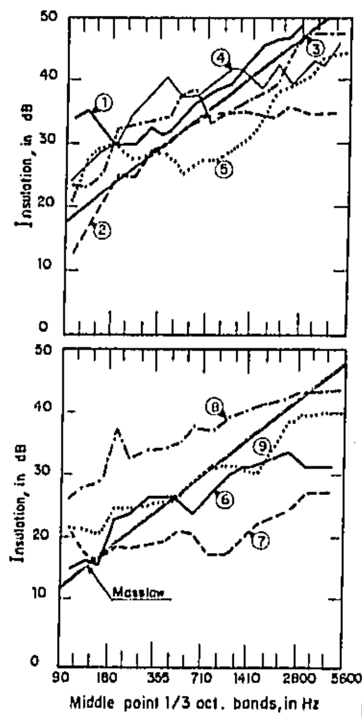


Fig.7_ Difference beetwen global isolation obtained with pink noise and with the noise of reference sound sources

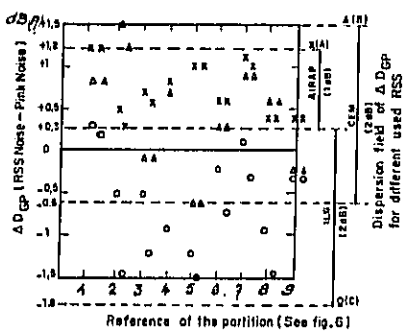


Fig.8_Preconised disposition to assume a radiation of the reference sound source (RSS) in a 1/4 of sphere

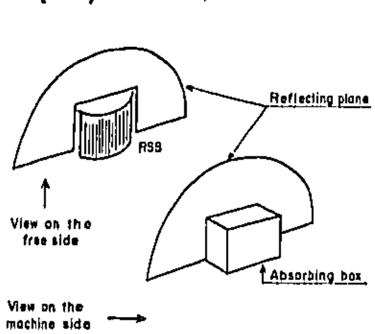
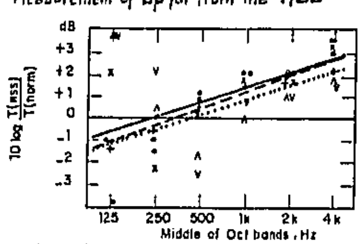


Fig.9_ Differences obtained by determining Reverberation Time directly with the Measurement of Lp far from the RSS



Rooms	•	•	x	+	∧	∨
Volume (m³)	64	33	18	330	48	43
T.R (s) at 500 Hz	2,5	2,2	1,1	7	0,15	0,25

SOUND POWER DETERMINATION OF MACHINES IN SITU

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INTRODUCTION

National and international groups now are preparing standardization documents for the noise measurement on machines. The main quantity characterizing the noise emission of a source is its sound power. This quantity, in contrast to the sound pressure, is relatively independent of the acoustical environment and is the most important quantity for acoustical planning and design.

Unfortunately, we do not possess any practical and inexpensive equipment for the measurement of the actual sound power, therefore sound power must be determined from sound pressure measurements under certain measuring conditions. Three methods for such a power determination are known:

- the reverberant method, by which the sound pressure of the sound source is measured in a closed space under reverberant field conditions,
- the free field method, by which the sound pressure of the source is measured on a surface enveloping the source under free field conditions,
- the reference sound source method, by which, under given environmental conditions the measured sound pressure values of the sound source are compared with measured sound pressure values produced by a sound source of calibrated sound power output under the same environmental conditions.

Each of these methods is based on some premises which can be fulfilled only under certain ideal measuring conditions. Therefore in practice the sound power output of a sound source can be determined only with more or less accuracy. The main question in this field is the correlation between accuracy in sound power determination and sound pressure measurement conditions. These conditions concern requirements

- for the measuring environment (sound field qualification)
- for the microphone and source locations and
- for instrumentation and for computation of the measured values.

This paper deals with the sound power determination for sound sources as for machines under exact and approximate free field conditions, using the so-called "Method of Enveloping Surface". The magnitude of the errors in this sound power determination will be discussed in two steps:

first: regarding exact free field conditions respectively free field sound radiation over a reflecting plane

second: allowing environmental conditions as usual in ordinary rooms ("In Situ").

THE METHOD OF ENVELOPING MEASURING SURFACE

The determination of sound power of a source in exact free field or nearly free field conditions with the aid of method of enveloping measuring surface is based on the premise that the sound power output of the source is directly proportional to the mean-square sound pressure averaged over a surface enveloping the sound source and averaged over time, $\langle p^2 \rangle$, proportional to the area of the measuring surface S, and otherwise depends only on the physical constants of air density ϱ_0 and velocity c of sound:

$$\tilde{W} = \frac{1}{\varrho_0 c} \cdot \langle p^2 \rangle \cdot S \qquad (1)$$

The measuring surface completely envelops the sound source ' . a given measuring distance d and ends on the reflecting ground plane (floor) on which the source is installed (see Figure 1).

$\tilde{W}$ given in Eq.(1) approximates the true sound power output W of the source better

- for greater measuring distances d and
- for higher frequencies, if atmospheric attenuation is neglected
- for sources with small directivity

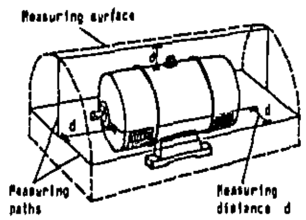


Figure 1: Measuring Surface

- for a large number of microphone positions completely covering the measuring surface (Methods only using one or two measuring paths, see Figure 1, instead of the total surface may lead to greater errors)
- for high quality instrumentation.
- for negligible environmental sound reflections (Such reflections interfere with the "free field condition").

These measuring conditions only are approximatively satisfied in practice, both for measurements in a semi-anechoic room (laboratory) and usually more in rooms in which machines are mounted as installed (machines "in situ"). These insufficiencies lead to errors which can be divided up into four partial errors:

$$\Delta_{tot} = \Delta_1 \cdot \Delta_2 \cdot \Delta_3 \cdot \Delta_4 \qquad (2)$$

Each of these partial errors Δ_i corresponds to one or several measuring condition parameters:

Δ_1, called "Near Field Error", is caused by the fact that the desirable very large measuring distances expressed in wave-length of interest usually cannot be chosen. (The exact theoretical definition of this partial error is given in a special paper, see reference [1]).

Δ_2, includes the influence due to the limited number of sound pressure values.

Δ_3, includes the fluctuations, which are caused by instrumentations, observers, metereological conditions etc.("Actual Measurement Error")

Δ_4, is caused by undesired sound reflections of room boundaries and of other objects in the room ("Environmental Error").

The partial errors Δ_1, Δ_2 and Δ_3 belong to the first step and the error Δ_4 to the second step mentioned in the introduction.

ERRORS IN SOUND POWER DETERMINATION FOR SOURCES UNDER EXACT FREE FIELD CONDITIONS ("first step")

The near field error Δ_1 was investigated in detail and discussed on the occasion of the last meeting of the Acoustical Society of America in Buffalo, April 1972. These results and further information about errors Δ_2 and Δ_3 will be published in JASA, January 1973(see reference [1] to the same subject see also references [2], [3] and [4]). Some results of these theoretical and experimental investigations are:

- Sound power $\tilde{W}$ determined from sound pressure measurements according Eq.(1) always remains larger than or equal to the true sound power W:

$$\tilde{W} \geqq W \tag{3}$$

(see also Figure 3: K-values of reverberant room)

- The enveloping measuring surface should follow the contour of the source at a constant distance (constant measuring distance d), in order to minimize the error Δ_1.
- The differences between $\tilde{W}$ and W remain within a range of 5 dB, if the frequency range of interest is extended only down to 100 Hz and if using a measuring distance not smaller than 0.25 m and having sound sources of "normal" directivity.
- Smaller and medium sized machines usually have a directivity comparable with that of a source between a monopole and dipole.
- At higher frequencies ($f > 4$ kHz) the accuracy of sound power determination depends on microphone diameter more than on any other measuring condition (1/2" microphones are recommended).

ENVIRONMENTAL ERRORS IN SOUND POWER DETERMINATION ("second step")

The environmental error Δ_4, caused by undesired sound reflections of room bounderies and of other objects situated in the room, can be investigated experimentally by a method which will be described as follows:

A sound source of constant sound power output first is mounted on the floor of a semi-anechoic room and measured according to the requirements of the method of enveloping measuring surface. The result of this sound power determination should be called:

$\tilde{W}_{\text{semi-anechoic}}$ respectively its level $L_{\tilde{W}\ \text{semi-anechoic}}$

Then this sound source is mounted in a given position on the floor of the ordinary room of interest. Sound power determination is repeated under exact the same conditions, meaning the same measuring distance, the same measuring surface, the same microphone array, same instrumentation, observer and, if possible, similar metereological conditions. Result:

$\tilde{W}_1$ respectively the level $L_{\tilde{W}_1}$

Then we define the environmental influence Δ_4, respectively $K = -10 \lg\ \Delta_4$, by Eq.(4):

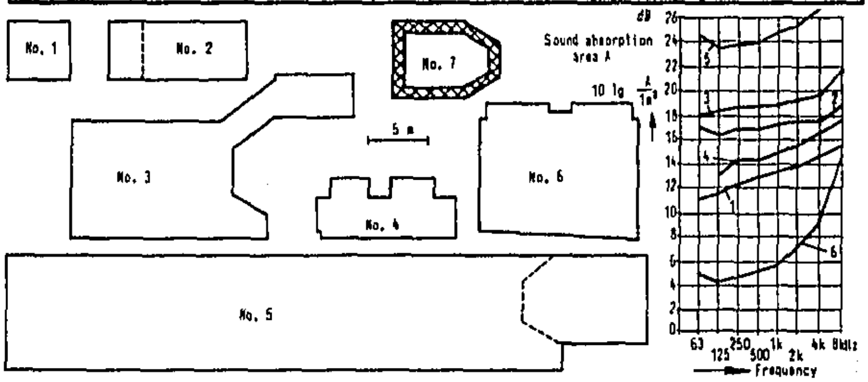
$$\Delta_4 = \frac{\tilde{W}_{\text{semi-anechoic}}}{\tilde{W}_1} \quad \text{respectively} \quad K = L_{\tilde{W}_1} - L_{\tilde{W}\ \text{semi-anechoic}} \tag{4}$$

This method to evaluate environmental influence was carried out for 5 ordinary rooms being explained by Figures 2 and for 4 sound sources different in sound spectrum and directivity. The sources used are:

1. Vacuum cleaner motor with fan, vertical shaft, positioned 0.5 m above the floor, size 0.17 m diameter,
2. Small rotating electrical machine, 0.7 kW, 3000 rpm, size: 0.18 x 0.18 x 0.20 m height
3. Medium sized rotating electrical machine, 11 kW, 2100 rpm, size: 0.87 x 0.36 x 0.35 m
4. Teletype, size: 0.56 x 0.46 x 0.30 m

The sources 2,3 and 4 had been mounted on the floor, all sources near the floor center.

room No	room	dimensions volume m^3	surface m^2	height m	size (rough values) m x m x m	$\frac{1}{3} V^{\frac{1}{3}}$ m	remarks
1	office room	83	119,4	2,9	5,6 x 4,9 x 2,9	1,45	absorption only on the ceiling, hard walls, hard floor, few furnitures
2	office room/laboratory	133	172	2,95	8,9 x 4,9 x 2,9	1,7	absorption only on the ceiling, a great lot of furnitures, equipments
3	cellar/machine basement room I	671	600	3,75	17 x 10,1 x 3,75	2,92	room with complicate groundplane, very hard bounderies, serveral equipments, machines
4	cellar/machine basement room II	161	224,7	2,63	12,9 x 5,5 x 2,63	1,83	very hard bounderies; few machines
5	laboratory test hall	4378	2139	9,0	54 x 9,8 x 9,0	5,5	large hall with hard bounderies; one of the long walles is formed by large windows; serveral low equipments
6	reverberant room	151	187,5	3,75	6,7 x 5,9 x 3,75	1,75	rectangular room; no diffusors
7	semi-anechoic room	222	208	5,75	7,7 x 5,3 x 5,75	2,02	no rectangular floor; walls and ceiling with absorbing wedges (90 cm lenght); floor reflecting



Figures 2: Test Rooms being Used

The determined K-values are plotted in the Figure 3. These results show:

- For a range of measuring distances 0.25 ... 1.0 m the increase of sound levels caused by environmental influence is usually less than 6 dB for frequencies down to 200 Hz. For the measuring distance of 0.25 m and smaller sources (diameter < 0.5 m) the room influence in most cases is slight (< 3 dB).
- If testing an ordinary room on a given measuring surface the influence of the individual characteristics of the test sound source usually can be neglected. (The K-curves in Figures 3 determined for different sound sources are situated near together).

The error K depends not only on the acoustical qualification of the room boundaries but depends for the same room also on the measuring distance and the source location. If characterizing the boundary qualification by the total sound absorption area A and if characterizing the source size and the measuring distance by the area of the measuring surface S the experimental results suggest to assume, that the K-values are mainly a function of the both parameters A and S, possibly a function of the ratio A/S. In order to proof this idea our K-values are plotted in function of this ratio (see Figures 4 on the next page).

(The A-values of our rooms are given in Figure 3, the areas S being used lie in a range of 1,7 m^2 to 18,2 m^2 resp. its level:2.3dB...12.6dB)

Figure 4 shows: if the ratio A/S is greater than 10 the (mean) error K_m is equal or smaller than 1 dB. For other A/S-values the following estimate can be used:

Table I

A/S	< 4.0	4.0...6.3	6.3...10	> 10
K_m in dB	> 3	3	2	< 1

or the formula:

$$K_m = \frac{15}{10 \lg(A/S)} \qquad (5)$$

Conclusions: The sound power level L_W of machines being mounted as installed can be determined by the method of enveloping surface and by correcting the "Environmental Influenced Sound Power Level" $L_{\overline{W}_1}$ according the following equation:

$$L_{\overline{W}} = L_{\overline{W}_1} - K \qquad (6)$$

The "Room Correction" K (=room influence error) usually can be neglected for area ratios A/S > 10, otherwise K can be determined:

- by using a test source of known and constant sound power output being measured on the same measuring surface as the sound source should be tested in the ordinary room or
- by an estimate survey method according Table I or Eq.(5) for A/S-values between 10 and 4.

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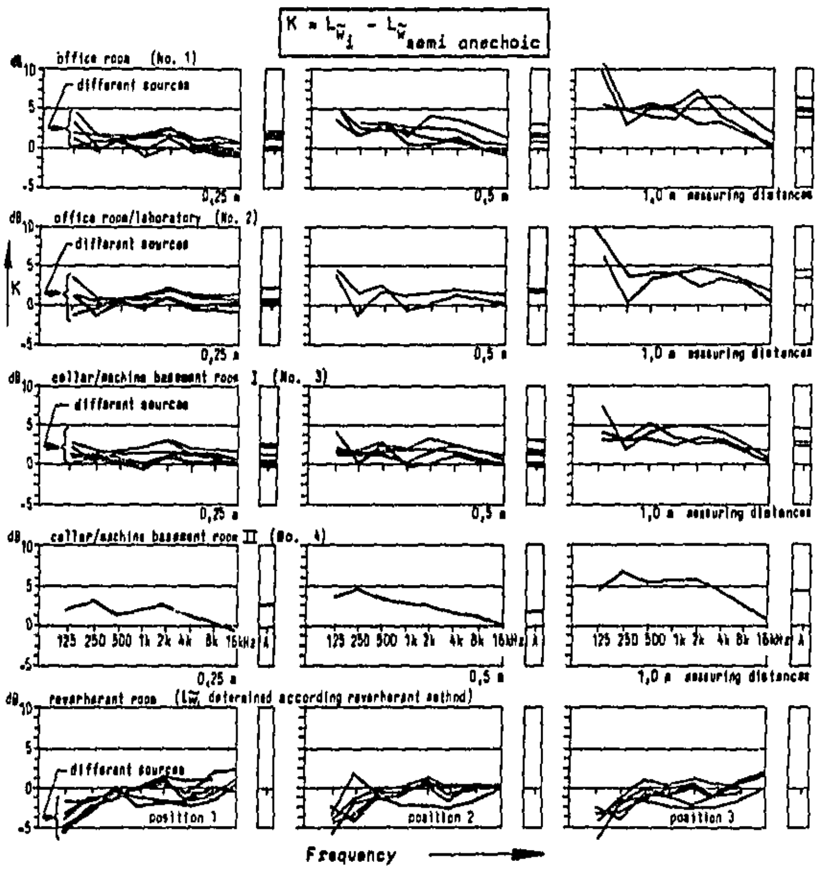

Figure 3: Differences K between Sound Power Levels Determined in Different Rooms

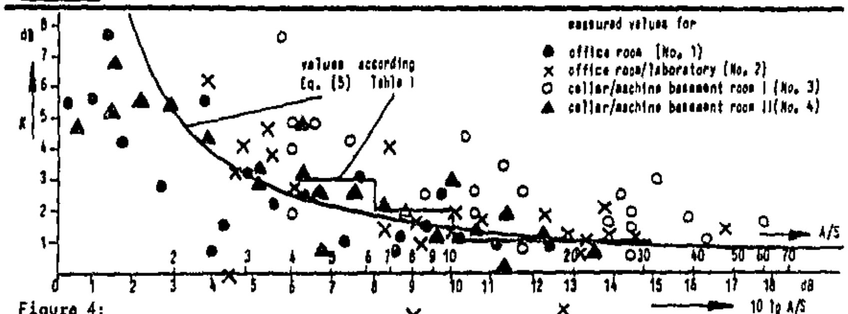

Figure 4:
Room Correction K over Ratio A/S (A sound absorptions area ; S area of measuring surface)

MEASUREMENT MICROPHONES.

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Brüel & Kjær, Denmark.

Noise measurements are carried out under a variety of environmental conditions demanding stable performance as well as correctly reproduced signals at levels ranging from below the threshold of hearing into high pressure fields.

Due to the nature of sound the preferred measurement microphones must respond to pressure variations. Although the piezoelectric microphones may be useful, they do not offer the well defined geometry and wide frequency range with a correct damped resonance as the condenser microphones does. Although very thin piezoelectric diaphragms may be made to day, they cannot yet give the same performance as a stretched metal diaphragm. The thickness is still some 50 times greater than that of the metal diaphragm and for about the same density of the diaphragm material, this will reduce the useful frequency range.

Electrets using Teflon or F.E.P. foils are being used also for low cost microphones. It is a promising solution for low cost measurement microphones, although a number of problems make them less useful for this purpose compared against the stretched metal diaphragm type. The problems are stability problems due to ageing with time, changes in sensitivity with normal handling, temperature stability, sensitivity to moisture and sensitivity to shock and vibration. Also the sensitivity for the same size is inherent lower for the electret type than for the well designed stretched diaphragm type.

The fundamental selection to be made in any sound measurement is that of a microphone of suitable frequency range sensitivity and dynamic range. Each of these parameters is a function of microphone size. A range of microphones are available to select between fig. 1 and fig. 2.

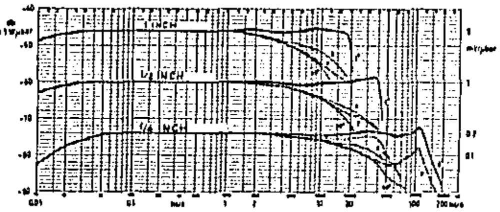


Fig. 1. Frequency response of Free-Field Microphones.

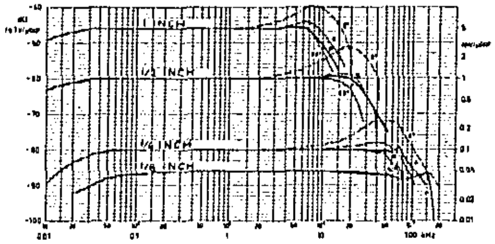


Fig. 2. Frequency response of Pressure Microphones.

The difference between the series of microphones shown in fig. 1 and those of fig. 2 is that of the damping of the diaphragm in the high frequency region. The "Free field microphones" are damped in order to compensate for the pressure increase caused by diffraction of the sound waves impinging at 0° of incidence on the diaphragm. The importance of using this type for general noise measurements may be illustrated by the situation shown in fig. 3.

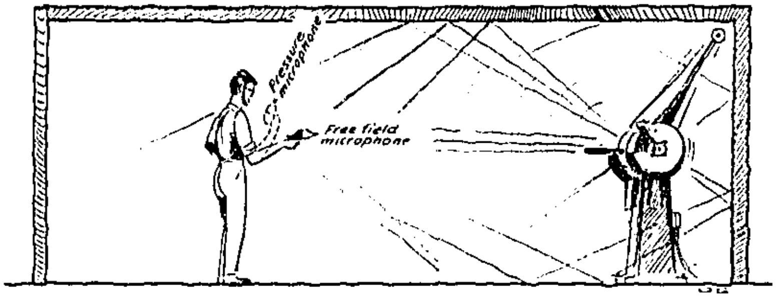


Fig. 3. Noise measurement using free field microphone or pressure microphones.

Using a free field microphone pointing at the sound source will always give a reading close to, and never higher than the correct value being the one at the microphone, when no microphone is present.

Using a pressure or random incidence corrected microphone will give a reading which may be correct, if no reflections occur, but a reflected wave may be measured up to 10 dB larger than it actually is, and will cause a larger reading than the correct one.

A free field microphone may be turned into an omnidirectional microphone with a flat random response over a fairly large frequency range by replacing the normal protection grid by either a random incidence corrector for 1" microphones or a nose cone for ½" microphones as shown in fig. 4.

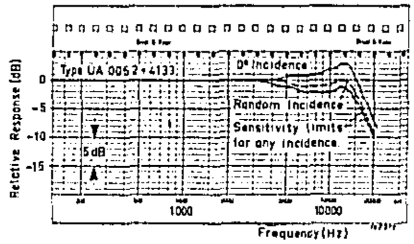


Fig. 4. Free field response of a half-inch microphone equipped with nose cone. The omniderectivity and linearity are effective within ± 3 dB up to 15 kHz.

The pressure microphones are used whenever one should like to measure the true pressure at the place of the diaphragm, such as closed coupler measurements, measurements on the surface of bodies and for tubes etc.

During a 14 year period the present design of the microphones has basically been the same. However small improvements in design and manufacturing technique have been made. The stability have been improved. The calibration accuracy has been improved in order to determine the long term stability, and the record over a number of microphones over four years, measured with 0.3 % accuracy, show less than 0.8 % change as maximum deviation. The ageing procedure is described by Erling Frederiksen in ref. 1. The present 1" standard microphone is shown in fig. 5.

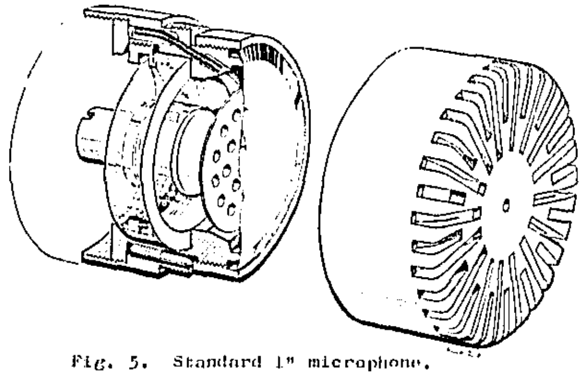


Fig. 5. Standard 1" microphone.

As shown in fig. 5 the microphone housing is made in one solid part with strong walls. Only the mounting thread is rather thin as prescribed in the standard for type "L" microphones. The diaphragm is highly polished 5 μ thick nickel and the back plate also highly polished nickel alloy made to within ± 0.5 μ, tolerances. The highly polished surface, clean room assembly and very tight tolerances ensure a high degree of accuracy and reliability.

A major problem in any condenser microphone is condensation of moisture. Due to the very fast heat exchange in the thin diaphragm material, condensation may easily take place when the microphones are operated close to the dew point. Therefore a new method has been developed to protect the microphones against moisture and to reduce noisy operation under extreme humid conditions. A thin film of around 0.5 μ of SiO_2 without pinholes is deposited on the back plate inside the microphone in order to protect against internal short circuits and a similar film is deposited outside the diaphragm to protect against corrosion, see fig. 6.

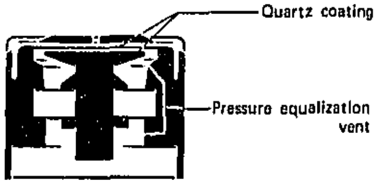


Fig. 6. Quarts coated 4149 microphone.

A series of tests have been carried out to check the value of this treatment under 100 % humidity. The result of one year test of 10 normal ½" microphones, 10 with back plate coating alone, and 10 with diaphragm coating alone, is shown in fig. 7.

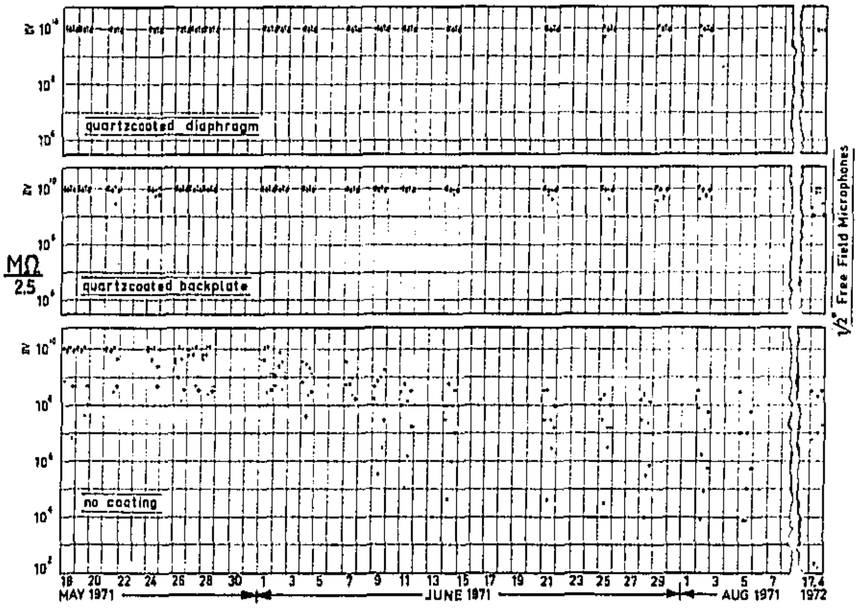


Fig. 7. Leakage as function of time for 3 x 10 ½" microphones exposed to 100 % humidity.

These microphones are especially made for severe environments as outdoor use over long time. Eight ½" 4149 microphones have been mounted outdoor and monitored every day over the last 1½ year. None of these have failed so far. Previous tests have shown a Mean Time Between Failure of 2 years for microphone for a similar number. So far the same M.T.B.F. one should already have some failures. It should therefore be safe to predict a good improvement in the M.T.B.F.

The lowest sound pressure that could be measured by a condenser microphone has been determined by the noise from their preamplifiers. Viggo Tarnow (ref. 2) has shown that the noise generated by thermal vibrations of the diaphragm is higher than the noise of a well designed preamplifier. The noise voltage density for as well 1" as ½" microphones and preamplifiers are shown in fig. 8 and fig. 9.

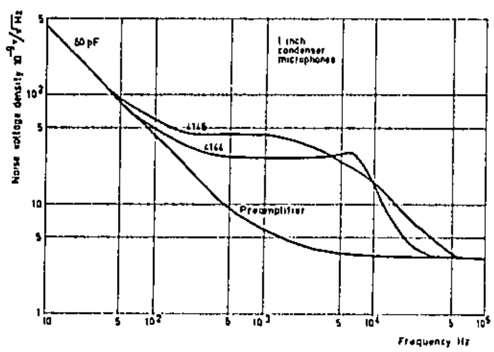


Fig.8.
The noise voltage spectral density for one inch condenser microphones and the preamplifier. The upper curves present the noise of the preamplifier alone.

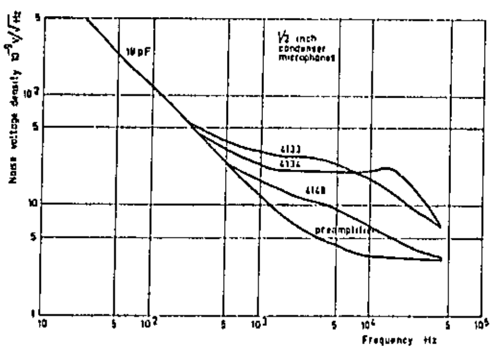
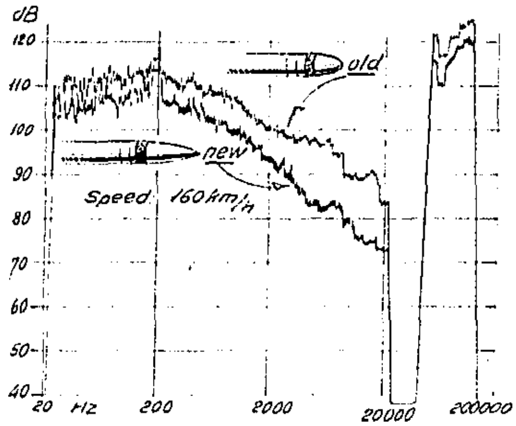


Fig.9.
The noise voltage spectral density for half inch condenser microphones and the preamplifier. The lowest curve presents the noise of the preamplifier alone.

Acoustical signal detection in turbulent air streams may require a good knowledge of the limitations set by the microphone and the improvements obtained by available windscreens, nosecones and other adaptors. A new series of nose-cones have been developed, they will minimize the selfinduced noise 6-10 dB in the high frequency range. (ref. 3). They have been measured in an anecoich chamber, using a swingarm technique, which allow one to measure the performance at speeds up to 100 mph, taking 1/3 octave spectograms at various constant speeds from 3 mph and up. See fig. 10 and fig. 11.

Fig. 10.
1/4" microphone with old and new nosecone, measured at 160 km/h and 0° incidence in 1/3 octave bands.



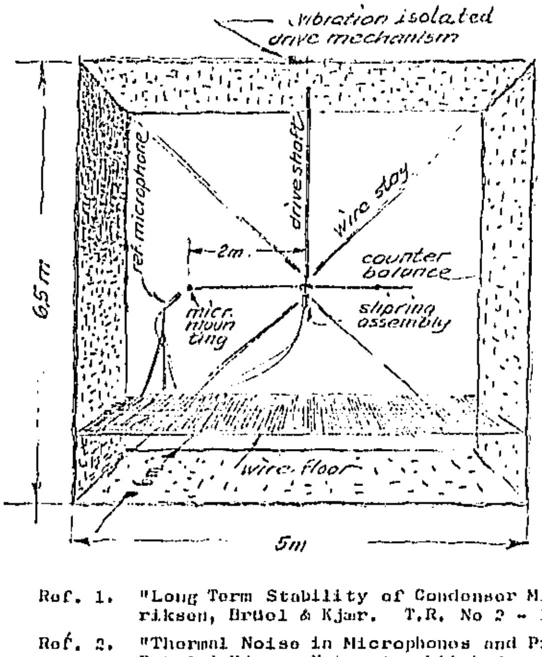


Fig. 11.
Anechoic chamber with rotating vane and motor drive control from control room. The rpm is monitored by photocell arrangement and electronic counter.

Ref. 1. "Long Term Stability of Condenser Microphones" by Erling Frederiksen, Brüel & Kjær. T.R. No 2 - 1969.

Ref. 2. "Thermal Noise in Microphones and Preamplifiers" by Viggo Tarnow, Brüel & Kjær. Not yet published.

Ref. 3. "Windscreening of Microphones" by Gunnar Rasmussen, Brüel & Kjær. Paper given at the 7th AES convention.

TECHNIQUES FOR SAMPLING ENVIRONMENTAL NOISE

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It comes as no surprise to those concerned with community noise problems that the last major noise survey in this country was done 25 years ago.[1] There have been more recent surveys in Europe. From these surveys came the Traffic Noise Index (TNI)[2] and the Noise Pollution Level (L_{NP}).[3] Both of these proposals take into account the fluctuating nature of environmental noise. The inclusion of the fluctuation into the rating scheme significantly improved the correlation between the ratings and the subjective response to the noise in all cases. Therefore, one must look at the noise in the frequency domain and the time domain. Noise in the time domain may be even more significant because people find it harder to live with a fluctuating noise level then with a steady, neutral background noise that they can get used to.

The first step toward simplification has been to use the A-weighted sound level instead of 9 octave frequency bands. The A-weighted level has proved adequate for transportation noise. Some believe the A-weighted level underestimates the annoyance of intense low frequency noise, especially in the 31.5 Hz and 63 Hz octave frequency bands. The A-weighted level does not adequately measure the annoyance of discrete frequency components in a broad-band noise environment. Both the low frequency and tone problem are most common near stationary noise sources. However, the A-weighted level is a useful descriptor for most noise sources and it is readily available on all sound level meters.

NOISE SAMPLING TECHNIQUES

The common method for evaluating the fluctuating character of environmental noise is to determine the decile A-weighted sound levels, L_{90}, L_{50}, L_{10}, L_{1}, respectfully, the levels that are exceeded 90%, 50%, 10% and 1% of the time, over a period of one or more hours. How does one determine these values with only a sound level meter? The American National Standards Institute (ANSI) has proposed a procedure.[4] The sound level meter is to be read as follows:

(1) Observe the A-level reading on the sound level meter (slow dynamic characteristics) for five (5) seconds and record (a) the best estimate of central tendency and (b) the range of the meter deflections, during that 5-second period in decibels.

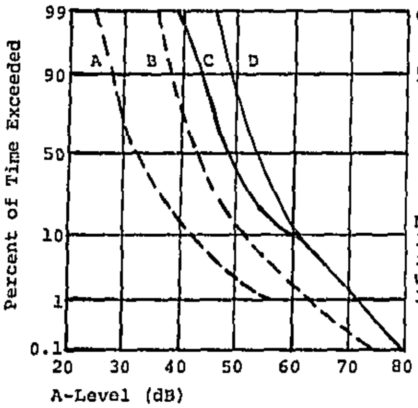
(2) Repeat the observations of step (1) until the number of central-tendency readings equals or exceeds the total range (in decibels) of all the readings.

(3) Find the arithmetic average of all the central-tendency readings in (1) and (2) above, and call this estimate the community noise level for this particular measuring time and location.

This procedure is an attempt to average out the fluctuations of outdoor environmental noise. Yet, Schultz has shown[5] this procedure can lead to gross errors for widely fluctuating noise.

To avoid these problems investigators of environmental noise usually tape record the noise in the field and perform detailed statistical analysis (cumulative distribution) of the A-weighted noise level. An example of this type of analysis is shown in Figure 1. With presently available instrumentation statistical analysis requires an analysis time equal to the noise recording time. In order to reduce the recording time, and analysis time the noise exposure is sampled a few minutes at each site rather than make a continuous day-long recording. A good example is the London noise survey.[6] In this survey the noise was sampled for a continuous 10 minute period at each site. The cumulative distribution (L_{90}, L_{50}, L_{10}) was then determined for that period of time.

Figure 1 - Noise Exposure Outside GWK Home



Cumulative Distribution of A-Weighted Levels Measured June 1971

Curve	Position	Time
A	Patio	0200-0400 (no aircraft)
B	Roof	0200-0400
C	Patio	0700-2000
D	Roof	0700-2000

Note that $L_{10}-L_{90}$ increases from 10 dB to 16 dB when the microphone is moved from the roof to the patio where the exposure is reduced from 360° to 180° respectfully.

A study was made by the C.S.T.B. (Scientific and Technical Center for Building Construction) Paris,[7] on the differences in the statistical results for sampling urban noise for a period of 2 minutes, 5 minutes, and 10 minutes each hour versus statistical analysis for the entire hour. Their study showed the gross errors that are encountered by selecting a finite sample (2, 5 or 10 minutes) from an hour long recording. They found that the probability of obtaining the correct level within ±2 dB for L_{90}, L_{50}, L_{10}, L_{1}, with a 2 minutes sample duration, ranged from 5% to 50%.

A more accurate method for determining the cumulative distribution over a one hour period using a 2 minute sample is to divide the 2 minute sample up into many micro-samples evenly distributed over the hour. For example, the noise environment might be sampled for 5 seconds every 2½ minutes to accumulate 2 minutes of data at the end of an hour. This technique is the main subject of this paper.

It is hoped that some day in the near future there will be an inexpensive and portable instrument available that will continuously compute L_{90}, L_{50}, L_{10}, and L_{1}, of the A (or X) weighted noise level. At hourly intervals the instrument would store the cumulative distribution decile values for read-out at a later time. The major disadvantage with this instrument would be the loss of noise source identification and detailed frequency spectra of the recorded noise. Unless the major noise source(s) present at the monitoring site are clearly identified there is no substitute for tape recording the broad-band noise spectra.

MICRO-SAMPLING

Numerous recordings were made outside the G. W. Kamperman home to validiate the micro-sampling technique. The first tests carried out two years ago consisted of a one inch Bruel and Kjaer condenser microphone above the roof feeding a cassette tape recorder and a Bruel and Kjaer statistical distribution analyzer. With the assistance of an electro-mechanical timer, the tape recorder was activated 10 seconds every 5 minutes. After several hours the tape recorded signal was played back through the Bruel and Kjaer statistical distribution analyzer (set for cumulative distribution) and the micro-sampled results compared against the full time cumulative distribution analysis for the same time period. The results looked encouraging. Some of the results were published this year in Sound and Vibration.[5]

Recordings were made with the microphone amplifer on both "flat response" and "A-weighting". Because of the finite (low) frequency response and limited dynamic range of the cassette tape recorder, A-weighting the signal before recording gave much better correlation with the full time cumulative distribution.

Errors with micro-sampling

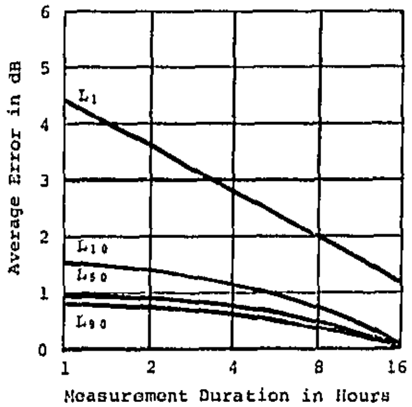
Last year the experiment was repeated with the microphone located outside the G.W.K. home. There were two improvements made: a camera to photograph the Bruel and Kjaer cumulative distribution counters at the end of each hour and a cassette tape recorder that operated continuously except that the tape capstan drive was engaged by a clock controlled solenoid five seconds every five minutes. This change in the tape drive eliminated all transients or dead spots between each five second recording.

Ten second recordings were adequately long to subjectively identify the noise source and perform detailed frequency analysis of any one sample. It was believed that the sample length could be reduced without a significant loss in statistical accuracy. However, it was found that it became difficult to subjectively identify noise sources when the sample length was reduced to less than four to five seconds although statistical accuracy was not materially affected.

The typical distributions measured outside the G.W.K. home are shown in Figure 1. A Gaussian distribution would give a straight line on this figure. The noise level exposure between 60 and 80 dBA is due to overhead jet aircraft operating in and out of O'Hare field, 15 miles away. Although there may be many overflights in any one hour, the duration of the sound above 60 dBA is relatively short. Many overflights are missed by sampling the noise only once every five minutes. However, the error decreases with increasing elapsed sampling time.

The micro-sampling (5 seconds every 5 minutes) error relative to the full time cumulative distribution analysis is shown in Figure 2. These data are for the skewed distributions shown in Figure 1. Aircraft noise controls the L_1 level. It is seen that the average error in the L_1 level decreases approximately 1 dB per doubling of elapsed time to about 1 dB after 16 hours (16 minutes of tape recorded data).

Figure 2 - Micro-Sampling Measurement Error



Average Measurement Error Using 60 to 1 time compression versus full time cumulative distribution analysis. See Figure 1 for cumulative distribution. L_1 levels determined by aircraft overflights.

The micro-sampling technique consistently underestimated the true contribution of aircraft noise (L_1). The maximum error observed in the L_1 level by micro-sampling five seconds every five minutes was 10 dBA for a two hour period. The maximum error in the L_{10}, L_{50}, L_{90} levels never exceeded 2 dBA over a two hour period (2 minutes of data).

The study by the C.S.T.B. on street traffic noise in Paris[7] showed very large errors using a continuous two minute sample. They found that the probability of obtaining the correct level within ±2 dBA (using a 2 minute continous sample) over a one hour period to be: L_1 5-50%, L_{10} 14-48%, L_{50} 30-50%, L_{90} 26-50%. It may not be fair to compare these two situations directly. However, it is significant that two minutes of micro-samples (5 seconds each) always met the ±2 dBA criterion, except the L_1 due to aircraft.

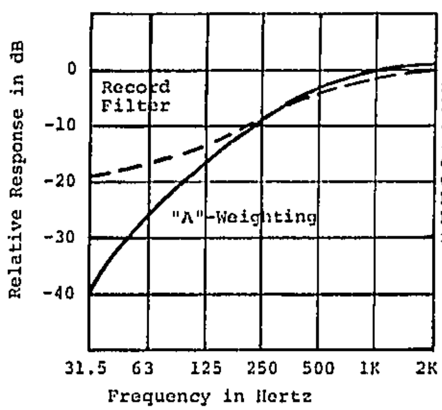
The micro-sampling procedure is a way of dealing with the fact that environmental noise level fluctuations are not, within the usual time span of observation and analysis, a truly random process. One would prefer many samples per hour, say one per minute or even two per minute if aircraft overflights are important.

A micro-sampling recorder

The objective is an inexpensive, reliable field data recorder. The micro-sampling recorders discussed thus far required AC power. Field data recording usually means a battery power supply. Thus, power consumption is important. To avoid undesirable gaps between each short data sample the tape drive system must have low inertia to permit rapid start and stop. It will probably be necessary to power and operate the noise recording microphone amplifier full time to avoid undesirable gain changes and transients. A clock mechanism will be necessary to start and stop the tape. It may be desirable to extend the recording over several days, thus the clock must be accurate, probably crystal controlled. The recorder and accessories must be protected from the environment. Wind and rain will present a severe challenge to the microphone. There is a need for the recorder described above, but few exist. What follows is a brief description of a recording system aimed at meeting these objectives.

The Sony TC-40 meets the objectives with a few modifications and additions. The built-in microphone is removed about a foot from the recorder to improve its non-directional characteristics and to avoid pickup of structure-borne motor noise. The automatic gain control is defeated. In its place is a simple attenuator that reads: 60, 80, and 100 dBA full scale. The microphone signal is not A-weighted but instead the microphone amplifier has the response shown in Figure 3. This is a 20 dB high-pass filter with ½ power points at 63 and 630 Hz. This filter permits the recorder to utilize more of its useful dynamic range. An inverse of this filter is inserted during data analysis to obtain a uniform frequency response from 25 Hz to 10K Hz for the complete record and playback system. A-weighting before recording on tape is not recommended. The A-weighting discriminates so severely against low frequencies that the finite dynamic range of the tape recorder (at low frequencies) may not permit reconstruction of the original noise spectrum on playback.

Figure 3 - Filter For Recording Outdoor Environmental Noise



Suggested Filter Response to be inserted between microphone and tape recorder. Filter will increase useful dynamic range of tape recorder and minimize wind noise. "A"-weighting before recording should not be used if spectra are to be reconstructed on playback.

The amplifier in the Sony TC-40 is energized at all times. Tests showed that it required 5 to 10 seconds for the record amplifier to stabilize after being energized. The capstan pinch roller is engaged whenever the amplifier is energized. The recorder drive motor is controlled by a digital clock. The motor is energized normally to start the tape. To stop the tape abruptly, the motor drive is baised off and the motor shorted for dynamic braking. The total stop-start time is less than 50 milliseconds. This short pause between recorded microsamples permits valid samples as short as one second or less. Such short samples have statistical value but they are hard to deal with subjectively or perform frequency analysis on.

The tape recorder sells for about $100 and the components for the digital clock can be obtained for about half the cost of the recorder. Unfortunately, the digital logic blocks and crystal oscillator must be assembled. The popularity of the new digital wrist watches should help solve this problem in due time. The crystal clock operates near ½ MHz and is divided down to seconds and minutes to control the tape drive motor as shown in Table 3.

Table 3 - Selection of Record Duration Time & Repetition Rate

Repeat Time Interval (minutes)	Record Duration Time (seconds)
0.5	15/16
1	1 7/8
2	3 3/4
4	7 1/2
8	15
16	30

From Table 3 one might choose a micro-sample duration time of 3 3/4 seconds repeated every 2 minutes or 7 1/2 seconds duration repeated every 4 minutes. In either mode the recorder could be operated unattended for 24 hours. The total recorded tape for the 24 hours would be 45 minutes (C-90 cassette). Four "C" size flashlight cells are adequate power for 24 hours of operation. A standard 6 volt dry cell lantern battery will power the system for one month.

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SOME HEARING DAMAGE RISK CRITERIA AND THEIR MEASUREMENT

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INTRODUCTION

In recent years substantial research has been conducted in an effort to determine the properties of noise which are hazardous to human hearing and to develop criteria for specifying the degree of hazard. While general principles have been established by this research, no single set of criteria has been universally accepted.

This paper reviews several criteria which have been suggested and examines them from the standpoint of an instrument manufacturer concerned with supplying practical instruments.

Pertinent factors examined include degree of conformance to a particular criteria, hardware complexity, and utilization of measurement results. Particular emphasis is given to instrumentation to meet the damage risk criteria adopted by the Occupational Safety and Health Administration and the Coal Mine Health and Safety Act. The need for standardization of performance specifications and performance evaluation will be discussed for this class of instrument.

FACTORS CONTRIBUTING TO NOISE-INDUCED HEARING LOSS

The risk of noise-induced hearing loss is a function of the following factors:

1. The level or intensity of the noise
2. The frequency content of the noise
3. The length of exposure
4. Individual susceptibility to hearing damage.

Most researchers also agree that interruptions or periods of relative quiet during an exposure period (sometimes called intermittency) are beneficial and, all other things being equal, reduce the risk of hearing loss.

Unfortunately, it is very difficult to establish a criterion based upon substantial experimental evidence. Many years of exposure to the hazardous levels normally encountered are usually necessary before an individual experiences a measurable noise-induced hearing loss.

Also, those jobs most likely to expose a worker to noise-induced hearing loss have a very high personnel turnover. For example, in the textile industry 100-300% turnover in one year is not unusual. Consequently, establishing and maintaining a controlled sample is very difficult.

Exposure to unmeasured and uncontrolled off-the-job noise of such as gasoline-powered tools and recreational vehicles, discotheques, etc., introduces additional uncertainties into the research results.

Differing susceptibility of individuals to hearing loss when exposed to identical conditions requires that relatively large samples of a population be used in order that results be representative.

This difficulty of establishing unequivocal experimental evidence has resulted in the use of several different criteria for estimating the risk of noise-induced hearing loss.

REVIEW OF CRITERIA

Frequencies between 250 Hz and 4 kHz are of primary importance to the understanding of speech, and damage risk is evaluated in terms of one's ability to understand speech. Most hearing damage risk criteria accept "A" weighting of the noise as providing an adequate measure of the importance of various segments of the frequency spectrum in contributing to noise-induced hearing loss. More elaborate criteria have also been developed based upon octave and 1/3-octave-band levels.

Equal Energy

This criterion is recommended by the International Standards Organization in ISO Document 1999, "Assessment of Occupational Noise Exposure for Hearing Conservation Purposes." It reasons that the hazard to hearing is determined by the total noise energy to which an individual is exposed in any given 40-hour period. Implicit, of course, is the assumption that one week's exposure is typical of that to which the worker will be exposed for many years. This is expressed in the form of an equivalent continuous A-weighted level calculated from the varying levels actually experienced. The equal energy aspect comes from the fact that levels having a 3-dB difference in their A-weighted value and which also differ in duration by 2:1 have equal contributions to the equivalent level (i.e., they are equally hazardous). The ISO Recommendation covers a range of 80 dBA to 120 dBA, but leaves to the user the choice of a specific damage risk limit. The ISO procedure involves classifying measured noise levels into each of nine bands and the selection of an index number based upon the total time accumulated in that band during the 40-hour interval. All index numbers are summed and an equivalent continuous level calculated from this sum. The procedure is shown in Tables I and II.

Equal Temporary Effect

A number of criteria exist which give more weight to the beneficial effects of intermittency than is allowed for by the equal energy rule. These criteria are based upon controlled studies of temporary threshold shift. These criteria postulate that hearing damage risk is directly related to the temporary threshold shift induced by the noise exposure. They are supported by studies which show that noise exposures which produce permanent hearing loss also produce temporary loss in normal ears and, conversely, that noise exposures which don't produce temporary shift also don't cause permanent loss.

CHABA Damage Risk Contours

Working Group 46 of the National Research Council - Committee of the National Academy of Science on Hearing Bioacoustics and Biomechanics (known as CHABA) issued a report in the Journal of the Acoustical Society of America, March 1966, describing a criterion for hazardous exposure to both intermittent and steady-state noises.

They generated a set of damage-risk contours for octave and 1/3-octave bands of noise, for pure tones, and for interrupted exposures to bands of noise. These curves are based upon the assumption that an individual's exposure continues almost every day for a period of at least 10 years. These are shown in Figures 1 and 2.

CHABA Table

Standing Committee - SC-7 of CHABA derived from this set of contours a recommended hazard criterion for use in coal mines. They made the basic simplifying assumption that A-weighting is an adequate single number measure of the differing hazard expected from different frequencies. These are shown in Table III.

As can be seen from the Table, the more times the noise occurs per day, the higher the allowable dBA level for a given total duration. Notice also that for a single occurrence per day, it follows the equal energy criterion. Thus, this criterion incorporates the primary thesis of both the equal energy and the equal temporary effect criteria.

ACGIH

The American Conference of Governmental Industrial Hygienists proposed a further simplification of the CHABA Table which was adopted by the U. S. Department of Labor. These are the well-known Walsh-Healey threshold limit values. The applicability of the W-H Public Contracts Act was expanded under the Occupational Safety and Health Act to all industries engaged in interstate commerce and also adopted for the protection of coal miners under the Coal Mine Health and Safety Act. These familiar values are shown in Table IV.

A distribution of noise during the day equivalent to an average of 7 occurrences per day is implicitly assumed. The equal energy criterion is thus modified to allow for the reduction in hazard of the assumed typical intermittency pattern and is simple enough to use in many practical monitoring applications.

INSTRUMENTATION

As instrument manufacturers we leave the question of which criteria or theory is "best" to the experts. However, the "best" criteria from the standpoint of accurate estimation of hearing damage risk may require elaborate instrumentation for its measurement.

Thus the instrument maker's ability to provide practical tools for the job must be considered when selecting criteria for practical use.

From this standpoint the needs may be divided into two major categories: research instruments to help establish and validate theories of damage risk and monitoring instruments to assist in determining compliance with a specific damage risk criteria.

Research Instruments

Often researchers use general-purpose instruments in their work for the simple reason that they are readily available. A sound-level meter, octave- or 1/3-octave-band analyzer, a stop watch, and an audiometer can make the measurements necessary for evaluating conformance to any existing damage risk criterion and in the development of any new ones. Also, ANSI and ISO specifications are available which establish acceptal performance levels and evaluation methods.

Collection of the large amounts of research data required to validate theory sometimes makes it desirable to have special-purpose instruments to speed and simplify the data collection task. James Botsford, a leading noise control engineer and a proponent of TTS as an indicator of hearing damage risk, has used experimental instruments small enough to be worn without discomfort to measure and display the TTS of the wearer at the end of the workday. This device eliminated the time and skill required for daily audiometric testing to determine TTS.

A block diagram of the experimental instrument is shown in Figure 3. Its operation has been discussed in previous papers[1] and will not be repeated here.

Suffice it to say that it uses A-weighting rather than providing data in each function of frequency as does an audiometer and it does not measure true TTS but only that component of TTS which from a damage risk standpoint appears to be significant. Figure 4 shows the prototypes used in some of Botsford's research.

Compliance Instruments

General-purpose sound-level meters are now in widespread use for compliance measurements to the criteria used by OSHA and CMHSA. In applications where noise levels are much higher than threshold limit values or where the noise is steady throughout the day, sound-level meters do a satisfactory job.

However, many plants and most mine environments are characterized by a highly variable distribution of noise levels and durations. For example, the beneficial effects of intermittency described by Sataloff[2] were based upon studies done in iron mines.

Under such conditions the use of sound-level meters and stop watches by compliance officers is difficult at best. Since these officers are responsible for many aspects of safety in addition to hearing damage risk, it becomes a strain to provide them with the time and skills necessary to make accurate calculations at a variety of work stations or to attempt to follow a workman throughout the day to obtain his exposure as called for by the threshold limit values.

In response to this need, a number of manufacturers have introduced commercial instruments aimed at automatically measuring and calculating noise exposure in accordance with the criterion adopted by OSHA and CMHSA. Availability of these instruments is a significant step forward in helping both industrial safety personnel and government compliance officers determine the existence of noise hazards to individual workers and, consequently, in speeding the application of remedial measures. Here the criterion established is a compromise to adapt the "ideal" of CHABA Working Group 46 to the practical realities of making measurements in an industrial environment. The simplifications in the criteria also make it possible to produce instruments small enough to be worn which also automatically produce the desired exposure data. Even with these compromises with "ideal" criteria, it is only recently that technology has made possible wearable instruments specifically designed for automatic measurement of the criterion.

Although these new instruments offer substantial advantages in some applications, it is important that those responsible for their acquisition and use understand them and appreciate how they differ from conventional instrumentation such as sound-level meters or audiometers.

A block diagram of a typical instrument is shown in Figure 5. Since it accumulates A-weighted sound level as a function of time, it can be thought of as an integrating sound-level meter. In the absence of a specific standard on such an instrument, those evaluating them borrow heavily from the current ANSI Specification for Sound-Level Meters S1.4-1971.

The instruments can, in fact, be divided into two sections, one similar to the front end of a standard sound-level meter and one unique to the integrating and display functions required to make measurements in accordance with the criterion.

Applicable sound-level meter standards and performance evaluation methods might include:

- microphone omnidirectionality and frequency response
- frequency response of weighting networks
- detection characteristics (rms-slow)
- crest factor capacity
- internal noise and distortion
- sensitivity to environment (temperature, humidity, vibration, magnetic fields, etc.).

Some current sound-level meter requirements such as those describing indicating instrument characteristics are not applicable. It may or may not be desirable to establish several classes of performance level.

New standards and performance evaluation methods must be developed which are peculiar to the integrating function of these instruments. Included are the integration system, the data display, and the effect of the wearer.

Integration System

The integration system has at least three specific properties which might be specified and tested:

1. The degree of conformance to the specified accummulation rate with time at a fixed dBA level (i.e., does doubling the exposure time double the accumulated exposure level indication?).
2. The degree of conformance to the specified accumulation rate with level for a fixed time (i.e., how closely does it follow the rule that a 5-dB increase in dBA level will double the accumulated exposure level indication obtained in a given time interval?).
3. Conformance to the end point specifications (i.e., does the integration stop below the level at which the criterion places no limit on the allowable exposure time and does it indicate when the maximum allowable dBA level has been exceeded?).

Display

The type of display used and its resolution should be described.

Effect of the Wearer

Current practice for noise exposure measurement specifies that measurements be made at the position of the exposed person without the person present. This presents obvious difficulties in establishing the performance of a device specifically intended to make its measurements while being worn.

One approach is to specify and measure the instrument's performance under free-field conditions as is done with sound-level meters. This has the advantage of using well established techniques and test conditions and should provide good agreement in measurement results among different groups.

Applicable tests include relative response level as a function of angle of incidence with respect to random-incidence relative response and the random-incidence frequency response of the instrument.

In addition, since the instruments are usually worn in semi-reverberant spaces such as industrial plants, mines, etc., they should also be tested in a diffuse field when worn by an "average" individual. The greater uncertainty of this method may make standard tolerances difficult to establish, but the test results are needed for intelligent application of the units in many practical cases.

Microphone location under these test conditions may significantly affect the results and may also need to be specified and/or reported.

Other factors of significance in practical application of a dosimeter which may or may not be deemed appropriate for inclusion in a standard but which do affect its practical utility as a tool for gathering data on noise exposure are:

size and weight
susceptibility to tampering
ease of calibration
access to measurement data.

Because of the interest in rapid field deployment of dosimeters of integrating sound-level meters, a number of government agencies are evaluating or planning to evaluate commercially available instruments with the objective of establishing purchase and maintenance criteria. It would be most unfortunate if these groups, operating independently, generated conflicting criteria or imposed conflicting requirements upon manufacturers.

It would be equally unfortunate if the requirements limited the ingenuity of instrument designers by presuming all instruments have a similar block diagram and imposing test methods or performance specifications which restrict designs to such a diagram. This seems to be a real danger if each of those establishing performance standards starts from what is now available (S1.4-1971) and makes his own extrapolation.

The solution to this potential dilemma is to get an ANSI Standard written for this new class of instrument. Working Group S1-45 on Sound-Level Meters and their Calibration has begun work on a standard for an integrating sound-level meter which encompasses (but is not limited to) instruments for evaluation of hearing damage risk by either the equal energy or the OSHA criteria.

Typically, generation of an approved standard requires four to five years from inception to final approval. The large potential usage of these devices for compliance measurements under OSHA and under CMHSA makes it imperative that this typical time interval be drastically reduced. All those interested in the use of these devices should encourage the working group to promulgate at least a draft standard for use by government and industry by the end of this year.

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1 J. H. Botsford and B. R. Laks, "Noise Hazard Meter," ASA Paper No. 6A9 (November 1969).
L. M. Moser, "Noise Hazard Meter Based on Temporary Threshold Shift," ASA Paper No. V2 (April 1970).

2. J. Sataloff, L. Vassallo, H. Menduke, "Hearing Loss from Exposure to Interrupted Noise," Archives of Environmental Health June 1969, Volume 18.

TABLE I – Partial noise exposure indices for sound levels 80 to 120 dB (A) and duration 10 minutes to 40 hours per week

Duration per week		Partial noise exposure indices								
		Sound level in dB (A) (Class midpoint)								
hours	minutes	80	85	90	95	100	105	110	115	120
	10					5	15	40	130	415
	12					5	15	50	160	500
	14					5	20	60	185	585
	16					5	20	65	210	665
	18					10	25	75	235	750
	20					10	25	85	265	835
	25				5	10	35	105	330	1 040
	30				5	15	40	125	395	1 250
	40				5	15	55	165	525	1 670
	50				5	20	70	210	660	2 080
	60			5	10	25	80	250	790	2 500
	70			5	10	30	90	290	920	2 920
	80			5	10	35	105	330	1 050	3 330
	90			5	10	40	120	375	1 190	3 750
	100			5	15	40	130	415	1 320	4 170
2				5	15	50	160	500	1 580	5 000
2.5				5	20	65	200	625	1 980	6 250
3				10	25	75	235	750	2 370	7 500
3.5			5	10	30	90	275	875	2 770	8 750
4			5	10	30	100	315	1 000	3 160	10 000
5			5	15	40	125	395	1 250	3 950	12 500
6			5	15	45	150	475	1 500	4 740	15 000
7			5	20	55	175	555	1 750	5 530	17 500
8			5	20	65	200	630	2 000	6 320	20 000
9			5	25	70	225	710	2 250	7 110	22 500
10		5	10	25	80	250	790	2 500	7 910	25 000
12		5	10	30	95	300	950	3 000	9 490	30 000
14		5	10	35	110	350	1 110	3 500	11 100	
16		5	15	40	125	400	1 260	4 000	12 600	
18		5	15	45	140	450	1 420	4 500	14 200	
20		5	15	50	160	500	1 580	5 000	15 800	
25		5	20	65	200	625	1 980	6 250	19 800	
30		10	25	75	235	750	2 370	7 500	23 700	
35		10	30	90	275	875	2 770	8 750	27 700	
40		10	30	100	315	1 000	3 160	10 000	31 600	

The values are calculated from the formula :

$$E_i = \frac{\Delta t_i}{40}\, 10^{0.1\,(L_i - 70)}$$

where

E_i is the partial noise exposure index;

L_i is the sound level A in dB corresponding to the midpoint of the class i;

Δt_i is the total duration in hours per week of sound levels within the class i.

TABLE II – Relation between composite noise exposure index and equivalent continuous sound level

Composite noise exposure index	Equivalent continuous sound level, dB (A)
10	80
15	82
20	83
25	84
30	85
40	86
50	87
60	88
80	89
100	90
125	91
160	92
200	93
250	94
315	95
400	96
500	97
630	98
800	99
1 000	100
1 250	101
1 600	102
2 000	103
2 500	104
3 150	105
4 000	106
5 000	107
6 300	108
8 000	109
10 000	110
12 500	111
16 000	112
20 000	113
25 000	114
31 500	115

The values are calculated from the formula

$$L_{eq} = 70 + 10 \log_{10} \Sigma E_i$$

where

L_{eq} is the equivalent continuous sound level in dB (A);

E_i is the partial noise exposure index (from Table I).

Table III – Permissible Average Noise Level in dBA

Total exposure	Number of noise-burst exposures per 8-hour workday: 1	3	7	15	35	75	150 or more
8 hours.........	90						
6 hours.........	91	92	93	94	94	94	94
4 hours.........	93	94	95	96	98	99	100
2 hours.........	96	98	100	103	106	109	112
1 hour..........	99	102	105	109	114	(115)	
30 minutes....	102	106	110	114	(115)		
15 minutes....	105	110	(115)				
8 minutes.....	108	(115)					
4 minutes.....	111	(115)					

Extrapolation between points in this table is permissible.

Table IV – Maximum Recommended Exposure to Noise in dBA for Eight Hours and less.

Daily Exposure Time (Hours)	Sound Level (dBA)
8	90
4	95
2	100
1	105
½	110
¼ or less	115

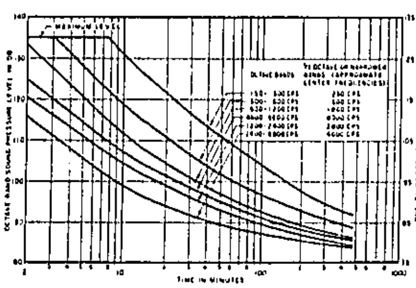

Fig. 1. Damage risk contours for one exposure per day to certain 1-oct (left hand ordinate) and certain 1/3 oct or narrower (right hand ordinate) bands of noise. This graph can be applied to the individual band levels present in broad band noise.

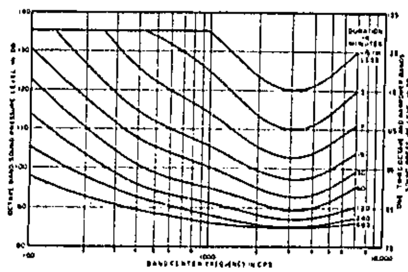

Fig. 2. Damage-risk contours for one exposure per day to 1 oct (left-hand ordinate) and 1/3 oct or narrower (right-hand ordinate) bands of noise. The graph can be applied to the individual band levels present in broad band noise.

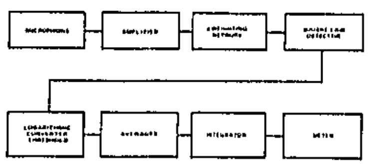

FIGURE 3. BLOCK DIAGRAM OF NOISE HAZARD METER

(FIGURE 4. NOT AVAILABLE)

FIGURE 5. BLOCK DIAGRAM OF INTEGRATING SOUND LEVEL METER

DISCRETE SOURCE IDENTIFICATION IN THE PRESENCE OF HIGH AMBIENT NOISE LEVELS

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Large horsepower internal combustion engines make noise; a fact that is not likely to surprise anyone. With the introduction of the Walsh-Healey legislation and more recently the Occupational Safety and Health Act, a considerable amount of data has been collected by both industry and governmental bodies documenting the amount of noise that is produced by internal combustion engines. The literature on the control of noise from these sources published thus far, however, seems to be limited to the lower horsepower units (i.e., under 1000 HP). The larger units (1000 to 12,000 and more HP) have, for the most part, been ignored in the published data. Two factors have had a significant influence on this lack of noise control information. One factor, to put it simply, is that large horsepower units are in fact large. A 3400 HP engine compressor unit such as the one that will be used later as an example, has major dimensions on the order of twenty-four feet by twenty feet by twelve feet. With a weight in excess of 100 tons, these units are not likely to be picked up and placed in an anechoic chamber.....even if you could find one large enough to accommodate such a unit. The relative immobility of these units brings up the second factor, which is their typical location. In most cases, where you find one of these units, you find more. It is common practice to have several of the units housed in a building with a separation of less than one major dimension between units. In fact, it is uncommon to find one of these units housed in a building where it is possible to get one major dimension away from the unit.....and still stay within the building. Thus we have one or more noise generators located in a space where little exists other than the commonly described near field. It is perhaps an understatement to say that given these conditions, meaningful data on the major individual noise sources on these units is difficult. Such data is, however, vital in determining the best possible approach to the control of the noise sources if, in fact, there is a feasible approach.

In an attempt to overcome the difficulties encountered in locating discrete noise sources in the presence of high ambient noise levels, several approaches were examined. Methods involving the use of vibration accelerometers were avoided due to the temperature sensitivity problems present with most of these devices. The techniques used in the past to plot the so called sound level contours were considered to be of some value, but they did not allow one to positively identify a particular source. In the end, a rather simplistic approach to the entire problem was taken.

In theory, it seemed logical that since what was needed was a method to isolate each individual source from all others, it should be possible to construct some sort of an isolator. This idea lead to the construction of the device shown schematically in Figure I. The tube, for want of a better name, was a section of dense plastic material lined with a lead/foam sandwich. Provision was made to mount a survey type meter at the closed end of the tube. The use of the survey meter rather than a more accurate form of microphone was felt justified in that it allowed a convenient check of the noise levels while positioning the tube. The fact that the length of the tube was something under three feet and thus would almost always be in the near field seemed of little consequence since almost anywhere within the buildings which house these units is in the near field anyway. The existence of the resonance and attenuation effects of the tube on the noise levels being measured was felt to be of major importance, but reconcilable due to the fact that most if not all of the sources would be altered in the same way. That is to say, one of the basic assumptions was that the levels and spectra obtained through the use of the tube would be different in some way from these actually being radiated by the source.

At first glance this assumption seemed to negate the value of the technique, but further study indicated that such was not the case. In that most, if not all, of the sources encountered were of rather heavy construction, the rubber gasket seal had a very small effect on the vibration of the source. In other words, the impedance change was virtually the same in all cases. Thus, while the data could not be considered to be accurate in an absolute sense, it did represent a good indication of the relative contribution of each source to the whole.

In order that some correlation of the tube data could be made with the actual levels and spectra emitted from the various sources on our engines, a series of tests were conducted at a site where only one unit was present in a building. The particular unit involved was a 3400 HP Cooper-Bessemer GMWH-10, two views of which are shown in Figures II and III. The engine room itself has a volume of approximately 2000 M^3 and a surface area of approximately 1200 M^2, of which approximately 335 M^2 has been treated with a 4" fiberglass layer covered with perforated aluminum. Due to a recent equipment problem it has not been possible to measure the reverberation time for this room. However, a rough check indicates close agreement with values obtained using the manufacturer's published data and the modified form of the Sabine equation as given by Beranek[1]. Using the calculated reverberation times and the space average mean-square sound pressure level, the sound power level for each of the nine preferred octave bands was calculated using:

$$L_W = \overline{L}p + 10 \log V - 10 \log T_{60} + 10 \log \left(1 + \frac{S\lambda}{8V}\right) - 13.5 \text{ dB}$$

where V = total volume of engine room (with volume of engine subtracted), M^3

T_{60} = calculated reverberation time of engine room, sec.

λ = wave length of the sound at the geometric mean frequency of the test band, M

S = area of all boundary surfaces in the room, M^2

$\overline{L}p$ = mean-square sound pressure level in test band, dB

The resulting spectrum is shown on Figure IV. Measurements were then taken at the surfaces of the various engine components. The sound pressure levels were recorded and converted to sound power levels. Table I shows the power levels obtained for the various components using:

$$L_W = \overline{L}p + 10 \log S \text{ dB}$$

where Lp = average sound pressure level in each band measured with the tube sealed to the source.

S = surface area of the source, M^2

The spectrum obtained by summing the contribution of all of the components is shown as the dashed line on Figure IV. As can be seen, the magnitude of the spectrum obtained by this summation is somewhat higher than that determined from the measurements without the tube. The general shape of the spectrum is, however, very similar.

It should be noted at this point that a correction factor was applied to the readings obtained from the tube. Several approaches were taken in order to account for the effect of the tube itself on the radiated noise. The method with which the best agreement of data was achieved involved a simple substitution. A space volume of approximately two meters on a side was found in the building where the measured sound pressure level was constant within one dB in each octave band. The sound level meter was first placed in the center of this volume and the output signal was measured in each octave band. The tube was then connected to the meter and another set of readings was taken. The difference between the two sets of readings was used as a correction factor for the tube.

While the validity of the data obtained thus far is certainly questionable from a rigorous technical standpoint, the general trend of the data produced is encouraging. In the raw sound pressure levels measured, a difference of more than ten decibels was obtained in reading on the fuel intake line and the exhaust manifold.....even though the meter location was virtually the same in each case while only the orientation of the tube was changed. This difference is taken to be an indication that a fair degree of isolation is achieved through use of the tube. A further indication of the discrete source identification possible with this technique can be seen in Table I. In almost all bands, the summed level can be seen to be controlled by four specific sources. Namely: the exhaust manifold, the rocker arm covers, the power cylinders and the air intake manifolds. This result is in agreement with commonly held theories about the noise radiated from these units. There appears to be, however, a significant difference in the relative contribution of the engine block itself. Using the levels suggested by Miller[2] we find that the power levels we obtained are below these predicted by some ten to twenty decibels. This factor, in conjunction with the fact that our summed power levels were greater than ambient tends to indicate that the block itself is not as great a contributor to the overall engine room noise as has been assumed.

In summary, we feel that a practical technique for the identification of the relative contribution of each of the many component noise sources on large internal combustion engines may be developed using the approach described. Much further work will, however, be necessary to verify and refine the technique. At the present time further research is being conducted to correlate the vibration levels of the various components to the measured sound pressure levels. An experimental measurement of the total sound power level as described by Beranek[3] is planned to be completed by the time of this presentation. In addition, the application of real time analysis to the technique is being investigated. Work to date has indicated some significant advantages to the use of a real time analysis. One advantage found thus far has been in the reduction of measurement and analysis time. By using the exponential averaging option available on most of these instruments, the same accuracy available with a conventional third octave analysis can be obtained in approximately one tenth of the time. Such a reduction in analysis time is certainly very important when a large number of sources such as found on a typical engine are to be examined. Another possible, though unproven at this point, advantage is in the ability to synchronize the analysis of a component noise spectrum to the rotational speed of the engine being examined. This quasi cross-correlation would be one further step in the isolation and thus identification of discrete noise sources in the presence of high ambient noise levels.

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2. L. N. Miller, "Noise Control for Reciprocating and Turbine Engines Driven by Natural Gas and Liquid Fuel," American Gas Association, Inc., 1969, p. 66.

3. L. L. Beranek, ed., Noise and Vibration Control, McGraw-Hill, 1971, pp. 149-150.

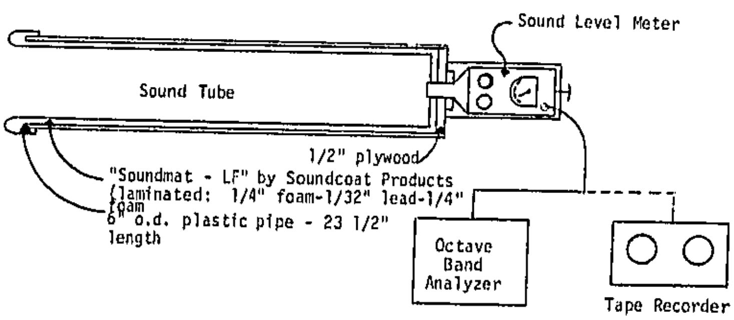


Figure I

"Tube"

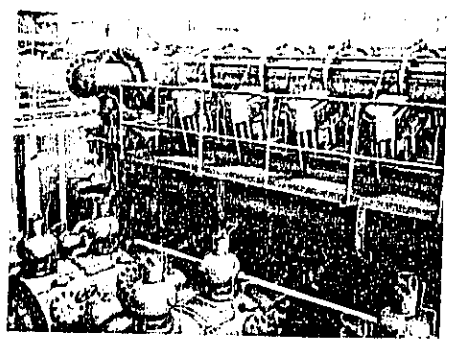


Figure II

3400 HP natural gas fueled engine compressor unit. Right to left at top - exhaust manifold and power cylinders and turbocharger. Partial view of compressor cylinders in left foreground.

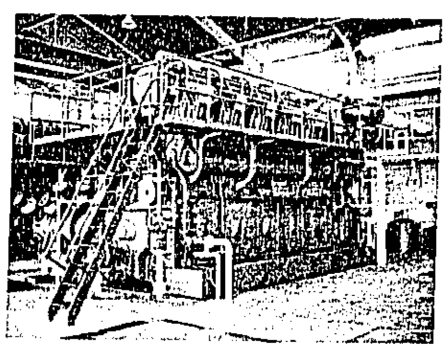
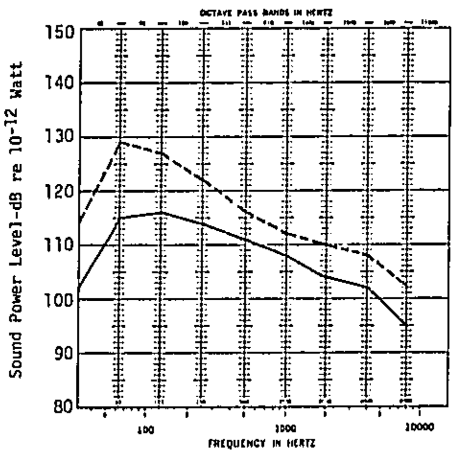


Figure III

Same unit viewed from the other side. Note intake air manifold in center and air intake line at upper right.



---- Summation of Individual Sources

—— Space Average

Figure IV

Relative Sound Power Levels
dB re 10^{-12} Watt

Source	Octave Band Center Frequency Hz								
	31.5	63	125	250	500	1000	2000	4000	8000
End Block	86	91	95	107	104	100	90	79	74
Intake Air Manifold (Rt.)	97	110	108	114	109	105	99	97	80
Oil Pump	81	91	89	85	79	81	77	70	69
Compressor Cylinder	103	113	113	109	104	100	96	94	85
Camshaft Cover (Rt.)	89	94	95	93	89	88	85	80	71
Camshaft Cover (Lt.)	84	94	99	96	93	91	86	78	71
Exhaust Manifold	111	128	126	109	106	105	103	99	101
Cylinder Head	90	95	95	103	100	94	88	81	78
Rocker Arm Cover	97	102	115	117	107	99	92	84	85
Turbocharger Housing	93	104	102	96	91	90	99	96	85
Air Intake Line	103	117	114	106	99	103	106	105	93
Power Cylinder	100	112	113	109	105	102	99	95	88
Fuel Gas Manifold	102	109	103	97	89	87	93	86	82
Crank Case Doors (Lt.)	97	105	104	110	104	95	87	83	77
Intake Air Manifold	103	114	112	116	111	106	101	99	78
Side Block	97	104	100	102	96	91	86	85	79
Lubricator Pumps	90	107	106	100	93	88	83	81	72
Aftercooler	91	110	110	102	100	95	97	93	74
Oil Pump	92	83	79	81	78	76	72	73	68
Exhaust Line	96	111	110	104	101	97	95	91	80
Flywheel Cover	95	100	102	100	92	88	85	94	77
Water Pump	87	102	98	95	91	94	93	86	79
Crank Case Doors (Rt.)	93	100	99	104	102	97	91	84	77

Table I

THE CONSTRUCTIVE ANALYSIS OF NOISE

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Convectional spectral analysis is not completely applicable to the investigation of noise produced by machines, transportation vehicles, etc. Due to the variability and complexity of the spectra the results of spectral analysis do not give, in general, any typical, stationary frequency-amplitude representation, fit for noise classification and noise pattern recognition.

In the Department of Physics in the Leningrad Shipbuilding Institute a new method of spectral analysis, the so called "constructive analysis" method has been developed, which may allow a solution to this problem. This method has received verification in many investigations of noise produced by different machinery noise sources. These investigations were undertaken by A. P. Gromov, O. P. Murylev, H. N. Miasnikova, V. O. Serajutdinov, B. A. Finagin and others [1,2].

The constructive spectrum to a considerable extent is defined as a sequence of segments, which are marked by appropriate features. Each segment is stationary within reasonable limits and possesses some definite patterns which are retained during an interval of time. The noise realization is represented by statistics of noise segments. The constructive spectrum is a histogram of segments.

When some spectral realization represented by a function $\psi(\nu, a)$, which depends on the frequency ν and the amplitude a is given, it may be written

$$\psi = c_1\varphi_1(\nu,a) + c_2\varphi_2(\nu,a) + c_3\varphi_3(\nu,a) + \ldots\ldots + c_n\varphi_n(\nu,a).$$

Here $c_1, c_2, c_3 \ldots$ are the weighting factors of the segments and $\varphi_1(\nu,a)$, $\varphi_2(\nu,a)$, $\varphi_3(\nu,a)$,....are the functions, which describe the segments. Owing to the ergodic property of noise, these weighting factors are connected with the frequency of the appearance of the different segments in the course of time--i.e. with the probability density of the segments. It is supposed that for each segment the probability density is known.

The histogram shows how the probability density depends on the segment type; it is called the "constructive spectrum".

The constructive analysis is based on the segment recognition process; the definition of a segment type gives some random conditions, the possibilities of a segment prediction depends upon the choice of the segment alphabet. Usually the segments are distinguished by the form of the spectral envelope of a section of variable spectrum flow, (i.e. of an average spectral presentation for a quantization time). The segment envelope form differs by the quantity of maxima, by the steepness, by the presence of a flat top etc. Practically, the segment alphabet is formed by a dozen segments, which differ by the form of the spectral envelope.

The judgment about the segment type is to be made during the time of quantization. This prediction can be made in an automatic way; the task is to find a standard with a spectral envelope form nearest to the spectrum envelope of the signal. It is necessary to "guess" the noise pattern, to attribute the signal to a definite segment type. These types are marked by symbols: for this purpose some letters may be used. If, for instance, the envelope has 2 maxima, the segment is called an M-segment, if it has one maximum it is called an I-segment, if it has a flat top it is called an Π -segment, and so on.

The linear segmentation can be replaced by a matrix one. In this case a frequency plane (with two rectangular, frequency coordinate axes) may be used. Then a segment appears as a figure on this plane and the intensity of the image depends on the amplitude.

Neither human observers or automatic devices are able to find the segment pattern quite exactly; it is only possible to make an approximate determination of the segment type. The result of segmentation (done by an observer or by a computer) is a histogram of segments, which shows the distribution of segment probability density P(x), where x designates the segment symbol.

The experimental definition of the constructive spectrum is realized in the following way. The measuring device consists of an analyser with a set of parallel band pass filters connected to a microphone or tape-recorder input; the output of the analyser forms a matrix. Further, the equipment consists of a segment selector, as a translator for an analog-code input to a computer which is used to construct the segment histogram (and other operations, including the automatic recognition of the histograms).

When a noise signal is put into an analyser, the latter gives some spectral sequences. In this spectral flow, divided into equal time intervals, corresponding to the quantization time, the selector distinguishes different segments, which are coded and put into the computer etc. The order of sequences is all the same, the segments may be repeated.

In our investigations a spectral pattern recognizer (a "Sceptron") was used. The segments were distinguished by sceptrongrams; these figures were transformed into an electric current distribution by means of a photo-electronic mosaic.

The series of segments (in each sequence they may be repeated) give the possibilities to plot a histogram.

The method of constructive analysis was applied to the investigations of machine noise and the noise from air turbines. A vocabulary of segments was developed, which was used for the automatic treatment of the segment flow. The method was applied to acoustic machine diagnosis.

During the investigations of the turbines it was found that the constructive spectra depended on blade inclination and axial clearance. The histogram classification may be used for the definition of turbine regimes and allows recognition of the noise (with a probability of not less than 90%).

Besides the automatic recognition of noise patterns the visualization of noise is also useful. The visualization of noise was realized not by means of a "visible speech" method, but by picking out the most frequent noise segments. As the result of segmentation, a set of pictures on a light tableau was produced, which allowed a judgment about the noise origin.

Constructive analysis can be applied to the analysis of different acoustic signals and to noise from transportation vehicles etc.

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APPLICATION OF THE COHERENCE FUNCTION TO ACOUSTIC NOISE MEASUREMENTS

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SUMMARY

The power of the coherence function technique to compute the frequency spectra and sound pressure levels associated with sources of noise in the presence of other, incoherent sources has been demonstrated. It is a very useful technique for noise survey and noise control measurements. But care must be exercised in its application to avoid erroneous results. Multiple coherences, time lag between the microphone and source transducer signals, and numerical errors due to limited dynamic range of the analysis instrumentation are some of the problems which may be encountered.

INTRODUCTION

In this paper the feasibility of using the coherence function to help locate sources of noise will be investigated. The coherence function determines the degree of coherence between two time signals. It has a value of one if the signals are coherent and a value of zero if they are completely incoherent. If one signal is chosen as the source then the coherence function can be used to determine what percentage of the output signal is coherent with the source, which is a measure of how much of the output is due to the source. For application to noise measurements, the output signal is the sound pressure at some location and the source signal is any representative measure of source motion.

Development of the mathematical background of the coherence function will be followed by a discussion of application techniques and presentation of test results.

BACKGROUND

The coherence function is defined as

$$\gamma^2 = \frac{|\overline{G_{yx}}|^2}{G_{xx}\ G_{yy}} \tag{1}$$

where

G_{yx} = cross power spectrum between input and output
G_{xx} = power spectrum of input
G_{yy} = power spectrum of output

It has the property that at any given frequency it is a measure of the power at point y due to an input at point x. Mathematically this is shown in Reference 1. To briefly review the important mathematical concepts, the power spectrum of input and output can be computed from the following:

$$G_{xx} = S_x S_x^* \tag{2}$$

where

S_x = Fourier transform of source signal x(t) (linear spectrum)
S_x^* = complex conjugate of S_x

and, $$G_{yy} = S_y S_y^* \tag{3}$$

where

S_y = Fourier transform of response signal y(t)
S_y^* = complex conjugate of S_y

and $G_{yx} = S_y S_x^*$ (4)

If there are sources other than x then the response at y is

$$S_y = HS_x + S_z$$

where

S_z = Fourier transform of noise signal
H = frequency response between x and y

Therefore substituting into (3)

$$\therefore G_{yy} = (HH^* S_x S_x^* + H^* S_z S_x + HS_x S_z^* + S_z S_z^*)$$

$$G_{yy} = (|H|^2 G_{xx} + H^* G_{zx} + H G_{xz} + G_{zz})$$

If the noise signal is uncorrelated with the input signal x, the cross spectral terms G_{zx} and G_{xz} become equal to zero in the limit,

$$\therefore G_{yy} = |H|^2 G_{xx} + G_{zz}$$

The cross spectral density term becomes

$$G_{yx} = (HS_x + S_z) S_x^*$$

or

$$G_{yx} = HG_{xx} + G_{zx}$$

again in the limit for uncorrelated sources

$$G_{yx} = HG_{xx}$$

Therefore, the coherence function becomes

$$\gamma^2 = \frac{|H|^2 \; G_{xx}}{|H|^2 \; G_{xx} + G_{zz}} = \frac{|H|^2 \; G_{xx}}{G_{yy}}$$

The numerator of the above equation is the power at point y due to the source at point x and the denominator is the total power at point y. Therefore, if the total power G_{yy} is multiplied by the coherence function, the power at point y due to the source at point x is obtained.

It should be noted that the frequency response H used in this presentation would include any transducer gains. Therefore, if the transducer gain were equal to zero at some frequency the coherence would also be equal to zero, when in reality there may be some power transmitted between x and y at that frequency. Also, the above mathematics is based upon linear theory. If the system is non-linear the coherence does not measure the power ratio. In fact in a system with no noise the coherence function can be used as a measure of the linearity of the system.

If the noise source is correlated to the input x, the coherence function likewise does not measure the power transmitted. In that case it is necessary to include multiple coherences. Unfortunately, all sources are not normally known; therefore, multiple coherences can become a serious problem (see Reference 2).

With the advent of the fast Fourier transform algorithm and modern computing equipment it is now practical to compute the coherence function. There are limitations connected with this technology. It is necessary to correct for sample window effects and aliasing errors, for example. There is also a major error connected with the fact that the sample period is limited. It is determined by the sample size and the sampling rates. Due to the limited sample period, if there is a significant time-lag between the source and the output measurements, poor coherence will be computed. This is not true for periodic signals, only random signals.

APPLICATIONS

The basic application of the coherence function to acoustic noise measurements is in determining the spectrum and total sound level associated with a given source in an uncontrolled acoustic environment. For example, in making plant noise surveys it is usually very costly to shut down all other machine operations to measure the noise of one particular machine. And in making noise control measurements it is often impossible to determine the noise components in the total noise spectrum that are associated with a particular machine, especially if the noise is broad-band or if a number of noise sources operate at the same speeds. In most situations such problems can be overcome with the use of the coherence function.

Application of the coherence function to noise measurements involves the use of two measurement systems: a microphone for measuring the total sound field at a given location, and a transducer for monitoring the motion of a source associated with sound radiation. Examples of the latter system are accelerometers, displacement probes, near-field microphones, and strain gages for noise radiated by solid surfaces; and pressure transducers and hot wire anemometers for aerodynamic noise. However, any type of transducer can be used that measures the behavior of a source which is associated with noise production. And, as shown previously, the results of the coherence function techniques do not depend on the transfer function of the particular transducer used, as long as its response is linear and non-zero throughout the frequency range of interest. Of course, the response of the microphone used to measure the total sound field is important and must be known.

There are several problems encountered in some applications of the coherence function which may lead to erroneous results. A problem mentioned earlier was that of multiple coherences. The coherence function cannot discriminate between sound radiators which are excited by the same source, so care must be taken that the sound producing element of interest is completely independent of all other noise sources. Conversely, if the noise producing element of interest is composed of several independent sources care must be taken to insure that the behavior of every source is monitored. For example, in a complex machine it may be necessary to add the signals of a number of transducers on different parts of the machine to accurately compute the noise associated with the whole machine. One problem that arises in theory is locating the source transducer at a nodal point for some frequency component of the source motion. However, in practice this problem does not frequently arise because of the difficulty of locating a transduced directly at a node point.

Another problem associated with the measurement system is the time delay between the microphone signal and the source transducer signal. Because the analysis system gathers discrete time samples of data, it is possible that the sound data of a given sample period may not correspond to the source motion data of the same period, which may result in inaccurate calculation of the coherence function. The microphone position should be chosen so that the time delay of the sound signal is significantly less than the sampling period of the analysis system. The sampling period is a function of the analysis frequency range chosen.

Reliability problems also arise where numerical errors in the coherence function computation process cause inaccuracies in the calculated coherent noise. In practice the coherence function should be examined after each calculation and should be corrected or noted at frequencies where the value exceeds 1.

TEST RESULTS

To demonstrate the performance of the coherence function technique in practice, several tests were performed: 1) a controlled laboratory test; 2) a field test with uncorrelated noise sources; and 3) a field test with correlated noise sources. Figure 1 illustrates the experimental set-up for the controlled laboratory test. The source motion was monitored with an accelerometer mounted on the radiating plate. A microphone was used to measure the total sound, and a loud-speaker was used to provide the background noise. The exciter system was programmed with a periodic random signal, and, of course, the sound radiated by the plate was a function of its response characteristics. The loud-speaker was driven by a white-noise source. Tests were performed with and without the background noise, and the total sound and coherent sound were determined in each case. Figure 2 compares the computed coherent sound spectrum with the total sound spectrum for the test with background noise. The total sound is significantly greater than the coherent sound over nearly the entire frequency range, and the total level is 9 dB higher, providing a fairly severe test of the coherence function. Figure 3 compares the

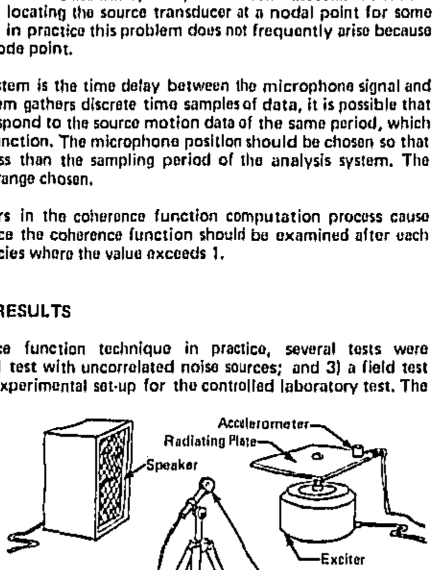

Figure 1
Experimental Set-up for Controlled Laboratory Tests.

computed coherent sound determined in the test with background noise to the total sound of the radiator/exciter system alone. As seen, there is very good correlation between the two spectra, and the total sound level corresponding to each plot is the same. Figure 4 compares the computed coherent sound determined in the test with background noise to that for the test without background noise. Again, the spectra compare very well, and the total sound levels vary by only 1 dB. The discontinuities in the coherent sound spectra are due to numerical errors in the computation of the coherence function. All data presented here result from 100 averages of the coherence functions and corresponding power spectral densities. In tests where the signals were time-averaged 1000 times prior to computation, the numerical errors were very much less.

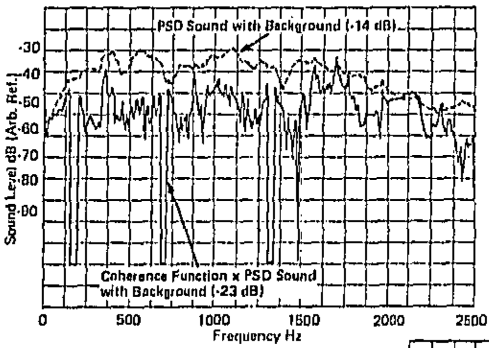

Figure 2
Comparison of coherent sound computed in test with background noise to total sound of radiator/exciter system and background noise source. Total levels are indicated in parentheses.

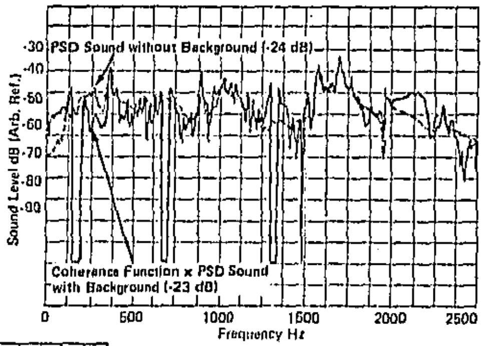

Figure 3
Comparison of coherent sound computed in test with background noise to total sound of radiator/exciter system and background noise source. Total levels are indicated in parentheses.

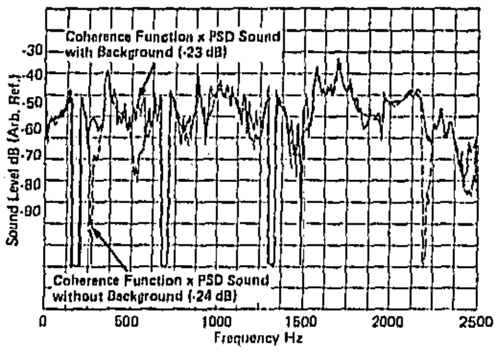

Figure 4
Comparison of coherent sound computed in test with background noise to coherent sound computed in test with radiator/exciter system alone. Total levels are indicated in parentheses.

Figure 5 illustrates the field test conditions for the uncorrelated sources. The source of interest is a motor and blower system, so the sound associated with it consists of discrete frequency components due to structural radiation and broad-band aerodynamic noise from the blower. However, in this test only the structural vibrations were monitored, with an accelerometer mounted on the housing. The background noise was produced by a complex machine which produced both broad-band and discrete frequency noise. Figure 6 compares the computed coherent sound determined with the background noise to the total sound of the blower and background source. It is seen that the coherent sound was completely buried in the total sound, the level of the coherent sound being nearly 20 dB lower than the total sound. Figure 7 compares the computed coherent sound determined with the background noise to the total sound of the motor and blower alone. The difference between the sound levels is partly due to the aerodynamic noise of the fan and partly due to errors in the coherence function. There is also some difference in the levels of the discrete frequency components, but the correlation is good considering the very high level of background noise. Again, more accurate results would be obtained by taking more averages of the coherence function and the power spectral densities.

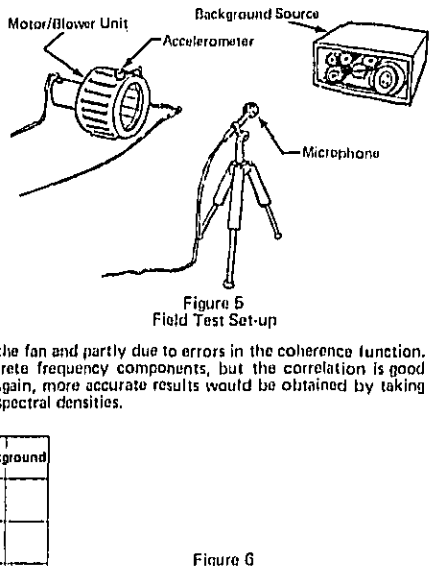

Figure 5
Field Test Set-up

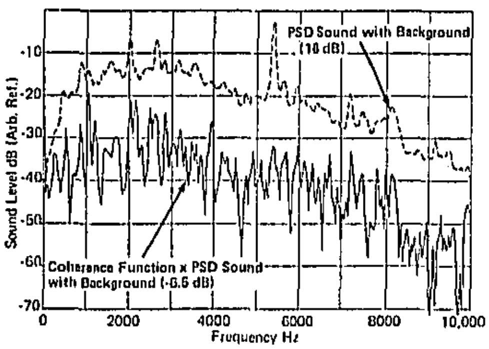

Figure 6
Comparison of coherent sound computed in field test with background noise to total sound of source and background noise. Total levels are indicated in parentheses.

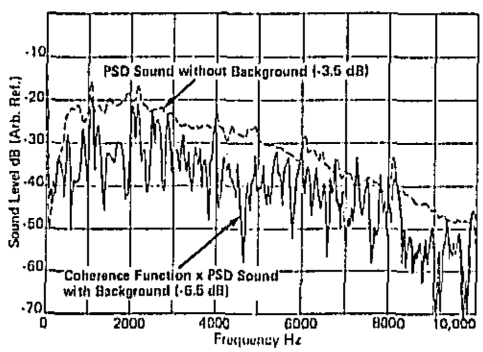

Figure 7
Comparison of coherent sound computed in field test with background noise to total sound of source alone. Total levels are indicated in parentheses.

The final test indicates the errors that may be encountered when attempting to discriminate between two sources that are partially or totally coherent. The data were gathered with a microphone mounted at the side of a standard agricultural tractor and with accelerometers mounted on the various components of the tractor. Table 1 lists the major components and the levels of computed coherent sound associated with each component. Through other tests it was determined that the major source of operating noise was the diesel engine, yet the results of applying the coherence function indicated a level of 98 dBA associated with the power take-off housing. The erroneous results were due to the relatively rigid connections between the various elements, which resulted in multiple coherences.

Table 1
Levels of coherent sound computed for major components of a standard agricultural tractor. The value of the total sound pressure level was 105 dBA.

Tractor Component	Level of Sound Coherent with Component dBA
Oil Pan	99
Cylinder Head	99
Cylinder Block	99
Muffler Shell	99
Valve Cover	98
Intake Manifold	98
Engine Cover	98
Power Take-Off Housing	98
Crankcase	97
Transmission Housing	97
Engine Hood	94
Differential Housing	94
Cab Enclosure	94

Figure 8 is a photograph of the Fourier analysis instrumentation used in this program. It consists basically of an analog-to-digital converter, a keyboard control section, a display unit, and a dedicated mini-computer. The system is easily portable, and may be programmed to automatically average and compute the coherence function and the power spectral densities of the input signals.

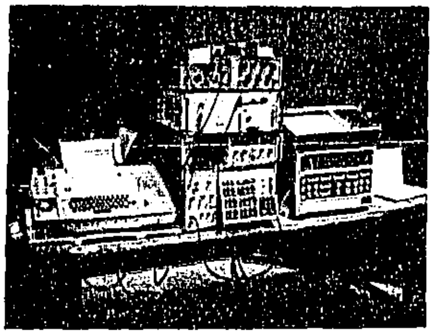

Figure 8
Fourier Analysis System used in Test Programs

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THE ACOUSTICAL FIELD OF TWO SMALL ROOMS

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ABSTRACT

Two small reverberation chambers were built to provide a test of some of the commonly accepted criteria for the design of reverberation chambers. Some notable disagreement was found between the experimental results and the suggested results from the design criteria.

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A program of acoustical studies was initiated at the State University College at Fredonia in the fall of 1970. It was originally planned that these studies would include experimental investigations into problems of noise control and as part of the experimental arrangement, a modest compliment of anechoic and reverberation chambers were to be constructed. Both financial expediancies and available room dictated the use of small chambers, and upon investigation, it was seen that a small anechoic chamber could be built which could satisfy the immediate needs of the experimental program.[1] However, the investigations into the design of the reverberation chamber proved less helpful. Quite apart from the real problem of defining what is meant by a diffuse sound field,[2] which one hopes to experience in the reverberation chamber, there are the questions of how to obtain the diffuse field and how to determine the degree to which a diffuse sound field has been achieved. From a convenient summary of standards,[3] it is readily seen that most reverberation chambers should have a minimum volume of about 3000 cubic feet for use with one-third octave band analysis. There is some disagreement among the standards on the configuration of the room and upon the use of diffusing vanes. With regard to room configuration, Schultz makes the interesting comment that the room modes are largely a function of the volume of the room, independent of the configuration.[4] Also, the absorbtion coefficient of the material coating the interior walls is obviously important with values given in the standards ranging from less than 0.06 to less than 0.1. However, Sioles[5] uses a semiempirical expression that the sound field becomes diffuse above a frequency of 20 c/a (where c is the speed of sound and a is the total absorbtion of the room). This indicates that one should not decrease the value of the absorbtion coefficient to arbitrarily small (as small as can be physically obtained) limits.

Theoretical investigations of the nature of the sound field for non-homogeneous case have required one to make a number of assumptions on the configuration of the room, the location of the source, etc,[6,7,8] so that a theoretical comparison of several configurations is very difficult at this time.

Consequently, it was decided that experimental investigations into the sound fields of some small enclosures would be beneficial. A set of four enclosures, each of approximate volume of one cubic meter and with six square meters of surface area has been planned. The first two enclosures constructed were a pure cube and a perturbed cube, the perturbed cube so called because it retains the general shape of the cube but with no walls parallel. The perturbed cube has volume and surface area within 2% that of the pure cube. Nearly all of the experiments planned for these two rooms have been completed but the studies will be continued into the next set of enclosures, an enclosure having preferred ratios of dimensions[9] and a similar but perturbed enclosure having splayed walls. Again, volume and surface area are one cubic meter and six square meters, respectively, within 2%.

The nature of the sound field was investigated by (1) searching for resonances by sweeping techniques, (2) an analysis of the reverberation time, and (3) measurements of the absorbtion coefficients of "standard" materials. The sweep techniques have shown that the number of resonances and the resonant frequencies change greatly with a change in configuration. This result must be expected for a volume as small as those tested and, in fact, Beranek[10] has shown much earlier that the resonances of a small enclosure were quite sharply defined which would lead one to expect that a change in configuration would change the resonances significantly. Figures 1 and 2 illustrate the modal structures of the cube and the perturbed cube, respectively. Note that the cube has the expected large resonances at 170 Hz and 340 Hz but smaller resonances than expected at 266 Hz and 297 Hz. However, the perturbed cube has only small resonances at these frequencies. Note also that the perturbed cube achieves many more modes in the frequency range from 500 Hz to 1020 Hz. One can assume that the degree of diffuseness of the sound field is related to the close spacing of the room modes. Further, assume that a diffuse field has been reached when at least two modes are contained within the bandwidth of the measuring instrument. In Figure 3, this is done for both the cube and perturbed cube. Note that at all bandwidths the perturbed cube achieves the diffuse field at a lower frequency. If one uses the criteria that a diffuse sound field occurs when the room modes are indistinguishably closely spaced, then, Figures 1 and 2 can give an indication of the lower frequency limits of the diffuse sound field. This result can be compared to the Sioles expression when the total absorbtion is inferred from the reverberation time measurements. The inferred absorbtion for the walls of the enclosures is about .15 (m^2) which would imply a lower limit on the diffuse sound field of 45 KHz. However, the experimental results (the experimental bandwidth was less than 0.3%) show a lower limit more in the neighborhood of 4.5 KHz. Results of the reverberation time measurements using octave bands of noise are given in Table 1. The walls of the chambers were painted with epoxy enamel, the cube receiving two coats which may account for the reduced reverberation time at the higher frequencies for the cube. The uniformity of the results of the reverberation time measurements indicate that at least nothing drastic happens at the lower frequencies where one would expect drastic effects due to the small size and small absorbtion coefficient. The structure of the decay time curves should be further examined by use of one-third octave band analysis. The reverberation time was measured using a technique described earlier.[11] Results of measurements of the absorbtion coefficients of standard materials[12] are unclear as yet due to difficulties in obtaining samples of Sillan. Other materials measured in at least two other large reverberation chambers have been tested with fair agreement above 250 to 500 Hz.

Concluding, it appears that the configuration of a small reverberation chamber is very important, that the Sioles expression may be somewhat high (perhaps by an order of magnitude) and that significantly smaller chambers may indeed be of some value in noise analysis, particularly where wide band measurements are employed.

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Table 1

EXPERIMENTAL DETERMINATION OF REVERBERATION TIME

Frequency (Hz)	Reverberation Time (sec)		Experimental Error (sec)
	Unperturbed Cube	Perturbed Cube	
63	1.14	1.08	$\pm$ .10
125	1.20	1.22	$\pm$ .20
250	1.07	1.09	$\pm$ .20
500	.79	1.16	$\pm$ .05
1,000	.55	1.12	$\pm$ .10
2,000	.86	1.34	$\pm$ .10
4,000	.53	.80	$\pm$ .15
8,000	.39	.76	$\pm$ .20
16,000	.77	1.28	$\pm$ .10

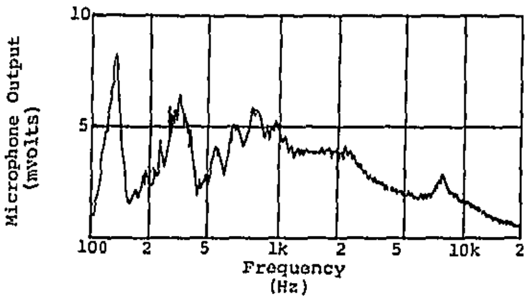

Figure 1. Resonances of the Cube. (Microphone and speaker located in opposite corners)

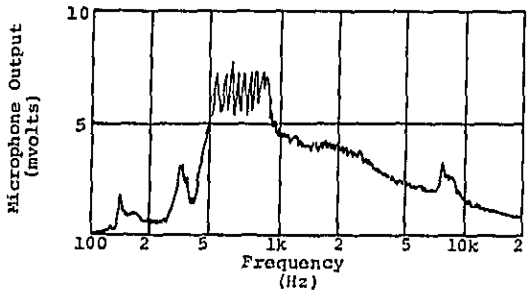

Figure 2. Resonances of the Perturbed Cube. (Microphone and speaker located in opposite corners)

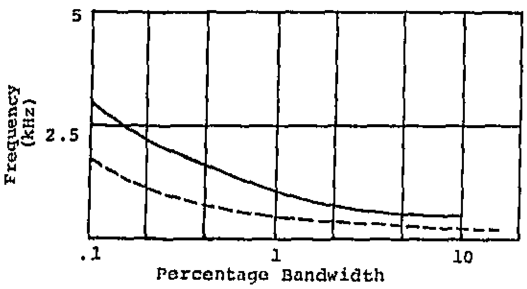

Figure 3. Onset of the Diffuse Field.

THE MEASUREMENT OF SOUND POWER IN A REVERBERATION CHAMBER AT DISCRETE FREQUENCIES

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1. INTRODUCTION

The measurement of sound power emitted by various noise sources is substantial for any kind of noise reduction program. Generally two methods are applied to measure the radiated sound power. The free field or anechoic chamber method consists of placing the source in an environment where the reflection of waves radiated by the sound source is minimized so that the sound pressure of only the radiated wave can be measured over a suitably chosen area around the source. Ideally the complete directivity pattern of the radiation should be known but in practical situation the sound pressure is measured in discrete points and the total sound power is then calculated as a sum of sound power radiated through partial sections into which the total area is subdivided. Several factors as e.g. number and positioning of the subsections, possibility of determination of the sound intensity from the sound pressure can affect and limit the accuracy of this method.

The second method for the sound power measurement consists of placing the source into a reverberation chamber with highly reflective walls. At large enough distances from the source the reflected sound wave creates a reverberant sound field and the total radiated sound power is proportional to the square of sound pressure measured in the reverberant field. The major advantage of this method consists of elimination of the necessity to consider the very often complicated directivity pattern of the source. On the other hand, the accuracy is limited by the modal character of the chamber, measured frequency range, source position etc. Also the sound power which the sound source radiates in a reverberation chamber differs from that radiated in an unbounded medium. However, the major factor affecting the accuracy is the spectrum of the sound radiated by the source. Sound power of sources with continuous and broadband spectra can be measured more accurately than the sound power at discrete frequencies or narrow band spectra.

The many aspects of the measurement were considered at the revision of the two major standards for the sound power measurement: The American National Standards Institute ANSI 1.21-72 and The American Society of Heating, Refrigerating and Air-Conditioning Engineers ASHRAE 36-72. Also, the International Standards Organization is currently revising its recommendation. This paper presents some new data related to the problem of measurement of sound power at discrete frequencies, resulting from the research performed at The Pennsylvania State University.

2. DETERMINATION OF SOUND POWER

The sound field in the reverberation chamber, generated by the source can be considered as composed from two parts: the direct field and the reverberant field, consisting of all the reflected waves. If we are far enough from the source, the energy density of the reverberant field is much larger than the density of the direct sound field so that the direct field can be neglected.

From simple energy considerations Sabine and Eyring derived the relation between the sound power W radiated by the source and the sound pressure p measured in the reverberant field

$$|p|^2 = \frac{4\,W\,\rho c}{\bar{\alpha}\,S} = 24.8\ \rho c\ \ W\frac{T}{V} \qquad (1)$$

where $\bar{\alpha}$ is the average wall absorption coefficient, S the surface of the room, V the volume of the room in m^3, T the reverberation time and ρc the wave impedance of the air. Eq. (1) indicates that the sound power is proportional to $|p|^2$. The derivation of equation (1) also involves the assumption that the energy density and sound pressure in the reverberant field are distributed uniformly. However, considering the existence of normal modes it can be found theoretically and experimentally that the sound pressure amplitude varies with the position in the sound field. These variations depend on the spectrum of measured sound and are very large if the chamber is excited at single frequencies. In order to use equation (1) for sound power calculations a suitable space averaging of the sound pressure squared has to be performed. The accuracy of the results depends on the method and way of averaging as well as on modal density at the frequency considered.

Another source of uncertainty and inaccuracy in the sound power determination is the radiation of sound into the reverberation chamber.

The sound power radiated by the source into a free space differs from the power radiated into the reverberation chamber. If the source is small compared to the wavelength the radiated power is generally given by

$$W_p = Q^2\ \mathrm{Re}\ (Z_{rad}) \qquad (2)$$

where Q is the volume velocity of the source surface and Re (Z_{rad}) is the real part of the radiation impedance.

In unbounded medium, the real part of the radiation depends only on the frequency but in a medium with boundaries or obstacles the reflecting waves affect the radiation impedance and therefore the radiated power depends on the source position.

3. SOUND FIELD IN REVERBERATION CHAMBER AT SINGLE FREQUENCIES

If the sound source radiates single frequency the reflected waves travelling through the room interfere in such a way that the sound pressure amplitude varies with position as it can be seen from Fig. 1. In order to determine the sound power from Eq. (1), the average value of the sound pressure squared has to be found. The error involved in averaging can be found if the sound pressure is considered as a random variable. Lubman[1] and Schroeder[2] derived the statistical distribution of p^2 assuming that the sound field is perfectly diffuse. They found that the amplitude of p^2 has an exponential distribution so that the cumulative distribution of p^2 normalized to the average $\overline{p^2}$ is given by

$$F\,(p^2/\overline{p^2}) = 1 - e^{-p^2/\overline{p^2}} \qquad (3)$$

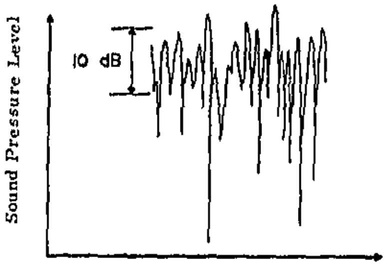

Fig. 1 Variations of sound pressure level with position if the sound field is excited by single frequency.

The variance of this distribution $\sigma^2 = 1$. If the average value of p^2 is found from measurements in N points with mutual distance larger than one-half wavelength, the variance of the average is $\sigma^2 = 1/N$. In practical situations the averaging can be performed either by installing in the chamber several microphones or by moving a microphone along a continuous path. Lubman and Waterhouse[3] derived that for a straight microphone motion the variance is given by $\sigma^2 = 1/1 + \frac{2L}{\lambda}$ where L is the length of the microphone travel path. If $L \gg \frac{\lambda}{2}$, one-half wavelength long path is equal to one point of measurement. Fig. 2 shows the expected error of determination of the average of p^2 as a function of the number of measurement points with the chosen confidence limits. It can be seen that relatively large number of microphones has to be used should the error be kept small.

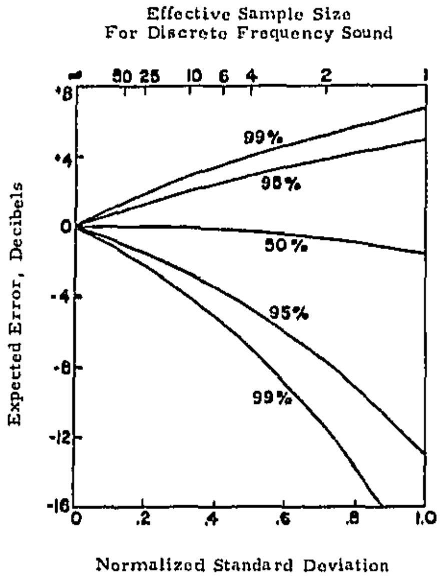

Fig. 2 Expected error as a function of standard deviation.

The mentioned relations and derivations are valid only if the sound field is diffuse and as the measurements show the relations hold only in the region of high modal density, when sufficient number of modes is excited. Detailed measurements performed by Tichy[4,5] show that even in the region of high modal density the magnitude of the variance for the single point may deviate from one, probably because one of the modes is excited with much higher energy.

Fig. 3 shows the normalized variance of the distribution of the sound pressure squared as obtained from measurements with 12 microphones at each frequency. All the measurements presented here were performed on a model reverberation chamber. In order to make the data generally applicable the frequency scale is expressed in terms of $f.\sqrt[3]{V}$ as it corresponds to the number of modes involved. The spread of the variance of Fig. 3 is rather large partly due to the fact that the variances were obtained from only 12 data points. The upper curve in Fig. 4 shows the average value of variances obtained by averaging the values from Fig. 3 for each 1/3 octave band. It can be seen that these averages also deviate from the theoretical value 1.

If the sound pressure average has to be determined with sufficient accuracy many microphones or long microphone traverses have to be used as it can be seen from Fig. 2. The number of microphones can be reduced by using a rotating vane[6,7,8,9]. Fig. 5 shows the effect of the vane on the variations of sound pressure with position. It can be seen in comparison with Fig. 1 that the variations are of the order of 8-10 dB.

The effect of the vane on the variance of the p^2 values shows Fig. 4 (lower curve). In order to express numerically the effect of the vane on the sound pressure averaging, Lubman suggested to use figure of merit defined as the ratio of variances. It can be seen from Fig. 4 which shows the values of figure of merit(dashed curve) averaged over 1/3 octave bands, that the values are between 1.9 and 5.6. In order to obtain the same accuracy of the space averaging with the vane at low frequencies about one-half of the microphones is sufficient.

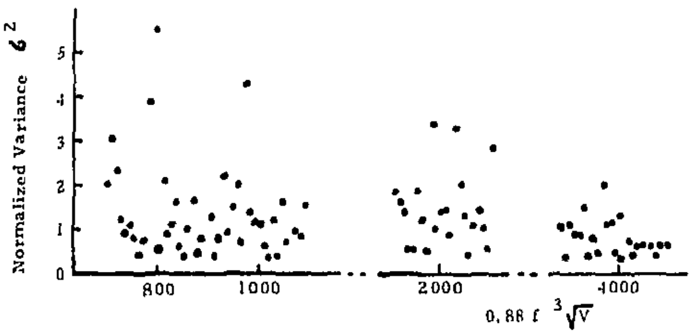

Fig. 3 Normalized variance measured at different frequencies.

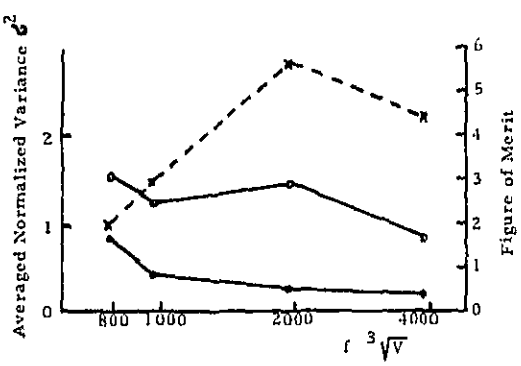

Fig. 4 Figure of merit (dashed curve) and average value of normalized variance of the amplitude of sound pressure squared. Upper solid curve with the vane standing, lower solid curve with the vane rotating.

Fig. 5 Sound pressure level as a function of position with the vane rotating.

4. QUALIFICATION PROCEDURE OF THE REVERBERATION CHAMBER

Another random variable affecting the accuracy of the measurement of sound power is the radiation impedance at the source position. Waterhouse[10] and Maling[11] did extensive calculations for different source positions and damping of the chamber. Lyon[12] and Maling made an estimate of the variance of the ratio of sound power radiated in the reverberation chamber and free space as a function of frequency.

In order to make a practical judgment of the usability of the reverberation chamber, both mentioned standards ANSI 1.21-72 and ASHRAE 36-72 require to apply a qualification procedure. According to this procedure a special sound source is placed in the actual source position and the space average value of the sound pressure level averaged on the pressure square basis is measured at about 22 frequencies for each 1/3 octave band of interest. After the averages are corrected for the loudspeaker frequency response the standard deviation of the 22 averages of sound pressure level for each 1/3 octave band is calculated. The standards mentioned require the standard deviation to be smaller than certain value depending on frequency.

An extensive study of various parameters as source position, modal damping, number of used microphones and primarily rotating vanes is being conducted at The Pennsylvania State University.

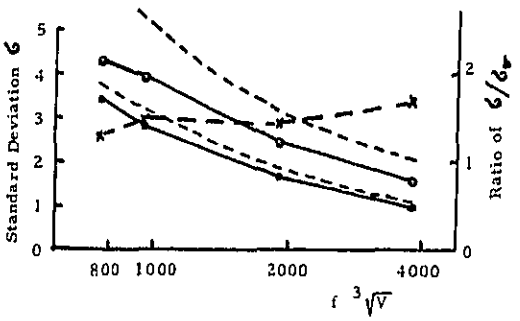

Fig. 6 Standard Deviation of Sound Pressure Level Obtained from Qualification Procedure. Upper solid curve is for vane standing, lower solid curve for vane rotating. The dashed curve shows the values of the ratio σ/σ_{w}. Figure also gives the range of σ as obtained from the international round robin test.

At this time only partial results of the effect of rotating vane and source position are available. Fig. 6 shows two curves of the standard deviation averaged over 5 source positions distant more than one-quarter wavelength. The upper curve has been obtained without the vane and the lower curve shows the reduction of standard deviation if a vane is used. The absorption coefficient of chamber wall was approximately 0.05. The dashed curves in Fig. 6 represent the boundaries for σ obtained from an international round robin test on the qualification of reverberation chambers.

As it can be seen from Fig. 6 (dashed curve) the gain obtained by the vane on qualification procedure is smaller than the gain on the sound pressure averaging as expressed by figure of merit, after we compare the squared values of the ratio of σ without and σ_{w} with the vane rotating. However, the vane can help and be substantial to qualify the chamber.

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A NEW METHOD FOR VEHICLE NOISE MEASUREMENTS

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INTRODUCTION

The Society of Automotive Engineers presently lists three standards for measuring noise from highway vehicles (J986a, J366 and J672a). The International Organization for Standards (ISO) Recommendation R362 also applies to highway vehicles. The acoustic measurements differ only in the microphone distance from the vehicle track. The SAE specifies 50 ft while the ISO specifies 7.5 meters (approx. 25 ft). Gatley and Frye [1] surveyed legal methods of enforcement in many cities and states in the U. S. and several European countries. They report measurement distances of microphone to vehicle track from 5 to 75 ft.

In reading the published literature on vehicle noise many different test distances are found which make comparisons difficult. Occasionally one also finds data with no test distance stated. Then no comparisons can be made to other measurements.

A vehicle noise measurement method is proposed here that produces a measure independent of distance. Field measurements are simplified, a more fundamental quantity and more information about the noise source is obtained.

VEHICLE NOISE SOURCE LEVEL

The acoustic pressure from a sinusoidally pulsating spherical source radiating into an infinite uniform medium is given by [2]

$$p = \frac{P}{r} e^{j(\omega t - kr)}$$

where r is the distance to the measuring point and P is the source pressure at a unit distance. The term (ωt-kr) is simply a phase factor. When the amplitude of the pressure is measured at a known distance r, the fundamental property of the source, P, is given by

$$P = pr$$

When p is measured in decibels referred to 2×10^{-5} N/m^2 (SPL) and r is in meters, a source level can be defined as 20 $\log_{10}P$. For a noise source this might be called a Noise Source Level (NSL) which can be calculated as follows:

$$NSL = SPL + 20 \log_{10} r \qquad (1)$$

Noise Source Level can be measured by measuring the Sound Pressure Level, filtered by any desired weighting function, and adding the distance term at the instant of the SPL measurement. A system to do this and provide a Noise Source Level read out will be described in the next section.

It is obvious that a noisy vehicle is not a spherical pulsating source radiating into an infinite ideal medium. However, in the far field of the source (at least one vehicle length) spherical spreading, or 1/r dependence of the sound pressure would be expected. If microphone distance is limited to a few hundred feet, the effects of atmospheric and ground surface absorption and wind shear should be minimal.

A noisy vehicle is a complex source so the measured Noise Source Level must be considered an equivalent Sound Pressure Level at one meter which is the result of many noise sources. However, since NSL is independent of distance, microphone placement is not critical. In addition, when range is measured continuously as the vehicle passes and NSL is plotted versus angle from the vehicle track, a directivity pattern for the radiated noise is produced. This is additional and important information which is easily obtained with the system to be described.

NOISE SOURCE LEVEL METER

To implement the measurements of NSL called for in equation (1), two very common devices shown in Figure 1 are needed. One is a standard sound level meter with a potential output proportional to meter reading and a traffic radar. Since most traffic radars are doppler type giving only velocity, a velocity potential must be integrated to give distance. Two logarithmic converters can convert sound pressure and distance to dB and these potentials summed to give a potential proportional to NSL. This quantity can be displayed digitally, on a simple meter, plotted versus time or plotted versus vehicle track angle.

In the present implementation, integration is started at a known range and a negative velocity potential is integrated to give a closing range as the vehicle approaches. Since the radar cannot sense positive and negative velocity, the velocity potential is reversed at the closest point of approach where zero doppler occurs.

To operate the system, the vehicle is tracked in angle with the radar as it passes. Integration and measurements are started at the known distance. Measurements are terminated at some convenient distance after the vehicle passes.

SYSTEM CALIBRATION AND MEASUREMENTS

Calibration

Initial setup and calibration of the system and some example measurements were made on a typical residential street. The geometry is shown in Figure 2. The microphone and radar were placed about 7.5 meters from the vehicle track as in the ISO standard. A total test run of 100 meters was used. Since the measurements were to start at 50 meters, an integrator initial condition corresponding to that range was used. A sound pressure calibrator producing 114 dB re 2×10^{-5} N/m^2 at 1 KHz was used. The "A" weighting on the sound level meter was used. The pen position on the recorder then corresponded to 114 dB at 50 meters. Adding 34 dB gives 148 dB at one meter or an NSL of 148 dB. When the sound level meter sensitivity was increased 40 dB the calibrate level corresponded to an NSL of 108 dB.

A critical test of the system consisted of using a known constant noise source. The NSL of this source should be constant with range and angle. A vehicle mounted loudspeaker that could be continually pointed at the sound level meter was driven by. The loudspeaker was excited by an amplifier driven by an electrical noise generator. An NSL of about 104 dB (A) was produced by the loudspeaker.

A plot of the sound pressure level of this noise versus time, as it passed, is shown in Figure 3. A variation of about 20 dB over the 100 meter path is seen. When the range correction was applied to produce NSL the plot of Figure 4 was produced. In this case a variation of $\pm$ 3 dB is noted. Since the data are from successive runs, accurate comparisons with time cannot be made.

When sound pressure level versus angle was recorded as the noise source passed, the plot of Figure 5 was produced. Again a variation of 20 dB is noted. A subsequent run produced the measured NSL of Figure 6. In this case, a variation of less than ± 2 dB is noted. Considering the statistical variability of a noise source, an uncertainty in starting the integrator at the exact range and the variability between runs, it is felt that the sound was propagating by spherical spreading and that the Noise Source Level Meter was compensating for it.

Measurements

The test area was not suitable for carefully controlled vehicle noise measurement. Thus, no extensive measurements are presented here. Figure 7 is a typical trial measurement. However, Figure 8 and 9 made with the horn on an ancient Volkswagen as a noise source show the importance of using NSL and the polar directivity pattern. The recording of Sound Pressure Level in Figure 8 shows a maximum of 84 dB(A) at a track angle of 75°. This level corrected to one meter gives a NSL of 98 dB(A) which is verified at 75° by the measured NSL plot in Figure 9. However, Figure 9 also shows that at track angles of 15°, 30° and 160° the NSL is 106 dB(A), or 8 dB higher than that observed by measuring only SPL.

CONCLUSIONS

At best, noise measurements on a moving vehicle are difficult. Transient conditions always prevail and exact environmental and geometric conditions are difficult to repeat. The direct measurement of Noise Source Level removes the dependence on geometry. When NSL is plotted versus track angle maximum levels are easily identified and the directional character of the source is obtained. When it is suspected that environmental conditions are causing other than spherical spreading of the acoustic signal, a calibrate run with a known noise source can be used to provide a correction to the measured data.

The use of NSL to characterize vehicle noise provides several advantages. In noise rating the number is fundamental to the vehicle and not dependent on the specific measurement environment. For vehicle noise enforcement, the measurement distance can be removed from the legal statutes. In addition, maximum NSL, which may differ from maximum SPL, can be used for enforcement. For predicting the effect of vehicle noise on local environments, NSL can be extrapolated under the sound propagation conditions expected to produce SPL contours. For vehicle noise research, the directivity patterns and frequency analysis of the measured noise provide new information to aid in discovering the sources of vehicle noise. It should also be pointed out that the principle is applicable to other vehicles such as aircraft, boats, snowmobiles, etc.

The Noise Source Level Meter described here is the device in its most rudimentary form. For noise enforcement, a hand held device with peak speed and peak NSL indication could be developed. Hand held radars and sound level meters are already on the commercial market. For vehicle noise ratings a tripod mounted system with a directional microphone to improve acoustic signal-to-noise ratio is recommended. For noise research the directional microphone and noise signal recording for later analysis is suggested.

ACKNOWLEDGEMENTS

Assistance in assembling the test system, performing the tests and preparing the report from J. L. Warnix, D. D. Reynolds, S. R. Shaw, J. Hubbs, T. Teasdale and Mrs. Shirley Thompson is greatly appreciated.

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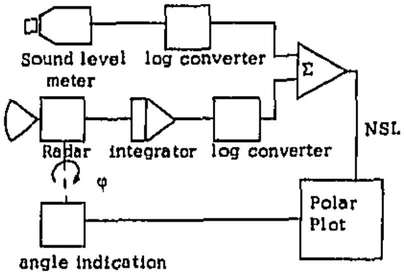

Figure 1. Noise Source Level Meter

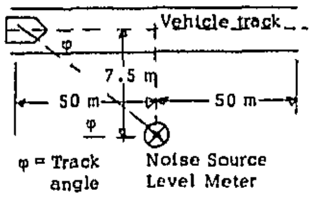

Figure 2. Test Geometry

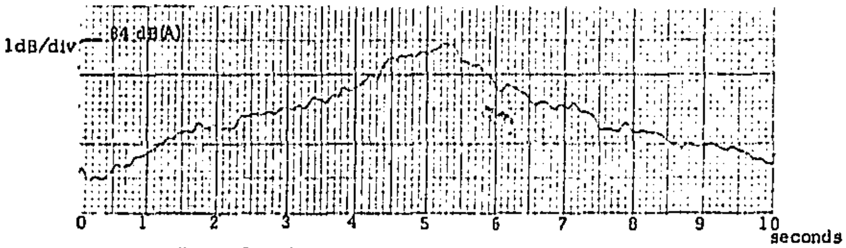

Figure 3. SPL vs. time, constant noise source.

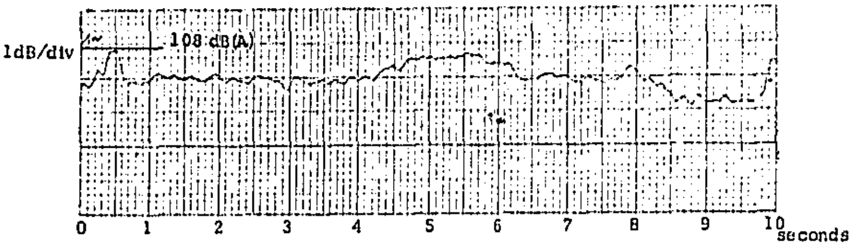

Figure 4. NSL vs. time, constant noise source.

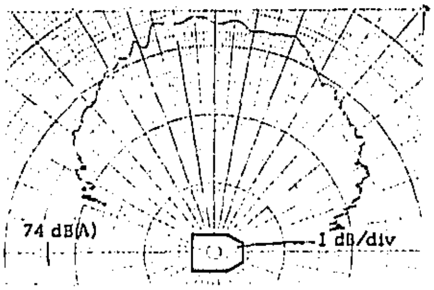

Figure 5. SPL, constant noise source.

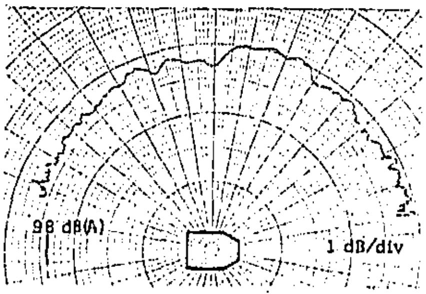

Figure 6. NSL, constant noise source.

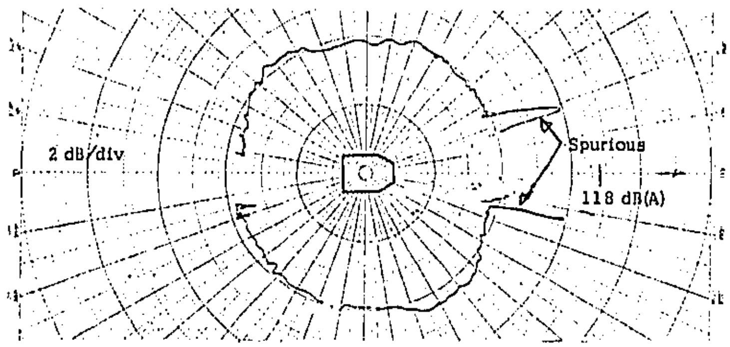

Figure 7. NSL for Ford pickup truck, 30 mph, 3rd gear.

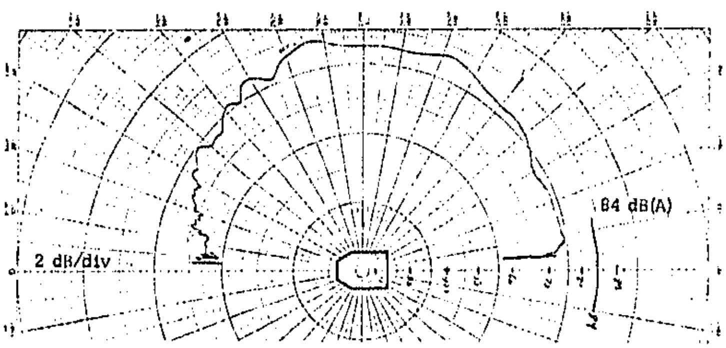

Figure 8. SPL for Volkswagen Horn, 20 mph.

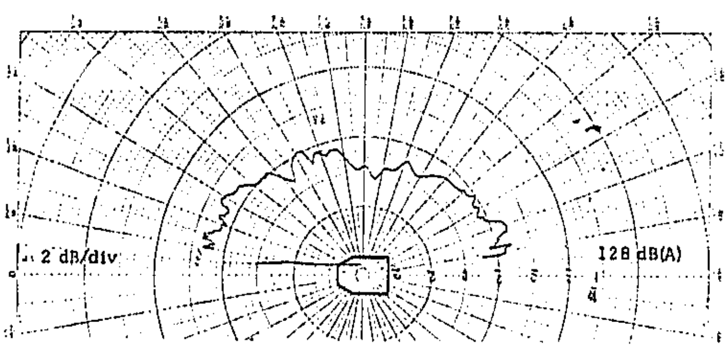

Figure 9. NSL for Volkswagen Horn, 20 mph.

CORRECTION PROCEDURE FOR OUTDOOR NOISE MEASUREMENTS

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1. INTRODUCTION

Noise measurements are of limited value unless they can be normalized to standard conditions and extrapolated to other circumstances. This procedure requires taking into account a large number of factors, including the operating conditions and directivity pattern of the source, spherical divergence loss, atmospheric absorption, refraction and diffraction from temperature and wind gradients and from obstacles, reflection and absorption from the ground and other surfaces, and possible various other sources. We will not consider the operating conditions or directivity characteristics since they will vary with the particular noise source. Likewise we will assume that spherical divergence, or 6 dB for double distance, which is universal and independent of frequency for a spherical wave, requires no further discussion. We shall deal briefly with atmospheric absorption and refraction, and in a little more detail with ground reflection.

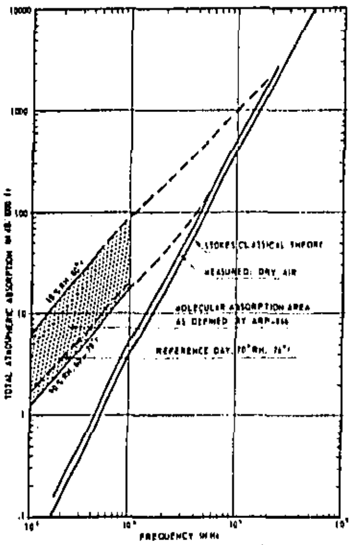

Fig. 1 Atmospheric Absorption

2. ATMOSPHERIC ABSORPTION

Figure 1 shows the absorption in dB per 1000 ft of the atmosphere between 1 kHz and 1 mHz. At lower frequencies the absorption never exceeds a few dB/1000 ft and can usually be ignored. At very high frequencies the controlling factor in atmospheric attenuation is the classical or Stokes absorption [1] which rises with the square of frequency and severely limits high ultrasonic propagation in air. We are most concerned with the region from 1 to 10 kHz, including frequencies which are heavily weighted in loudness and Perceived Noise Level (PNL) computation. Here, as shown by the hatched region, the absorption in dB per unit distance [2] varies over a range of one to five, depending on the temperature and, particularly, the humidity. The dashed line is the attenuation in air at a temperature of 76°F and 70% relative humidity - the so called "reference day" in aerospace sound measurements.

Since airplane tests are often carried out on hot dry days when atmospheric absorption is near the maximum, normalization to the reference day requires addition of quite large corrections to measured spectrum data, especially in the higher frequency bands where adjustments of 40 to 60 dB/1000 ft are not unusual. Experimental data is always limited by some sort of noise threshold where for some spectrum bands the desired data is lower than a noise floor introduced by ambient acoustic noise near the microphone or, more often, by electrical noises in the measuring, recording and analyzing system. Adding the atmospheric correction to such noise floor - limited data leads to the absurdity of a signal apparently getting louder as distance increases. Accordingly the view has often been expressed that the absorption values, which are published for each filter band as function of humidity and temperature in the SAE Aerospace Recommended Procedure (ARP) #866, are excessive.

We believe this question has been resolved by a study performed for NASA Langley by R.L. Miller and Oncley [3]. A very large volume of data has been accumulated by Boeing in flight tests of various airplanes over arrays of as many as 27 microphones. Records taken at 0.5 second intervals are analyzed into 1/3 octave bands prior to computation of Perceived Noise Level (PNL) and Equivalent Perceived Noise Level (EPNL). Supplementary flight performance and attitude data recorded on the plane, position data from radio and photo-theodolite measurements, and meterological data from sounding balloons, instrumented aircraft and ground instrumentation are all gathered at the same time with a synchronized time code.

Data used in the attenuation study came from 24 separate flyovers with climbouts at constant angle and constant thrust over an on-line array of 8 or 9 microphones. On the assumption that for a given emission angle from the airplane the acoustic output is unchanged during the few seconds of the climbout, measurements at the various microphones represent different transmission distances of the same signal. A computer routine selected and interpolated the analyzed spectra, subtracted spherical divergence, and plotted each band level as a function of path distance. The slope of the best-fit straight regression line through the points is the empirical attenuation constant. When distant points were distinctly off the slope indicated by closer points it was taken as an evidence of noise floor in the data and the points were discarded.

At the same time the attenuation constants for each frequency were computed following ARP 866 using a stratified atmosphere in 500 or 1000 ft layers as determined from upper atmosphere soundings. The differences between the empirical attenuation constants and those from ARP 866 were plotted as a function of frequency band for each of 10 emission angles on each flyover. Only sound reaching the ground at incidence angles of less than ± 25° was used to minimize ground reflection and refraction effects.

About 5760 regression lines were calculated from over 35,000 data points. While individual flight tests sometimes gave discrepancies of 5 to 10 dB/1000 ft between the measured and calculated attenuation rates, there was no systematic error and an average of all 24 flights, as shown in Figure 2, reduced the discrepancy to about 1 dB/1000 ft except in the upper two bands, where residual noise floor effects appeared to be present. It was concluded that the ARP 866 tables are valid but that rather large errors can exist in a single flight test largely as a result of errors in humidity as measured by the radiosonde system. In any case ARP 866 corrections should improve accuracy if there is no noise threshold problem and if atmospheric measurements are accurately representative of the total transmission path.

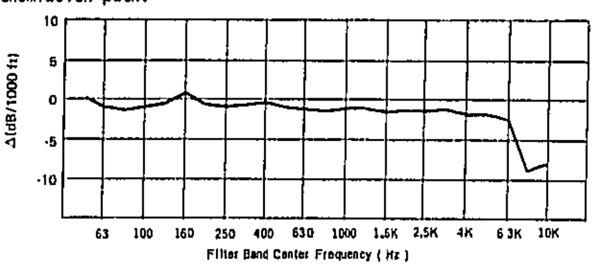

Fig. 2 Difference Between Empirical and ARP 866 Atmospheric Absorption Coefficients (Mean of 24 Tests)

3. REFLECTION EFFECTS

The spectrum distortion arising from ground reflection and absorption is harder to correct. A typical geometry is shown in Figure 3. The direct wave from the source S at height h_s goes to the receiver R at height h_r over a direct path length r, while the reflected wave travels a distance $r' = r + \Delta r$. The microphone measures the algebraic sum of the direct and reflected wave, which, as seen by the solid curves in Fig. 4, will be a maximum for $k\Delta r$ equal to zero or $2n\pi$, and a minimum for $k\Delta r = (2n-1)\pi$, n being any

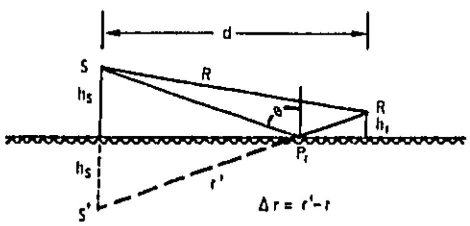

Fig. 3 Diagramatic Representation of Test Geometry

integer. When either source or receiver height is small in comparison to total distance d, the first minimum, when $k\Delta r = \pi$, occurs at a frequency f_0

$$f_0 = \frac{r\, c_0}{4\, h_s h_r} \qquad 1$$

where c_0 is the speed of sound. Minima also occur at $3f_0$, $5f_0$, etc., and maxima at $2f_0$ $4f_0$, etc., and at near zero frequencies.

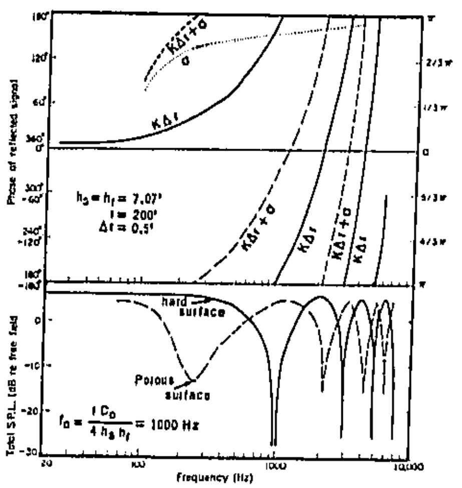

Fig. 4 Phase of Reflected Signal and Pressure Difference between Total Pressure and Free Field Value. ($h_s = h_r = 7.07$ ft, $d = 200$ ft, $\Delta r = 0.5$ ft, $c_0 = 1000$ ft/sec)

When the surface is porous, not only is reflection incomplete, but there is a phase delay δ in the reflected signal [4]. Minima now exist when $(k\ \Delta r + \delta) = (2n-1)\pi$, and maxima when $(k\ \Delta r + \delta)$ = zero or $2n\pi$. For surfaces which have been studied, including gravel, sand, grassy turf and absorptive building materials the values of δ lie below 90° for low audio frequencies and approach asymptotically to 180° in the high audio range as shown by the dotted curve in Figure 4. The interference pattern shown by the dashed line in Fig.4 shifts to the low end, with higher order minima shifting almost to the next lower maximum and the lowest minimum falling somewhere around 1/3 to 2/3 the value of f_0 for a hard surface.

The above analysis holds for a point source and narrow band filtering. W. L. Howes [5] has discussed the case of finite band filters including fixed band width and constant percentage band width filters. For octave and 1/3 octave bands the difference between measured data and free field values is given by

$$\Delta N_i(\mathrm{dB}) = 10 \log_{10} \left\{ 1 + \left(\frac{r}{r+\Delta r}\right)^2 + \frac{2r}{r+\Delta r} \left[\frac{\sin(\alpha\, k_i\, \Delta r)}{\alpha\, k_i\, \Delta r} \right] \cos(\beta k_i \Delta r) \right\} \qquad 2$$

where α = .1155 (1/3 octave) or .3535 (octave)
β = 1.0066 (1/3 octave) or 1.0606 (octave)

and k_i is the propagation constant or wave number in air corresponding to the i th filter band center frequency ($k_i = 2\, f_i/c$).

This study has been further extended by Hoch [6], and Hoch and Thomas [7]. For an absorbing surface they modify the above equation to give

$$\Delta N_i = 10 \log_{10} \left\{ 1 + |Q_i|^2 \left(\frac{r}{r+\Delta r}\right)^2 + \frac{2|Q_i|r}{r+\Delta r} \left[\frac{\sin(\alpha\, k_i\, \Delta r)}{\alpha\, k_i\, \Delta r} \right] \cos(\beta\, k_i\, \Delta r + \delta_i) \right\} \qquad 3$$

Here $Q_i = |Q_i| e^{-j\,\delta_i}$ is the complex ratio of reflected to direct sound in the i th filter band. Hoch uses it interchangeably with the reflection coefficient R_p, giving as its value

$$Q_i = R_p = \frac{(Z_i/\rho c)\cos\theta - 1}{(Z_i/\rho c)\cos\theta + 1} \qquad 4$$

where Z_i is the complex acoustic impedance of the surface, ρc is the characteristic impedance of free air, and θ is the incidence angle of the sound. This is a useful formula for correction of data from aircraft overflight, but as Hoch points out, it is not valid near grazing incidence (θ = 90°). It is in fact questionable for incidence angles larger than probably 70°.

For near grazing cases, such as road traffic noise, airplane taxi and runway noise or jet engine fixed ground test stands, the situation is considerably more complex. An analysis for narrow band spectra and periodic tones is given by Oncley [4]. It is based on early work by Rudnick [8], and shows that the ratio of reflected to direct wave (the Q of earlier equations) should be represented by the more general expression

$$Q = |Q| e^{-j\delta} = R_p + (1-R_p)F(w) \qquad 5$$

where R_p is the reflection coefficient (similar to the definition for Q in equation 4 and F(w) is given by

$$F(w) = 1 + j2\, w^{1/2} e^{-w} \int_{-jw^{1/2}}^{\infty} \exp(-u^2)\,du \qquad 6$$

and

$$w = \frac{j\,2\,k\,r}{|1-R_p|^2 \sin^2\theta} \left(\frac{\rho c}{Z}\right)^2 \left[1 - \left(\frac{k}{k_2}\right)^2 \sin^2\theta\right] \qquad 7$$

Equation 6 can be expanded as

$$F(w) = 1 - 2w\, e^{-w} \left(1 + \frac{w}{3(1!)} + \frac{w^2}{5(2!)} + \frac{w^3}{7(3!)} + \cdots\right) + j(\pi w)^{1/2} e^{-w} \qquad 8$$

or, for large values of w (w > 10)

$$F(w) = -\left(\frac{1}{2w} + \frac{1\cdot 3}{(2w)^2} + \frac{1\cdot 3\cdot 5}{(2w)^3} + \cdots\right) \qquad 8a$$

In equation 7, k_2 is the complex propagation constant in the surface and k is the propagation constant in air, $k = 2\pi f/c$. The reflection coefficient R_p, also should include an additional term

$$R_p = \frac{(Z/\rho c)\cos\theta - \sqrt{1 - (k/k_2)^2 \sin^2\theta}}{(Z/\rho c)\cos\theta + \sqrt{1 - (k/k_2)^2 \sin^2\theta}} \qquad 9$$

which for small θ reduces to equation 4. For small values of θ, equation yields very large values of w, and F(w), by equation 8a, approaches zero, so in equation 5, $Q = R_p$.

At grazing incidence, cos θ goes to zero and $R_p = -1$. Since the total signal is

$$P_{tot} = P_{dir} + P_{refl} = P_{dir}\ (1 + Q),$$

as R_p goes to -1, Q becomes $(-1 + 2F(w))$ and

$$P_{tot} = 2\ P_{dir} \cdot F(w)$$

Figure 5 and equation 8a show that in the region where $w > 10$, F(w) is approximately inversely proportional to w. In equation 7 we see that w is proportional to distance r, as well as to frequency. This leads to a total signal falling off at 6 dB for distance doubling in addition to the spherical divergence loss, or a total of approximately 12 dB/distance doubling when w is large. Moreover the signal falls off with frequency at a 6 dB/octave rate in this regime. These effects are in addition to the frequency selective interference pattern discussed earlier.

For finite bandwidth filtering equation 3 can still be used but for angles within 10° or 15° of grazing incidence, the value of Q should be calculated from equations 5 through 9 instead of equation 4.

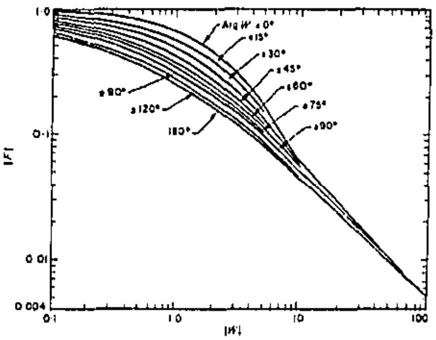

Fig. 5 Graph of the Reflection Function F(w) as a Function of the Numerical Distance w. Magnitude of F(w) vs. magnitude of w; phase of w as a parameter.

It is sometimes possible to largely eliminate reflective interference distortion by mounting the microphone flush with the surface, especially when a concrete or other hard surface can be provided. In this case, in equation 1 with $h_r = 0$, the value of f_o approaches infinity, or at least can be made to lie outside the measurement range, and all measurements are 6 dB above the free field value. This procedure is effective for indoor model testing and outdoor tests over concrete test pads, airplane runways or parking lots. It could presumably be used over water with some care to keep the microphones dry. It also gives good results for flyover aircraft tests when the microphone is set into a surface such as a metal-faced plywood baffle of at least 4' radius.

For near grazing applications over porous surfaces the dips and peaks of the interference pattern are eliminated by flush mounted microphones as the Δr terms in equation 3 go to zero. The values of $|Q_i|$ and δ_i still vary with frequency, but the resulting frequency distortion may perhaps be more easily compensated than that for higher microphones.

There is not time today to discuss the effects of refraction from thermal and wind gradients or the diffraction from obstacles and rough surfaces. These topics are discussed elsewhere, but it should be noted that they can cause severe high frequency losses with the flush microphone technique. Heat storage by concrete slabs in the sun may cause exaggerated ray bending. Data from flush microphones should be compared for high frequency losses with that from microphones at higher positions unless very careful meteorological measurements of near-surface thermal and wind gradients are taken.

Computer programs have been developed by The Boeing Company to adjust and extrapolate spectra from airplane overflights and jet engine static tests, incorporating the directional characteristics of jet and turbomachine noise in the source, atmospheric absorption and ground reflection. It is necessary to point out that the correction for ground reflection is still rather unreliable because of the inadequate data on impedance and propagation constants of the ground. The best impedance data available is that of Dickinson [9], Figure 6, taken for various soil samples in Wales. The general range is shown by the hatched regions. Moisture in the soil generally increases both real and reactive impedance components, although Dickinson says that slightly moist soils have lower impedance values than totally dry ones. His measurements cover only the range from 200 to 1000 Hz and there is no assurance that they can be extrapolated linearly. They also are taken for nominal incidence, and no technique for grazing incidence impedance measurements of undisturbed soil has been published. Values for the propagation constant k_2 in soil are essentially non-existent, but the equations are not highly sensitive to it, and ratios of k_2/k of the order of $3 + j\ 3$, or higher, give results which are generally in accord with experimental results.

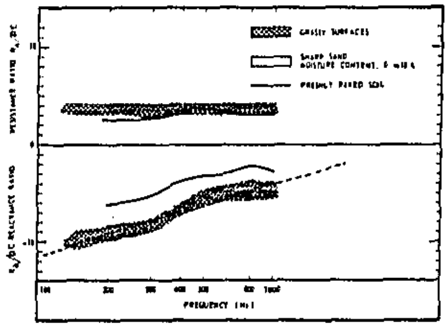

Fig. 6 Some Measured Values of Normal Impedance of Different Ground Surfaces

Because of the variations of ground impedance with moisture content, cultivation, vegetation, solar radiation and other factors, plus distortions of the interference pattern by winds and thermal gradients, compensation for interference distortion in experimental data has had only limited success. In prediction however, where a calculated or known free field spectrum must be extrapolated to positions over porous soils, the above equations give curves and values within the range of those usually observed. A great deal of the data scatter in measurements along the surface comes from impedance variations and much better repeatability would be obtained in airplane certification tests (particularly at sidelines) if data could be taken with flush microphones and then be extrapolated to ear level position assuming a standard soil impedance, just as atmospheric absorption is referred to a reference day.

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TRANSIENT AND STEADY STATE SOUND ABSORPTION COEFFICIENTS OF FIBERGLASS AND POLYURETHANE FOAM

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ABSTRACT

The transient sound absorption coefficients of Fiberglass and polyurethane foam of different thicknesses are obtained by the short pulse method. The average normal absorption coefficients (350 Hz to 4000 Hz) are compared with those obtained by the impedance tube method.

Oblique incidence tests have been performed on the Fiberglass at an incidence angle of 30°; the directivity of the speaker has been considered. The oblique sound absorption coefficient was found to be higher than the normal absorption coefficient.

The pulse method is found better than the impedance tube method in obtaining sound absorption coefficients for engineering purposes.

1. INTRODUCTION

When sound waves interact with a solid surface, the sound energy is partially reflected and partially absorbed by the surface. The absorption coefficient is defined as the ratio of the sound energy absorbed by the surface to the sound energy incident upon the surface. The absorption coefficient of a material, however, not only depends on the mechanical properties and the thickness of the material, but also depends on the frequency and the angle of incidence.

The measurement of these coefficients has been mainly conducted in the steady state as in the reverberant room method and the impedance tube method. In the reverberant room method the time rate of decay of sound energy density is measured with and without a patch of the absorbing material under test in the room. Due to some reasons such as the diffraction of the sound field around a finite sized absorbing patch, the measured absorption coefficient has often been found to vary for the same material.

In the impedance tube method, the standing wave ratio of a specimen is measured. It is assumed that the specimen interacts with a plane wave at a single frequency. It ignored the vibration conditions of the tube and the mechanical limitation of the speaker. The data measured by this method are also very scattered.

In the pulse method a transient wave is used in obtaining the ratio of the reflected sound pulse from an absorbing material to the direct sound pulse from the speaker at the same distance.

The pulse method was first used by Spandöck[1] and was often used since then. But the propagation of this transient wave has always been treated with the concept of geometric optics, which is a weak assumption for the common frequency range of 250 Hz to 8000 Hz.

In this paper, the use of the pulse method is reviewed. The normal sound absorption coefficients obtained by pulse method are compared with those obtained by the impedance tube method. Oblique sound absorption coefficients of Fiberglass are measured taking into account the directivity of the speaker.

2. PULSE METHOD

Spandöck used the pulse method testing the reflection coefficient of some materials at 800 Hz and 4000 Hz with normal incidence. The average sound pressure of the short pulse was assumed inversely proportional to the distance from the sound source. Tests of oblique waves at 800 Hz, transmission loss and the sample size were also performed. In the tests, the short pulse contained many sinusoidal cycles. In the oblique wave tests, the speaker was assumed to generate a wave uniformly propagating in all directions. This assumption is not usually true because the speaker has certain directivity.

Rogers and Watson[2] used the tone burst method for measuring the sound absorption coefficient of Celotex C-4 and some building boards. By using the analogy of geometric optics, a speaker was put in the focus of a paraboloidal sound mirror hoping to produce a plane wave. In the test the speaker was facing toward the paraboloidal mirror. It was assumed that the speaker did not generate a sound wave propagating backward toward the sample. Also the interaction of the reflected sound wave from the sound mirror with the finite sized speaker was ignored. Furthermore, because of the finite times required for the transit of sound waves from the speaker to the mirror and out again, there is no real assurance that reconstruction of the "rays" into a plane wave front was actually achieved. In Reference 2, the relation used is:

$$\alpha = 1 - \left|\frac{P_r}{P_i}\right|^2$$

where

α = absorption coefficient

P_r = the average value of the scattered or reflected sound pressure,

P_i = the average value of incident sound pressure

which was useful, nevertheless, in obtaining interesting data.

Davern and Hutchinson(3) used the tone burst method for testing the reflection coefficient of polyurethane ether foam wedges. Deviations from the inverse distance law as a function of the frequency were measured. It was found that the deviation was negligible for frequencies above 125 Hz.

The use of the short pulse method may be limited by the intrinsic nature of the speaker. Although the oscillator and tone burst generator can produce a nearly true single frequency pulse, the speaker cannot be expected to reproduce the frequencies of the excitation faithfully. However, the speaker does produce a pulse which constitutes a wide band frequency spectrum, with an envelope peak in the neighborhood of the excitation, or the tone burst, frequency. The consequence of this is that the absorption coefficient thus obtained is the value not at a single frequency, but rather it is the absorption coefficient over a fairly wide band of frequencies, and for most engineering purposes such a coefficient is more useful than that obtained at single discrete frequencies. Hence some commercial materials only used the value at 500 Hz or often by a noise reduction coefficient obtained by averaging the Sabine absorption coefficient at 250, 500, 1000, and 2000 Hz.(4)

Since the short pulse is a combination of wide band frequencies, the general shape of the pulse is relatively immaterial. It is convenient in this test to compare the peak amplitudes of the reflected pulse and the incident pulse. The wave front can give the true arrival time. This technique is preferably used in the anechoic chamber because there are no other reflections. The pulse method is not very convenient for low frequencies; the testing distances must be larger in order to avoid overlap of the incident and reflected pulses. In the present test, the frequency tested was between 350 Hz and 8000 Hz.

3. METHODS OF MEASUREMENT

The testing assembly is shown in Figure 1.

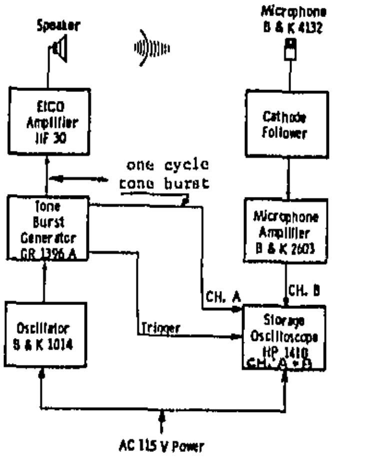

Fig. 1–General assembly

A single cycle of sinusoidal wave at a specified frequency was produced by an oscillator and a tone burst generator. After amplification, the signal was fed into a speaker to generate a short pulse. The microphone was placed at a distance "Y" and the speaker was placed at a distance "X+Y" in front of the specimen as in Figure 2. The size of the specimen was approximately 5 x 6 ft. The inverse square law was checked before testing the sound absorption coefficients. The microphone first received a pulse P_1 which was the direct wave from the speaker. The signal P_3 was the pulse reflected from the absorbing material, or from a rigid smooth reflecting wall, and had traveled a distance of X+2Y.

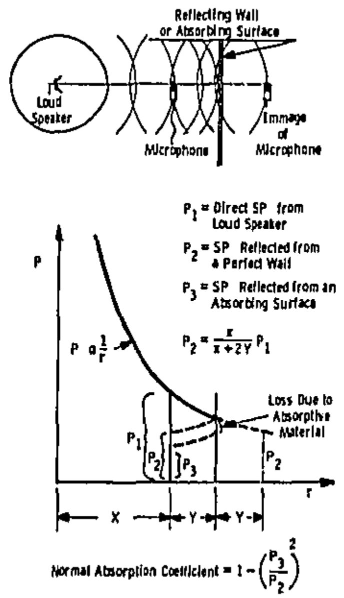

Fig. 2–Sound absorption coefficient due to wave of normal incidence

With the reflecting wall in place, it was verified that the direct pulse pressure P_1 and the reflected pulse peak P_2 were related as

$$\frac{P_2}{P_1} = \frac{x}{x + 2y}$$

With the absorptive sample in place, P_2 was reduced to P_3, due to absorption. The intensity reflection coefficient, defined as

$$R = 1 - \alpha = \left(\frac{P_r}{P_1}\right)^2$$

where α is the absorption coefficient, was then calculated from

$$R = \left(\frac{P_3}{P_2}\right)^2$$

Figure 3 shows a typical oscillogram of a normal incidence pulse on a 6" thick polyurethane foam. The oscilloscope was triggered synchronously with the one-cycle sinusoidal tone burst pulse. Due to the limitation of the mechanical response of the speaker, the direct pulse was seen to have more oscillations before it decayed out. The magnitude P_1 of one nominal cycle was

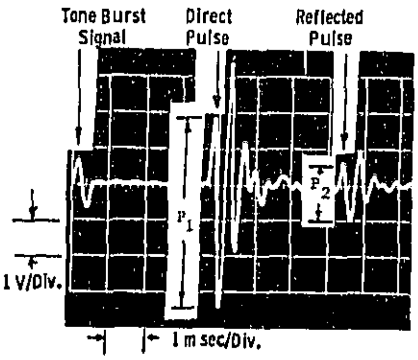

Fig. 3–Typical oscilligram results of a normal incidence short pulse on a 6" thick polyurethane foam at tone burst frequency 2000Hz.

(Channel A add channel B)

chosen such that it corresponded to one peak and one valley. The reflected pulse P_3 was chosen in the same way as P_1. The arrival times of P_1 and P_3 were found to agree with those calculated from the distances among the speaker, microphone and the specimen surface.

The direct pulse from the microphone was fed to a Federal Scientific Real Time Analyzer. Figure 4 showed the actual frequency spectrum from a speaker with the tone burst generator frequency at 1000 Hz. It may be seen that the speaker generated a wide band noise having an envelope peak near 1000 Hz.

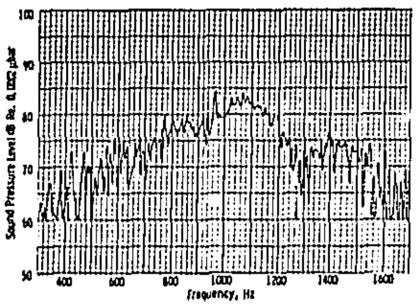

Fig. 4–Actual frequency spectrum from a speaker with tone burst generator frequency f = 1000 Hz.

4. RESULTS OF NORMAL INCIDENCE TESTS

The absorption coefficient of Fiberglass 6" thick and 3 1/2" thick, polyurethane foam 6" and 1" thick under normal incidence wave were tested with the pulse method. The absorption coefficients were plotted against the nominal frequency as shown in Figure 5. The nominal frequency was assumed to be the tone burst frequency. From the figure we see that the absorption of 3 1/2" thick Fiberglass is higher than that of 6" thick polyurethane foam. The 7" thick fiberglass absorbs practically all the incident sound energy in the frequency range of 350 Hz to 8000 Hz. It is significant that with the pulse method, the data in each test are reproducible with essentially very little scatter.

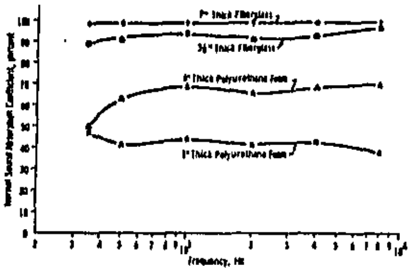

Fig. 5—Normal sound absorption coefficient of various materials measured by short pulse method

The sound absorption coefficients of these materials were also obtained from an impedance tube test. For the frequencies below 800 Hz, the sample size was 10 cm in diameter and for the frequencies higher than 800 Hz, the sample size was 3 cm in diameter. By measuring the first maximum sound pressure level L_{pmax} and minimum sound pressure L_{pmin} in the tube, the pressure ratio n was calculated.

$$L_{pmax} - L_{pmin} = 20 \log_{10} n$$

and $\quad n = P_{max}/P_{min}$

For each frequency the normal sound absorption coefficient was calculated from the following formula[5]

$$\alpha = \frac{4}{n + \frac{1}{n} + 2}$$

The results are shown in Figure 6. From the figure we see a wide scattering of data during the change of sample size at the frequency of 1000 Hz. The impedance

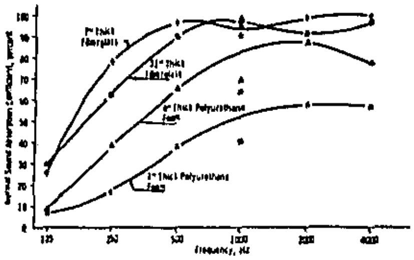

Fig. 6—Normal sound absorption coefficients of various materials measured by impedance tube method.

method showed the materials had lower absorption coefficient as the frequency was lowered. Again the absorption coefficient of a 3 1/2" Fiberglass was higher than that of the 6" thick foam.

For most engineering purposes, a single absorption index is useful. By averaging the sound absorption values over a frequency range (350 Hz to 4000 Hz) the results are shown in Table 1.

TABLE 1

Average Normal Sound Absorption Coefficient for Various Materials (350 Hz to 4000 Hz)

	Impedance Tube Method (Steady State Test)	Pulse Method (Transient Test)
Fiberglass 7" thick	0.95	0.99
Fiberglass 3 1/2" thick	0.90	0.91
Foam 6" thick	0.70	0.64
Foam 1" thick	0.45	0.42

In the impedance tube one often finds large variations in P_{max} and P_{min} leading to some uncertainty in the values measured by this method. Nevertheless, the impedance method is a standard method, and may be used for comparison purposes as was done in Table 1, where reasonably good agreement is found. No values for the absorption coefficients obtained in a reverberation room are available.

5. RESULTS OF OBLIQUE INCIDENCE TESTS

The equation being commonly used to evaluate the sound absorption coefficients of oblique incidence is given as follows:[5]

$$\alpha_\theta = 1 - \left|\frac{(Z/\rho c)\cos\theta - 1}{(Z/\rho c)\cos\theta + 1}\right|^2$$

In this equation, the incident wave is assumed to be plane, and the acoustic impedance Z is independent of the angle θ. ρ is the density of the air and c is the sonic velocity in air. The theoretical value of α_θ is always less than 1. However, very little experimental data on oblique absorption coefficients are available.

In an experimental test, it is difficult to produce a true plane wave. Figure 7 shows a test with the microphone positions 15° apart along an arc whose radius was 8'-0" from the speaker. The relative magnitudes of the sound pressure at the tone burst frequencies 4000 Hz and 8000 Hz are plotted for each position. It is seen that the speaker had a strong directivity. It is equally important to know the directivity of the reflected pulse. Figure 8 shows a test of a pulse from a small reflecting board 6" in diameter in the anechoic chamber. The speaker position was at 30° to the left of the normal to the reflector and 4'-0" away from it. The microphone was placed at various angular positions on this 4 ft radius. The reflected sound pressures measured were divided by the maximum sound pressure measured at 8'-0" away from the speaker without the reflector in place. In the figure we see that the reflected sound profile has a similar directivity. The maximum sound pressure of the reflected pulse was 30° right to the reflector normal. In this experiment the magnitude of the reflected pulse from the small perfect reflector was found not to obey the inverse square law. This is probably due to the diffraction from the reflector edges.

Figure 9 shows a test with a large size of Fiberglass replacing the small perfect reflector in the anechoic chamber. Again the microphone positions were at various positions at a 4'-0" radius from the point O. The sound pressures were compared to the maximum sound pressure measured at 8'-0" away from the speaker. The oblique intensity reflection coefficient R_θ or $1-\alpha_\theta$ was obtained by dividing the reflected sound intensity at 30° to the incident sound intensity measured at 8'-0" away from the speaker. The results of the 7" thick and 3 1/2" thick Fiberglass subjected to a 30° incidence wave at the tone burst frequency 4000 Hz and 8000 Hz are shown in Table 2.

TABLE 2

Oblique Sound Absorption Coefficient α_θ (θ=30°) of Fiberglass at 4000 Hz and 8000 Hz

	f = 4000 Hz	f = 8000 Hz
Fiberglass 7" thick	0.99	0.99
Fiberglass 3 1/2" tk.	0.98	0.96

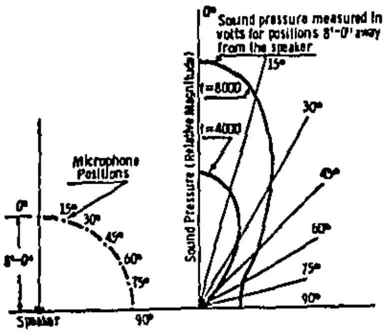

Fig. 7—Sound directivity from a speaker at tone burst frequency f = 4000, and 8000 Hz.

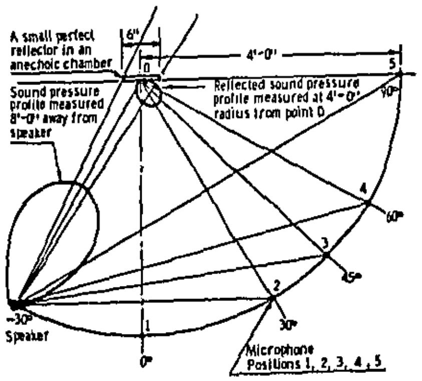

Fig. 8—Directivity of a reflected pulse from a small perfect reflector at tone burst frequency 4000 Hz.

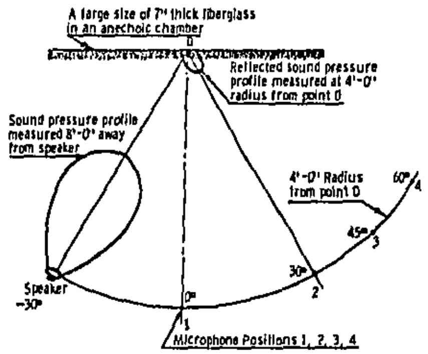

Fig. 9—Oblique sound reflection of 7" thick fiberglass at tone burst frequency 4000 Hz.

By comparison of Table 1 and 2, we see that the oblique sound absorption coefficient of 3 1/2" thick Fiberglass is higher than its normal incidence sound absorption coefficient. Hence, if a higher absorption coefficient is desired for the same thickness of a material, the surface can be made oblique to the impinging waves. This may have added to the effectiveness of the wedges used in the anechoic chambers.

6. DISCUSSION

The mechanical properties of sound absorption materials subjected to a steady state sound wave have been studied with the help of electric circuit analogy.[6,7] By matching the data from the impedance tube test, an acoustical impedance of the material in terms of the material porosity, density and its thickness can be calculated theoretically. But the impedance method tube is based on the assumption of a plane wave impinging on the small circular sample. Due to the wave reflections from the tube walls as well as from the microphone rod, and the speaker behavior, the situation would probably be more complex than the simple theory allows. This difficulty leads to some uncertainty in determining L_{pmax} and L_{pmin}. The error introduced into the value of n is even more difficult to predict because L_{pmax} and L_{pmin} are of the same order of magnitude.

The pulse method, by applying a transient wave interacting with the absorbing material, is free from most of the above difficulties. The data obtained by this method are highly reproducible. Since numerous common noises are in the form of transients, such as machine noise and impact noises, in obtaining the sound absorption coefficient for engineering purposes, the pulse method is probably more suitable than the impedance tube method.

7. SUMMARY

1. The normal sound absorption of Fiberglass and polyurethane foam were tested by the steady state wave (impedance tube method) and by the transient wave (short pulse method). The average values of these coefficients over a frequency range of 350 Hz to 4000 Hz by the two methods agreed reasonably well. The results are shown in Table 1.

2. The characteristics of the pulse method is that the data are highly reproducible in each test.

3. The pulse method may be used to obtain the absorption coefficient at oblique incidence, which was found higher than the normal absorption coefficient for the same thickness. The typical results are shown in Table 2.

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JET, COMPRESSOR AND AIRCRAFT NOISE SOURCES

JET NOISE RESEARCH - PROGRESS AND PROGNOSIS

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1. INTRODUCTION

The objectives of the present paper are two-fold. First, to indicate the role of jet noise research in the progressive reduction of jet engine noise during the past two decades (Sections 2 and 3). The second objective is to review recent new theoretical and experimental developments which point to a more complete understanding of noise generation mechanisms, and new possibilities for noise reduction in the future. Sections 4 to 6 include brief discussions of improved diagnostic techniques based on updated versions of aeroacoustic theory, modern probing tools and instrumentation, and emerging new insights into the fluid dynamics of noise production by jet engines. The paper focusses primarily on the noise of subsonic jets in view of recent trends away from supersonic transport in North America.

2. EVOLUTION OF ENGINE TECHNOLOGY

Todays advanced technology turbofan engines, comprising the General Electric CF6, Pratt and Whitney JT9D, and Rolls Royce RB211 reflect almost two decades of concerted effort to combat jet aircraft noise. The results of this effort are extremely encouraging; as depicted in Figure 1, the flyover OASPL at 1000 feet is more than 20 decibels lower for the new L 1011 aircraft than for the early turbojet powered versions of 707 and DC-8 transports. This reduction has been achieved in spite of substantial increases in thrust capability of individual engines (now in excess of 40,000 lbf per engine). In fact during the 20 year period, the acoustic efficiency (noise power/thrust power ratio) has been reduced by a factor of almost 1000, from about 1% to .001%. The reduction in noise exposure area for a constant 110 db contour is equally impressive, as depicted in Figure 2. To bring about this substantial improvement the modern high bypass turbofan engine incorporates a number of advanced design features, as illustrated in Figure 3.

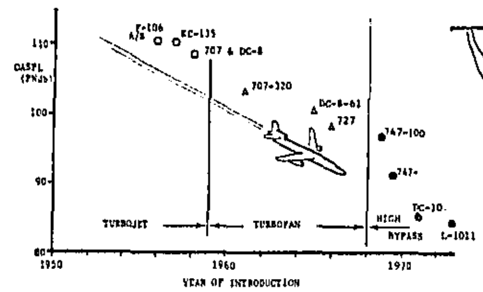

FIGURE 1 - FLYOVER AIRCRAFT NOISE
(1000 Feet Altitude)

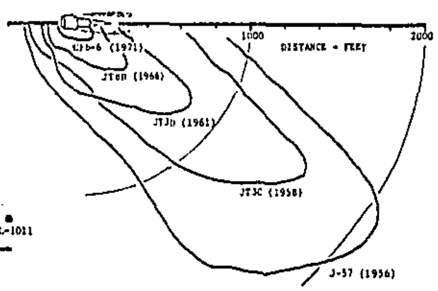

FIGURE 2 - FOOTPRINTS OF TURBINE ENGINES
Equivalent Distances for 110
db Contours

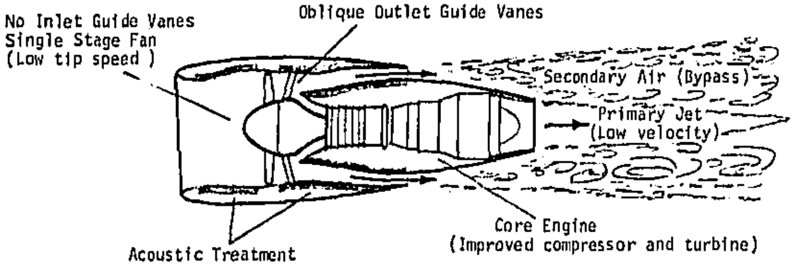

FIGURE 3 - MODERN HIGH BYPASS TURBOFAN ENGINE (Bypass ratio 5 - 6)

The early turbojet engines were mechanically quite similar to the core engine of today's high bypass turbofan (i.e., comprising a multistage axial flow compressor, combustion chambers, and an axial flow power turbine). In order to provide the required thrust, these engines were characterized by a high temperature, high velocity exhaust jet ($M_{exit} \rightarrow 1$). The vigorous mixing of rapidly moving exhaust gas with the surrounding quiescent air leads to the development of an intense turbulent flow accompanied by a loud jet roar. This was the dominant source of aircraft noise in the early days, especially during take-off. Principally by ad-hoc testing it was soon discovered that moderate reductions in the noise of the primary jet could be effected by weakening and spreading the region of turbulent mixing with the aid of special multi-cellular exhaust nozzles. However the reductions achieved with such devices rarely exceeded 4 - 6 PNdb, and the performance and weight penalties were substantial.

During the mid 1950's it became evident that jet velocity played a dominant role in the noise of jets. Thanks to the pioneering work of M.J. Lighthill[1] it was learned that the sound power of gas jets increased according to the function $\rho U^8 D^2$ while useful thrust power only increased proportional to $\rho U^3 D^2$. It was apparent that if the exit velocity U could be reduced with a corresponding increase in the effective jet diameter D, the propulsive power could be kept constant while effecting a significant reduction in jet noise. This thinking lead to the evolution of the first bypass engines, in which some thrust was provided by propelling quantities of cool secondary air around the main engine, with the aid of an additional turbofan stage on the front end. These first generation turbofan engines had a fairly low bypass ratio, with only 1.4 times as much air passing through the fan duct as through the main engine. Nevertheless, jet noise reductions in excess of 10 PNdb were realized, together with other operational advantages. As a result the most common of these engines, the Pratt and Whitney JT3D and JT8D are in widespread use in the majority of non wide-body jet transports flying today.

With the advent of the turbofan engine came a new problem; annoying high frequency tones emerged to dominate over the much reduced jet exhaust noise. These tones are generated by complex blade interactions occurring in the multi-stage inlet fan and primary compressor, a subject to be treated extensively by Mr. Sofrin and Mr. Lowson in subsequent papers. It is sufficient to note here that discrete frequency fan noises have been the principal target in development of the advanced high bypass engines. Substantial reductions in this component have been achieved by the use of new fan technology, and by the introduction of special acoustical linings into the inlet and exhaust duct of the main bypass fan.

3. ROLE OF BASIC RESEARCH

While credit is due to engine designers and manufacturers for much of this progress, it is important that we do not overlook the contributions of a sometimes "silent" partner. The concerted efforts of dedicated research workers in industrial, government, and university laboratories have fostered an ever expanding knowledge of the theory and basic physical mechanisms of aerodynamic noise production. Space does not permit a detailed account here of the numerous individual contributions; several recent publications and symposia give a fuller perspective.[2 - 5]

Perhaps the most fundamental result of Lighthill's original theory [1], as generalized by Curle[6], is the retarded potential solution for the acoustic radiation from a region of unsteady flow containing an embedded surface S:

$$p(\underline{x},t) \approx \frac{1}{4\pi x}\int_S \left[\rho\dot{u}_n\right]_{t'} dS + \frac{x_i}{4\pi x^2 c^2}\int_S \left[\frac{\partial}{\partial t}(\dot{f}_i + \rho u_i u_n)\right]_{t'} dS + \frac{x_i x_j}{4\pi x^3 c^2}\int_V \left[\frac{\partial^2 T_{ij}}{\partial t^2}\right]_{t'} dV \qquad (1)$$

where $T_{ij} \equiv \rho u_i u_j + \tau_{ij} + (p - c^2\rho')\delta_{ij}$

In lay terms, this equation says that the instantaneous radiated sound pressure p results from the combined effects of several types of velocity, force, and temperature fluctuations occurring in, on, and around the noise source (e.g., a jet engine). These source fluctuations are represented by the terms in square brackets appearing on the right hand side. As written here, the equation is approximately applicable in the acoustic far field of any noise source with non-supersonic velocities and relatively constant speed of sound c.

An equivalent alternate formalism due to Ribner[7] (or Meecham and Ford[8]) is equally applicable to subsonic flow sources:

$$p(\underline{x},t) \approx -\frac{1}{4\pi x c^2}\int_V \left[\frac{\partial^2 p^{(o)}}{\partial t^2}\right]_{t'} dV \qquad (2)$$

In this approach the sound is viewed as being generated by a volume distributed assemblage of dilation sources, represented by the non-acoustic part of the pressure fluctuations $p^{(o)}$ in the source region.

Until recently, solution (1) has been the starting point for most empirical and experimental investigations of aerodynamic noise. Coupled with extensive measurements of directivity, spectra, and turbulence structure, the theory has produced an elaborate physical description of the elementary source mechanisms, based on the classical concepts of monopole, dipole, and quadrupole radiators. One of the first empirical predictions was Lighthills U^8 law. It has been shown that the basic eddy sound sources in jet turbulence probably radiate in an almost omni-directional manner; that the highly directional nature of jet engine exhaust noise (Figure 2) arises from acoustic beaming phenomena known as convection and refraction[2]. Evidence of dynamic similarity in jet turbulence structure has led to notions about axial distribution of sound source strength and frequency, enabling performance predictions for suppressor nozzles and ejector shrouds.

Extrapolations of the theory have provided explanations of some aspects of sound radiation from supersonic jets. For example, the sound power is found to increase according to a $\rho U^3 D^2$ law, and the directivity is dominantly influenced by supersonic convection of the source "eddies". These and many other insights have helped the engine designer to understand the physics of jet noise.

4. THE NEED TO KNOW

In spite of these positive achievements many unresolved problems remain. Due to the complexity of the current "quieter" engines it is often difficult to detect the small changes in overall decibel level which may result from localized geometrical modifications. Researchers and designers need more direct, effective methods for quantitatively monitoring or predicting the acoustic fractions, spectral nature, and spatial extent of sources in and around engines. Traditional attempts to measure local source strength have been based on squared and time averaged versions of Equation (1) which yield the radiated acoustic intensity ($I \propto \overline{p^2}$) in terms of complex two-point correlations of the source fluctuation variables. Perhaps the most successful example of such an experiment is that of Chu[9] in which the full three dimensional structure of turbulence near a point in a jet exhaust was used to predict the acoustic source radiation from that point. To duplicate Chu's experiment for many source points proves to be impractical because of the enormous number of individual measurements which must be made near each source point.

A simpler technique is needed which entails only one measurement per source point. Coupled with this we look for improved, reliable probing devices for monitoring source fluctuations both in the free stream and on surfaces.

5. NEW FORMALISMS AND DIAGNOSTIC TOOLS

Recently there has been renewed interest in the problems of measuring fluctuating variables in unsteady fluid flows. While some attention has focussed on the measurement of velocity field (e.g., developments in laser and hot film anemometry), the property of greatest interest to flow noise researchers seems to be the so-called "pseudosound" pressure [10-13]. This is at least partly prompted by a resurgence of optimism surrounding the dilatation model of aerosound, as expressed by Equation (2). The scalar nature of $p^{(o)}$ and the availability of new miniature transducers and microphones make this approach attractive. Several recent studies hinge on the development of new probing technology; for example a crossed beam laser technique described by Wilson et al[4]. and a special pressure probe devised by the present author to eliminate turbulence interaction errors[4]. Others have used uncorrected pressure probes to make measurements in turbulent shear flows and, more reliably, in the essentially non-turbulent core region of round jets. Some of this work has led to speculation about a coherent structure of ring-like vortices as a principal source of jet noise, but the contention is yet unproven[10,11].

Unfortunately it has been found that mere measurement of time average fluctuations and simple two-point correlations within the source flow does not provide adequate information in view of the complex relationships between the fluctuation variables and the resulting sound (as indicated by equations 1 and 2). What is needed is a more quantitative correlation between radiated sound and appropriate forms of the fluctuation variable. A recently developed technique establishes just such a correlation between the "cause and effect" pair[13-17].

A simple illustration of the new formalism follows. If we multiply Ribner's dilatation solution (equation 2) by the sound pressure p on both sides and take a time average:

$$\overline{pp}(\tau) \approx \frac{-1}{4\pi c^2 x} \int_V \left[\frac{\partial^2}{\partial \tau^2} \overline{p^{(o)}p}(\tau')\right]_{\tau + \frac{r}{c}} dV \tag{3}$$

In differential form we find:[15]. $(\tau = 0)$

$$\frac{\overline{dp^2}}{dV} \approx \frac{-1}{4\pi x c^2} \left[\frac{\partial^2}{\partial \tau^2} \overline{p^{(o)}p}(\tau')\right]_{\tau' = \frac{r}{c}} \tag{4}$$

The quantity on the left hand side represents the contribution to the total mean squared sound pressure associated with the region where the turbulent pseudosound $p^{(o)}$ was detected. The term in square brackets establishes the coherence between $p^{(o)}$ and p. Experimentally, the procedure is as follows. A pressure probe is inserted into the source region as shown:

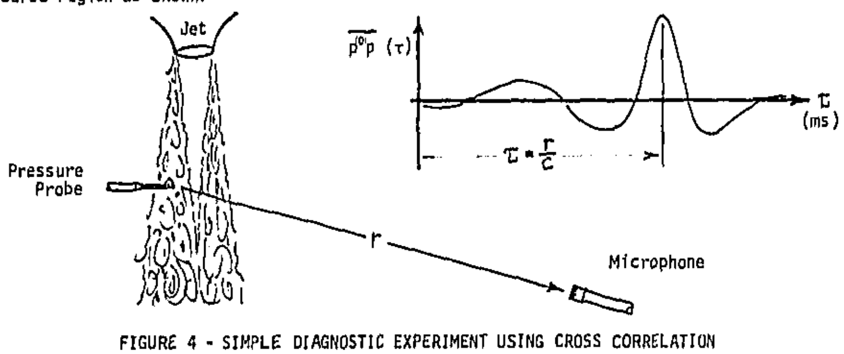

FIGURE 4 - SIMPLE DIAGNOSTIC EXPERIMENT USING CROSS CORRELATION

The detected psuedosound $p^{(o)}$ is multiplied instantaneously by the far field sound pressure p, using a special time delay computer (correlator). The resulting function $\overline{p^{(o)}p}(\tau)$ will vary with time delay τ in the manner depicted on Figure 4. The function is observed to peak at a time corresponding with the acoustic travel time between pressure probe and microphone. This "bump" on the function implies that there is something in common between the source fluctuation and the overall radiated sound. From its shape and magnitude the local source strength may be extracted, using equation (4).

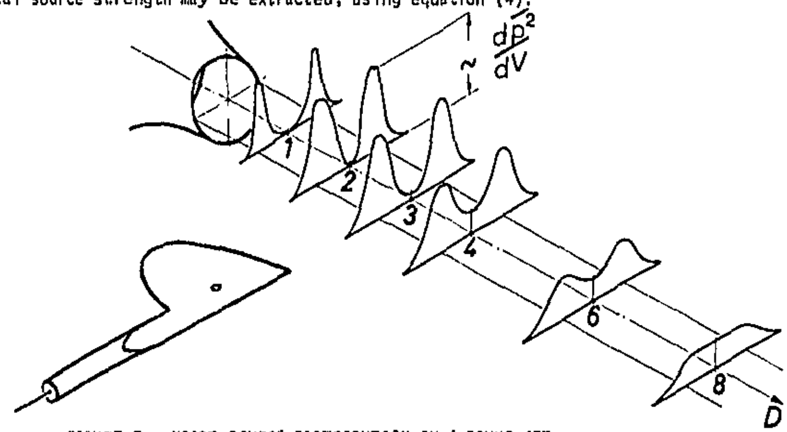

FIGURE 5 - NOISE SOURCE DISTRIBUTION IN A ROUND JET

The results of such an experiment for a round turbulent air jet are illustrated in Figure 5. The complete spatial distribution of source strength is plotted for a plane passing through the axis of the jet. It is seen that the strongest sources are concentrated in the regions of most energetic turbulence (i.e., the shear layers) as might be expected. There is no apparent radiation associated with the central core of the jet. As a bonus the correlation function $\overline{p^{(o)}p}$ (τ) also carries important spectrum information. By taking the Fourier transform of (3) the elementary frequency spectrum associated with each source point is obtained.[15]

To achieve the results of Figure 5 it was necessary to devise a special pressure probe, for two reasons[13]. First the velocity field interacts with a probe of conventional cylindrical geometry in an adverse way, producing measurement errors which are not generally negligible. Secondly, extraneous dipole radiation from a conventional probe can lead to serious contamination of the correlation functions, especially in small laboratory jets. A thin foil-shaped probe of special configuration, as depicted inset on Figure 5, overcomes this problem.

A similar version of the present formalism, more complicated but also very useful, results if equation (1) is multiplied through by sound pressure on both sides[15].

$$\overline{p^2}(\underline{x}) = \frac{1}{4\pi x}\int_S \left[\overline{\rho \dot{u}_n p}\right] dS - \frac{x_i}{4\pi x^2 c}\int_S \left[\frac{\partial}{\partial\tau}\left(\overline{f_i p} + \overline{\rho u_i u_n p}\right)\right] dS + \frac{x_i x_j}{4\pi x^3 c}\int_V \left[\frac{\partial^2}{\partial\tau^2}\overline{T_{ij}p}\right] dV \qquad (5)$$

Terms on the right hand side of this equation could be used to describe the concentrations of source strength associated with numerous phenomena in aircraft engines. For example tailpipe pulsations and jet turbulence, rough combustion (entropy fluctuations), edge tones and other surface interaction sources arising from flow impingement on engine casing, suppressors nozzles, and blading in both fans and turbines can be assessed by forming the appropriate correlations with the far field sound p.

On a laboratory scale, Lee and Ribner[16] were the first to apply this technique to an investigation of jet mixing noise by computing narrow band (filtered) correlations $\overline{T_{ij}p}$, using a hot wire probe to measure an approximation of the stress tensor T_{ij}. Other experiments have demonstrated the applicability to surface dipole radiation from rigid aerofoils [14,17]. Currently we are investigating the distribution of surface dipole strength on rotating fan blades, using a radio link to transmit the blade pressure fluctuation, which is subsequently correlated with the far field sound.

A completely different approach to the measurement of noise source distribution utilizes directional far field microphones. This concept, which also offers some encouraging prospects, will be described in a subsequent paper by Chu, Laufer, and Kao.

6. WHAT OF THE FUTURE?

Due to the substantial reductions of jet and fan noise which characterize the current high bypass engines, numerous new and hitherto unexplored sound sources are being unmasked. These reside dominantly in the hot end of the core engine. In the near future we must develop probing devices and transducers suitable for use in hot, high speed gases, which will enable separate measurements of thermal and aerodynamic fluctuations. Then, with the aid of techniques such as those discussed earlier, we shall continue to lower the baseline levels of jet engine noise. Simultaneously, we must recognize that a majority of our present aircraft are still powered by the noisier, low bypass turbofans. Again we must exercise new diagnostic technology to develop optimum retrofit packages or replacement engines for these aircraft. Future prospects of large STOL aircraft flying near to the cores of our cities offer perhaps the greatest challenge of all to engine designers and noise researchers. High lift devices such as blown flaps and augmentor wings will introduce powerful new sources of surface interaction noise. The high bypass engine promises to one day become a ducted turboprop. And what of the SST? If we are to continue the trend of Figures 1 and 2 there is much hard work ahead.

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FAN NOISE MECHANISMS AND CONTROL – I

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INTRODUCTION

Fan noise is a prominent component of aircraft powerplant noise. As shown in Figure 1a, it is generated internally by the fan components, propagates through the inlet and fan discharge ducts, and radiates to the ground from both inlet and discharge duct openings. Additional important engine noise sources are indicated and include jet mixing noise from fan and primary streams, turbine noise, and core compressor noise radiated forward through the fan.

During an airplane flyover, noises from these engine sources peak at different instants and contribute to a complicated history of the flyover noise event (Figure 1b). Since all engine noise sources are also present during static ground noise tests, special means must be used to separate fan noise so that its characteristics can be studied. Some of the special test rigs used in fan noise programs will be described. Through their use, supplemented by analytical methods, a fairly clear picture of fan noise has emerged.

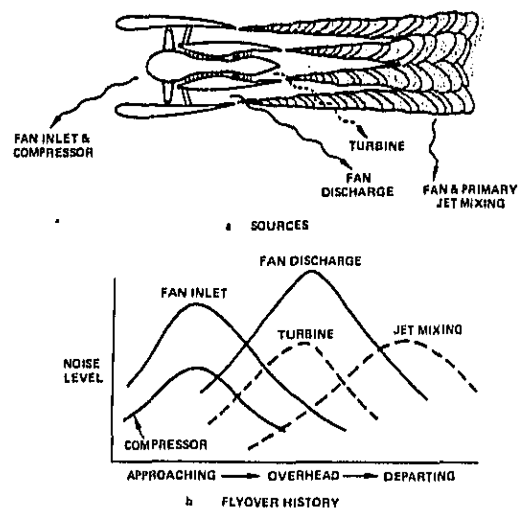


Figure 1 Major Powerplant Noise Sources

Fan noise radiates in a complicated directivity pattern from the inlet and discharge ducts. At each angular position in the far field, the noise is conveniently described by means of its spectrum, or frequency distribution of acoustic energy. Figure 2 gives two illustrative spectra of forward radiated fan noise. In Figure 2a, typical of subsonic blade tip speed single stage fan operation, a broad base of continuous sound, called broadband noise, exists over a wide frequency range. This noise is generated by random pressure fluctuations in the fan, and sounds, generally, like jet mixing noise. Prominent features of the spectrum are the two sharp peaks or "spikes" at fan blade passage frequency and its harmonic. These discrete tones correspond to fan whine and are generated by periodic pressure fluctuations on the fan airfoil surfaces.

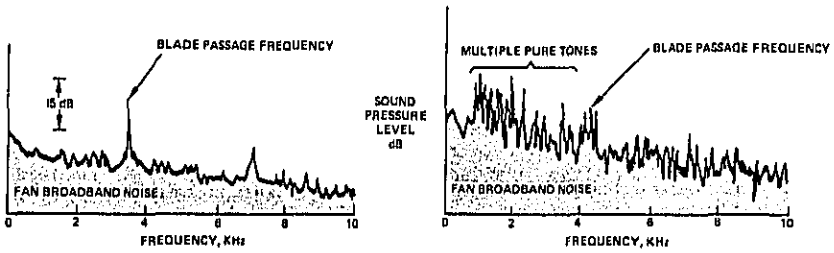


Figure 2 Fan Noise Spectra at Subsonic and Supersonic Tip Speeds

At supersonic fan blade tip speeds, noise of a different character appears. It is exhibited in Figure 2b as a closely spaced series of discrete frequency spikes separated by shaft rotation frequency and has a distinctive sound described variously as deep, rich, raspy or buzz saw. Multiple pure tone (MPT), combination tone noise, and buzz saw are the usual descriptive terms. In brief, it results from the rotating pattern of shock waves from the fan blades when their tip speed is supersonic. Due to the non-linear propagation behavior of shock waves, slight variations in the shock structure from blade to blade, caused initially by small manufacturing deviations, are amplified with distance from the fan into a pattern having marked irregularities. Since this irregular pattern turns with the rotor, its basic frequency is that of shaft rotation speed and all multiples thereof rather than being restricted to blade passage frequencies which would result from theoretically identical blading.

PHASES IN FAN NOISE EMISSION

Three processes are involved in fan noise; generation, duct propagation, and radiation. These are suggested in Figure 3. Understanding of fan noise is facilitated by examining the features of these processes, and noise reduction concepts are made more feasible to develop by separating the overall problem into these areas.

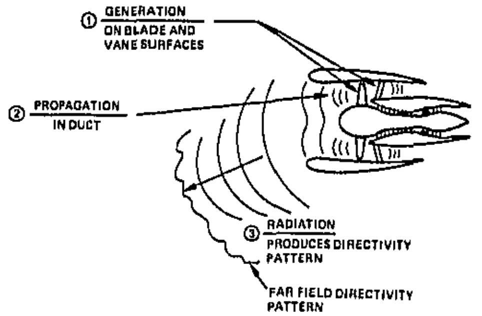


Figure 3 Phases in Emission of Fan Noise

Basically, fan noise is generated at the rotating blades and stationary vanes. Both broadband and discrete blade passage noise result from fluctuating blade surface pressures. These fluctuations arise from inflow velocity variations into a stage due to the turbulent flow structure, airfoil wakes from upstream stages, and interaction of blade and vane potential fields. At supersonic fan blade tip speeds, additional noise is produced by the blade shock wave patterns. These generation processes will be more fully reviewed a little farther on. It will serve to point out here that the essential generation problems are aerodynamic rather than acoustic in nature. While the basic performance of a fan is understood, analytical description of the flow details still leaves much to learn. And it is exactly these details, in particular the unsteady features, that are responsible for the infinitesimal fraction of the mechanical power that is converted to acoustic energy. Several theories for unsteady blade airloads exist, but to this date they have not been critically compared in the contex of fan geometry. More importantly, no comprehensive measurements of the airfoil fluctuating surface pressure distributions have yet been made in a fan to check theory. A rich and important field of aerodynamic research is here awaiting much needed analytical and experimental exploration.

As will presently be described, both discrete frequency blade passage noise and multiple pure tone noise generated by the fan aerodynamic loads take the form of coherent, periodic acoustic wave patterns in the fan inlet and exit ducts. These patterns propagate in spiral paths to the free ends from which they then spread out and radiate to free space. The process of duct propagation has received a great deal of attention because it is subject to control both by sound-absorbing duct lining and also by selecting fan rotor and stator geometry. Duct lining is a separate major topic and will not be reviewed here. But propagation in a hardwall duct also has features which can be exploited for noise reduction. It is well known that a pipe, as exemplified for example by a "speaking tube", is a good conductor of sound. These familiar cases usually involve the equivalent of a pulsating piston or diaphragm at the transmitting end which vibrates to and fro in a synchronous manner. In a fan or compressor, the excitation or driving force on the duct is fundamentally different. Instead of pulsing back and forth in an umbrella fashion, there is a rotating pattern of circumferentially alternating high and low pressure ridges extending radially from the centerline to the walls. This spinning excitation pattern may be visualized by recognizing qualitatively that the pressure very near the face of the rotor alternates periodically as the blades spin by a fixed point. This spinning pattern, containing B (B = blade number) cycles of pressure variation around the circumference provides the excitation, and is easily seen to differ markedly from a snychronous piston-like structure.

Whereas a synchronously pulsing pattern always propagates axially in a duct, a rotating pattern behaves in a more complex manner. It has been learned that the circumferential velocity component with which the pattern sweeps the wall must actually be supersonic to allow it to propagate in a spiral path along and out the duct. If the pattern spins more slowly, such that its curcumferential wall speed is subsonic, then it will not truly propagate. Instead, the magnitude of the pressure fluctuations will decay exponentially with axial distance from the source plane. Thus, the rotating pressure field of a subsonic tip speed fan rotor decays inside the inlet and discharge ducts and does not contribute to the noise heard from the powerplant. However, the interactions between rotor blades and stator vanes or other circumferential disturbances produce other patterns of noise at blade passage frequencies. It will be shown that in many cases interaction noise is produced in patterns that spin at high supersonic speeds. These patterns spiral easily through the fan ducting and are responsible for the noticable discrete frequencies present in the spectra of fan noise. By controlling the blade to vane number ratio, it is possible to bring the circumferential wall speed of the interaction patterns below sonic speed, cause them to decay inside the powerplant, and thus effectively eliminate this source of noise. Unfortunately there is sufficient flow distortion due to other influences affecting the fan so that there remains a considerable level of blade passage noise after blade-vane interaction has been eliminated. Many details of duct propagation remain to be explored. These include primarily the effects of varying duct contours and airflow gradients. Methods for analyzing the propagation of fan broadband noise are considerably more involved since the processes are basically random and statistical considerations must be employed.

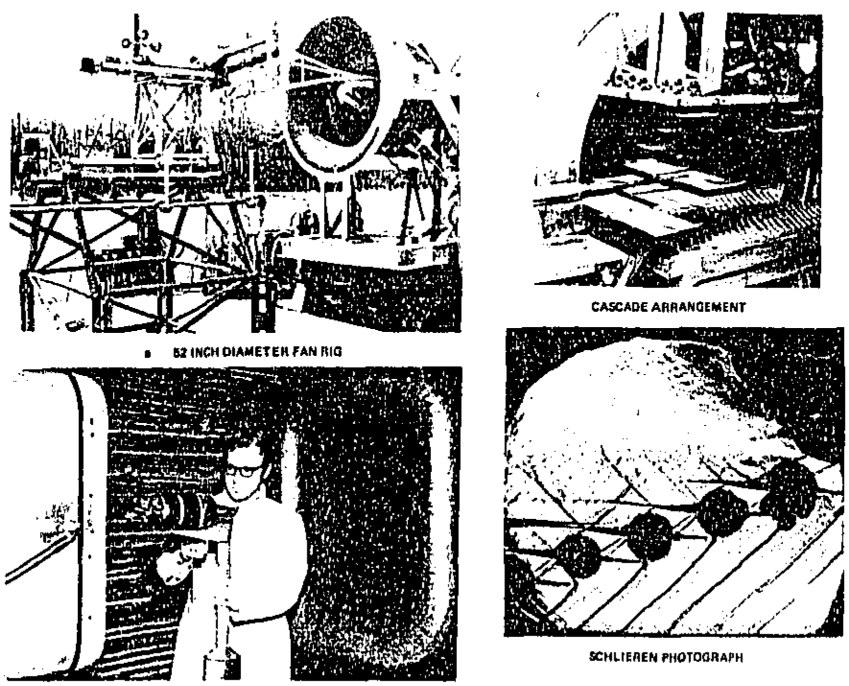
The last stage in fan noise transmission involves radiation into the far field of the patterns spinning at the duct faces. Comparatively little control of this part of the noise chain is possible, but studies of radiation from the spinning duct patterns have led to understanding of the complex directivity patterns of fan noise. On axis there is a minimum level, which then rises and falls in a series of lobes as one surveys in an arc around the engine. Angular locations of these peaks and valleys have been linked to the details of the pattern shapes at the engine inlet and exit ducts. Since the radiation directivity patterns govern both far field peak levels and duration in an overflight, consideration of this phase of fan noise emission completes the tracing of the important fan noise processes from their source to the ground observer.

FAN NOISE FACILITIES

Special facilities have proved indispensible in fan noise studies. These include both rigs for running isolated components and special instrumentation systems for recording and processing test information. Figure 4 presents a small sample of the variety of equipment used.

Figure 4a shows the inlet to a 52 inch diameter single stage research fan rig in an outdoor noise test facility that allows far field surveys to be made of inlet and discharge noise. Prominent in the foreground is a probe microphone system with axial, radial and circumferential traversing capability. It is by means of such traverse equipment that the complicated spinning patterns of fan interaction and combination tone noise are mapped out.

The effects of forward flight speed upon fan noise generation, propagation and radiation are being explored (Figure 4b) in an acoustic wind tunnel in the open jet of which is installed a miniature working model of a fan powered nacelle driven by a high pressure air turbine. Anechoic construction of the room containing the working section enables free field measurements to be made reliably. In spite of its size, the small (4.2" diameter) fan can be driven at supersonic tip speeds, and multiple pure tones are clearly produced.



CASCADE ARRANGEMENT

a 52 INCH DIAMETER FAN RIG

SCHLIEREN PHOTOGRAPH

b POWERED NACELLE IN ACOUSTIC RESEARCH TUNNEL

c SUPERSONIC CASCADE TUNNEL

Figure 4 Fan Noise Research Facilities

Studies of multiple pure tone noise involve details of the blade shock wave patterns at supersonic speeds. To visualize these patterns, optical methods such as schlieren photography have been employed in supersonic cascade wind tunnels as shown in Figure 4c. Work is underway to conduct similar optical programs in special rotating fan rigs.

DISCRETE BLADE PASSAGE NOISE GENERATION

Discrete frequency noise at blade passage frequency and its harmonics was one of the most conspicuous noise characteristics of early turbofan powerplants. These fans employed inlet guide vanes, and it was found that the cutting of the vane wakes by the rotor blades was a major noise source. Current high bypass ratio engines dispense with inlet vanes, but the shedding of wakes from the rotor blades into the downstream exit stators can be a similar source of blade passage interaction noise. It remains to show here how the previously described interaction patterns are generated.

Figure 5a portrays in a developed view a row of moving blades shedding wakes into the vanes of a downstream stator. As a wake passes by a stator vane, both the effective angle of attack and velocity change, producing a fluctuating lift distributed over the vane surface. The lift fluctuates with a base frequency equal to the blade passage rate and constitutes what is called a dipole source of sound. In the illustration equal numbers of blades and vanes are employed, so that when one vane is about to receive a blade wake all the other vanes are at the identical point in their cycles. As shown, the stator vanes would thus produce an array of sources pulsating in unison. In practice, however, the numbers of blades and vanes are different, so that while one vane would be on the verge of intercepting a blade wake, another vane might just have recovered from a wake passage. Generally there is a sequential phasing of the wake interaction events around the stator assembly. This phasing forms a pattern that sweeps circumferentially around the vane array rather than in a synchronous, umbrella-like oscillation. The number of complete cycles in the pattern has been derived analytically in terms of blade and vane number, and the existence of such patterns has been frequently confirmed on a variety of rigs and engines.

However, the plausibility of this interaction pattern effect is most directly perceived by use of a simple optical analogue called the Moiré effect. If the stator assembly is represented by an array of radial spokes drawn on a stationary background and the rotor blading is similarly represented on a clear sheet of plastic, when the two patterns are overlaid the interference of light and dark regions will produce a pattern of resultant intensity that suggests the blade-vane interaction effect. Figure 5b is a photograph of two such arrays, one containing 46 spokes and the other having 48. The resulting 2 (= 48-46) cycles of intensity variation is conspicuous. In a live demonstration one wheel can be turned slowly about a common concentric axis. The two-lobed interference pattern will be observed to spin comparatively rapidly. In fact, if the 46 spoke pattern is turned the interference pattern will turn 23 (= 46/[46-48]) times as fast as the simulated rotor in the opposite direction. The number of lobes or cycles of variation in the interference pattern and its rotational speed can be changed by changing the number of vanes in the stator. Parenthetically this does not change the blade passage frequency, just the associated acoustic mode structure. Figure 5c shows the Moiré pattern for 46 rotor blades in conjunction with 60 simulated stators. Though the pattern is not as well defined in the illustration as the first case, 14 interference cycles may be counted. If, in a working model, the plastic 46 spoke rotor overlay is now turned, the interference pattern will be observed to turn more slowly than in the first example. In fact it turns at a multiple of rotor speed given by the expression $46/(46-60) \approx -3$, the minus sign indicating a backward motion.

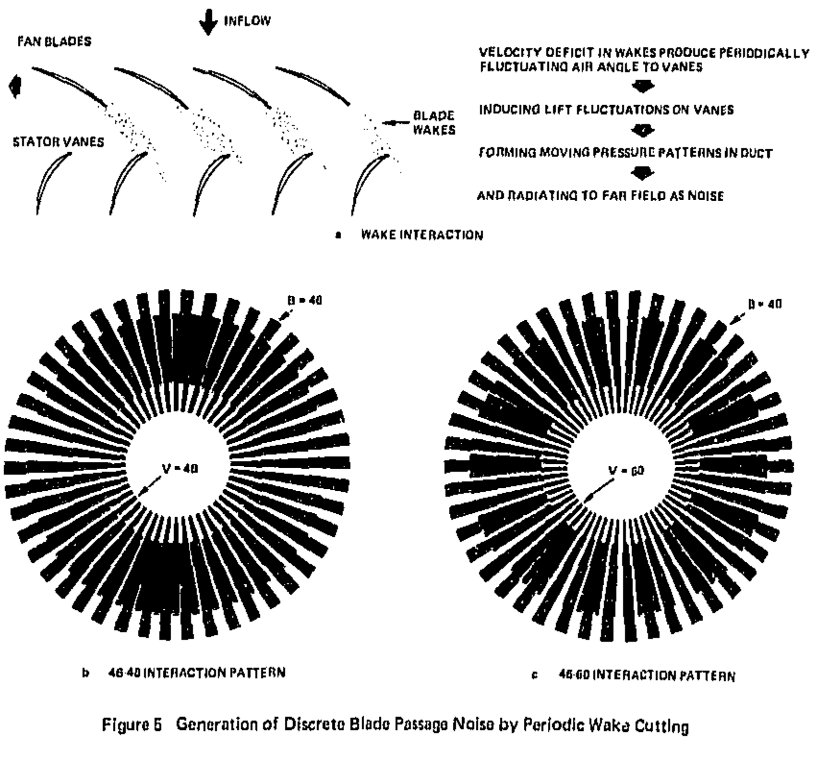


Figure 5 Generation of Discrete Blade Passage Noise by Periodic Wake Cutting

It will be recalled that the requirement for a spinning acoustic pattern to propagate in a duct is that its tip speed at the wall be supersonic. These optical diagrams make it quite clear that supersonic interference patterns can be generated by interaction effects when the rotor itself is subsonic. They also show how the pattern can be slowed down by increasing the number of stator vanes without changing rotor speed. When the interference pattern itself drops below sonic speed, its strength decays rapidly with axial distance from the generating zone, and noise due to this interaction is essentially eliminated. These features of interaction noise have been exploited in the design of modern turbofan powerplants. While major interaction noise sources have been eliminated, discrete frequency noise still exists due to non uniform inflow. However, there has been a marked improvement over early turbojet and turbofan discrete frequency noise.

BROADBAND NOISE GENERATION

Sources of fan broadband noise are suggested in Figure 6. As with discrete interaction noise, it is generated by airload fluctuations on the blade and vane surfaces. In this case these fluctuations are random in time instead of periodic. Turbulence in the air entering the rotor, turbulence from the rotor impacting the stator, and boundary layer turbulent fluctuations all constitute random broadband noise sources. Quantitative understanding of this category of noise is in a primitive stage compared with the discrete frequency case due to the inherently greater complexity associated with describing turbulence and random processes generally.

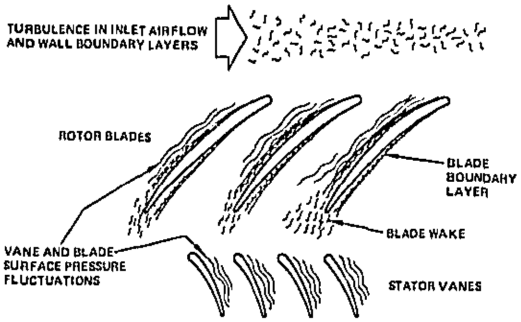


Figure 6 Broadband Noise Generation Due to Random Blade Load Variations

Simplified analyses of one part of the problem, the interaction of a rotor with incoming turbulence, have provided insight into the shape of the broadband spectrum. If the axial scale of the turbulence is small compared to blade gap, the correlation between lift fluctuations occurring on neighboring blades will be low, and the resulting noise versus frequency distribution will be relatively flat. On the other hand, if the axial scale of inflow turbulence is large, corresponding to long streaks, several blades will successively cut through each such non uniformity, generating bursts of noise at blade passage frequency. The resulting spectrum will contain peaks centered around blade passage frequencies, and the peaks will become progressively sharper as the turbulence axial scale is increased. Consequently, in controlling broadband noise, it is clearly important to insure that the flow passages are designed to produce the smoothest possible flows.

COMBINATION OR MULTIPLE PURE TONE NOISE

As described previously, combination tone or multiple pure tone noise is associated with the shock waves produced by rotors operating at supersonic tip speeds. It is helpful to consider two cases; first, an ideal rotor containing perfectly spaced identical blades, and secondly, an actual rotor assembly incorporating small blade-to-blade deviations in shape and orientation.

Figure 7a portrays the shock wave structure attached to the leading edges of an ideal supersonic rotor. On the right hand side of the figure is a representation of the pressure-time history that would be detected by a probe microphone in the inlet duct. A repetitive sawtooth pattern results as the succession of shock waves passes by.

In a relatively short distance from the rotor, the amplitude of these shocks will have attenuated due to non-linear effects to what is called an acoustic disturbance or Mach wave. This symmetric wave pattern propagates in a spiral path out the inlet since its circumferential wall speed is supersonic. It would be detected in the far field as a sharp discrete noise at blade passage frequency and its harmonics.

In practice, what happens is significantly different. Actual rotors contain small blade-to-blade differences due to manufacturing deviations and service wear. These variations are usually small and this has been confirmed by moving an inlet duct probe microphone up close to the blade leading edge plane of test rotors. In this very close location, the shock wave structure of a normal production rotor is extremely uniform. Correspondingly, the spectrum of the noise is clearly dominated by blade passage harmonics. However, there are detectable small variations in shock strength from one blade to another. The importance of these normally negligible deviations is that shock wave behavior is nonlinear: the higher amplitude shocks propagate a little faster than their lower amplitude neighbors. Consequently, the uniform pattern existing very close to the rotor becomes warped as distance from the rotor is increased. Despite the small variation of the initial shock amplitudes, there is sufficient difference in their non-linear behavior to produce a marked non-uniformity in amplitude and spacing of the pattern within a short axial distance of the rotor.

Figure 7b portrays the phenomenon just described. To the right of the figure is shown the pressure-time trace recorded by a probe microphone placed a few chord lengths ahead of the rotor. Here, the pressure irregularities are conspicuous, in many cases there being no visible evidence of blade periodicity. It has been observed that the pattern repeats faithfully with every turn of the rotor, so that the spectrum of the resulting sound will have a fundamental frequency of shaft rotation speed rather than blade passage frequency. Since the pattern contains many sharp irregularities, a large number of harmonics result, giving rise to the multiple pure tone noise descriptive term.

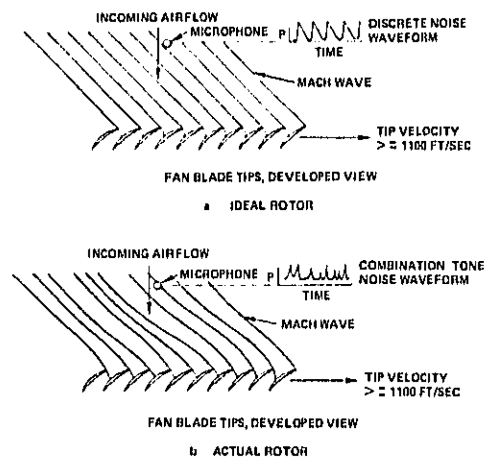


Figure 7 Multiple Pure Tone Noise at Supersonic Tip Speeds

It might be supposed that it would be relatively simple to eliminate MPT noise by sufficiently close control of rotor construction, but this has not proved possible. Even if a sufficiently perfect rotor could somehow be produced, unequal blade wear in service would soon cause enough irregularity to develop MPT noise. Nor would it be desirable to have such a perfect rotor; its sound would be the shrill whine of blade passage frequency which is much more disturbing than the distributed tonal quality of MPT or combination tone noise. Reduction of this noise component is being explored with blading designed to reduce the strength of all the blade shocks and by means of sound absorbing wall liner constructions. Wall lining has proved quite successful in reducing multiple pure tone noise.

FAN NOISE CONTROL

An understanding of these basic principles of fan noise has been used to significantly improve the characteristics of modern turbofan engines now entering service and planned for future use. Inlet guide vanes have been eliminated and the spacing between rotating and stationary blade rows has been increased. The well-known effect of increased rotor-stator spacing is shown in Figure 8. Figure 9 compares the fan geometry of a modern high bypass ratio single stage fan with an earlier generation low bypass ratio two-stage fan engine. The noise reduction features include: removal of inlet guide vanes, reduced fan tip speed, elimination of second stage fan and its interstage stator, increased rotor-stator spacing, and selection of stator vane number to control interaction effects.

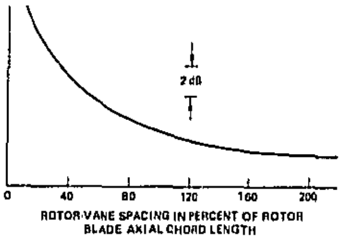


Figure 8 Effect of Fan Exit Guide Vane Spacing on Fan Discrete Noise

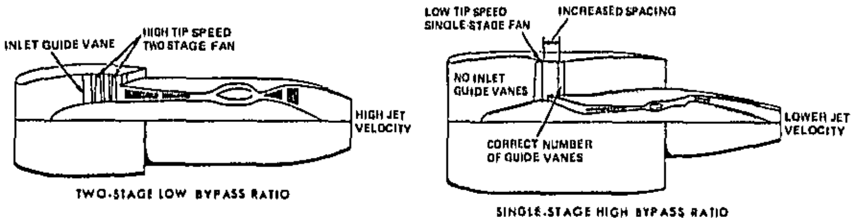


Figure 9 Comparison of Current and Early Turbofan Designs

FAN NOISE PREDICTION

This review of fan noise fundamentals has identified the sources of the several types of fan noise and has indicated measures that can be taken in powerplant design to reduce noise. However, the problem of predicting actual levels of noise produced by a specific design configuration contains many uncertain elements. Two general routes are available. Noise levels may be calculated from the results of theoretical analyses of the generation, duct propagation, and radiation phases or they may be obtained empirically from scale model test data. The latter route is usually more reliable. There are so many steps in the theoretical calculations involving unconfirmed or doubtful assumptions at several stages that the reliability of the calculated end product is questionable. This argument does not indicate that the theoretical aspects of noise generation should be ignored. Theory has suggested many useful concepts for experimental evaluation, several of which are currently in use. But a great deal of theoretical work and experimental verification remains to be done before reliable noise level predictions can be calculated on a paperwork basis.

Figure 10 is intended to portray the alternative theoretical and empirical routes to prediction of one type of fan noise, blade-vane interaction noise at blade passage frequency. Both processes start with given information about the fan geometry and operating conditions. The vane and blade numbers and rotor speed are such that interaction noise propagates. On the left hand part of the diagram are shown 3 sources of this noise: impacting of velocity defects from the rotor into the stator, peturbation of the stator vane loads by the passage of the rotor blade potential fields, and the effect upon the rotor blade loading due to its cutting the upstream potential field of the stator vanes. These load fluctuations must be calculated at several spanwise locations. From the rotor fluctuations can be calculated the acoustic field generated in the duct and its propagation through the duct can be worked out. The acoustic field generated by the stator has to pass upstream through the rotor before reaching the inlet ducting. Calculation of the transmission process through the rotor is a highly involved, completely unchecked procedure. The rotor and transmitted stator fields combine in the inlet to produce a resulting pattern at the inlet face. This pattern involves significant radial variations that add to the complexity of the far field directivity pattern which is the final stage of the calculation. It may be appreciated that the outcome of such calculation procedures are subject to considerable uncertainty.

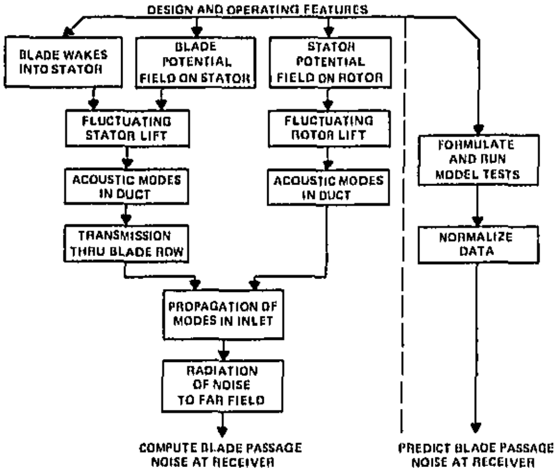


Figure 10 Fan Noise Prediction Sequences - Forward Radiated Blade Passage Tone Noise

On the other hand, test results from an appropriate model of the fan geometry are relatively straightforward to obtain. By normalizing the data, fairly reliable predictions of blade passage noise as a function of operating parameters can be made. Eventually, as more data on a greater variety of configurations are compiled and as theoretical aspects of the processes are confirmed, it will be possible to combine both theoretical and empirical methods to establish reliable prediction methods for new fan configurations.

FAN NOISE MECHANISMS AND CONTROL II

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Summary and Conclusions

The sources of noise radiation by fans and other rotating aerodynamic devices have now become reasonably clear. At low speed, that is, below sonic tip velocity, harmonic noise levels from the fans are dominated by the effect of distorted inflow into the fan. Recent experiments show a one to one correspondence between measured aerodynamic distortions going into the fan, and measured acoustic output from the fan. Broad band noise radiation from the fan, at least at low frequencies, can also be correlated directly to turbulent input flow into the fan.

Thus the fan can be regarded as a machine for converting unsteady flow into noise, and its aero-acoustic transfer function measured, both for discrete frequency and broad band noise. This fan aero-acoustic transfer function has been found to agree well with theory, which can thus be used for prediction and control of fan noise with reasonable confidence.

At higher frequencies, possibly above about $fc/U = 4$, a self-interactive source, independent of inflow, can be important. This appears to be associated with the blade tip. Modifications to the tips can result in noise reductions at these frequencies of well over 10dB in some cases. This source may be particularly important for large or low speed fans.

The results appear to be generally applicable to noise control of all rotor systems except those moving at transonic speeds or greater, which are outside the scope of this paper. Control of fan noise at source is therefore a matter of minimising non-uniformities in the fan inflow and, particularly for a low speed or open rotor, choice of an optimum tipshape.

Some Recent Experiments

A series of experiments have been carried out on a low speed fan mounted in an anechoic room at Loughborough University of Technology - details are given in Table 1. For most of the tests the fan was run in a seven-bladed configuration at speeds from 750 to 2750rpm. The fan axis coincided with a center line of the room, with the fan being 2.5m from one end of the room.

Noise measurements were made with standard Bruel and Kjaer equipment. In addition a series of aerodynamic measurements have been made using DISA hot wire anemometers operated in the constant temperature mode. The fluctuating velocity input into the fan was measured in two ways: by stationary hot wires mounted in front of the rotor, and by rotating hot wires moving with the rotor. The signal was taken from the probe through a commercial slip ring unit to the hot wire balancing electronics.

A major objective of the work has been to compare the fluctuating aerodynamic input to, and the acoustic output from, the rotor. Theory suggests that the noise output from a low speed rotor is due, predominantly, to the interaction of inflow distortions with the rotor. Now the rotating hot wire will measure these spatial distortions directly, converting them to a time varying signal. Thus a frequency analysis of this hot wire signal will give the levels of the circumferential harmonics of the rotor distortion. If the theoretical model is correct these should have a direct relation to the noise output of the rotor. Theory shows that calculation of noise at off-axis locations requires evaluation of an infinite series of Bessel functions. But on-axis the modulated Doppler shift effects of rotation disappear and there should be essentially a one to one relation between inflow distortion harmonics and noise output harmonics. Thus the noise measurements reported here have been confined to the on-axis case, allowing this important simplification.

As has been mentioned, the test rotor was mounted inside an anechoic room. As a result, under steady conditions the rotor induced considerable recirculating flow. This was used as the base unsteady inflow test condition. However full recirculation took several seconds to build up and the onset of recirculation could be observed to coincide with a pronounced increase in noise level. This meant that each fan configuration was automatically tested in two inflow conditions: before, and with full recirculation. The effect is shown in Figure 1. The solid curve shows a narrow band spectrum of the fan noise before the onset of recirculation, while the dotted curve shows the same spectrum after the recirculating flow has been established. Note that both spectra consist of a series of tones at the blade passing frequency (140Hz in this case) and its harmonics over a broad band background. The effect of recirculation is to raise and broaden the discrete frequencies and also to raise the lower frequency part of the broad-band spectrum. But note particularly that the higher frequency part of the spectrum (above about 5000Hz) is unchanged by the presence of recirculation. This point will be taken up again later.

Figure 2 shows a typical stationary hot wire spectrum. This spectrum was measured about one third of a chord in front of the fan and the presence of significant tones at the blade passing frequency and its harmonics is clear. Presumably these are associated with the potential flow field of the fan. However the rotating hot wire gives a rather different result, as shown in Figure 3. Here the harmonics appear at multiples of the rotational speed, with no particular effects at blade passing orders. This is expected, because all harmonics would be anticipated in the distorted inflow. Note that the harmonic levels again increase with the presence of recirculation.

According to theory, sound harmonics which are not integral multiples of blade number will cancel, so that only every seventh harmonic of the inflow field will combine to radiate sound. Direct comparison of the aerodynamic input and sound output can now be made, and by subtracting the aerodynamic harmonic levels from the sound harmonic levels the aero-acoustic transfer function of the fan can be determined. This is shown in Figure 4. If theory is correct the measured aero-acoustic transfer function should be identical for the same fan, whatever the inflow. It will be seen that these preliminary measurements do give the same result for both circulating or non-recirculating flow, at least for the lowest harmonics. The apparent fall-off of the circulating case at the higher harmonics is thought to result from interference from the background turbulence levels, but is the subject of further investigation. In addition, a theoretical curve is shown. This estimate is based on the author's theory for rotor sound radiation, and includes an estimate of the blade fluctuating forces based on Sears theory. Agreement is clearly most encouraging.

Investigation of the broad band case has also been made and it was found that for the one case studied so far broad band noise levels could be predicted theoretically from the inflow turbulence levels to an accuracy of better than 1dB.

It will be recalled that the high frequency part of the noise field was unaffected by inflow conditions. Presumably it must be generated by a self-interaction of the rotor. This has been traced to the fan tips. Several experiments have been made confirming this, and it appears possible to control the noise level at these frequencies by tip shape modifications.

The effect of one modification extensively tested is shown in Figure 5. This modification involved clipping away part of the outer trailing edge of the rotor blade. It will be seen that the low frequency parts of the noise are virtually unaffected. This is to be expected because they are inflow dominated. But the high frequency parts of the sound field are reduced by over 10dB. The effect is a strong function of tip incidence, but further tests have shown that it can be found at lower frequencies and for ducted as well as free rotors, and it appears probable that the effect will scale roughly as Strouhal number. Thus for large or low speed rotors such as presently being proposed for low noise applications, it appears that tip noise sources would peak at about 2000Hz and could dominate community noise levels.

Thus for the fan tested here it has been shown that the noise has three principal source components:

(1) discrete frequencies, which are governed by inflow distortion;

(2) low frequency broad band noise governed by inflow turbulence;

(3) high frequency broad band noise governed by the blade tips.

This is thought to provide a sound basis for any future work on fan noise control.

Acknowledgements

The author would like to acknowledge the help of Tony Whatmore and Charlotte Whitfield in these experiments which were jointly supported by grants from the National Aeronautical and Space Administration and the British National Gas Turbine Establishment.

Number of blades:	2, 7 or 14
Hub diameter:	0.24m
Rotor Disc diameter:	0.66m
Blade Chord at Tip:	0.064m
Blade Chord at Root:	0.085m
Maximum Blade Thickness at Tip:	0.003m
Blade Tip Angles:	5^{o}, 10^{o}, 15^{o}, 20^{o}
Blade Root Angles:	25^{o}, 30^{o}, 35^{o}, 40^{o}
Speed range between 0 & 3000rpm approx.	
D.C. Motor of 7.5kW.	
½" B & K Microphone position -	variable on 2.14m radius from fan centre
Hot Wire location:	0.26m from axis
	0.02m from Blade L.E.

Table 1: Rig Parameters

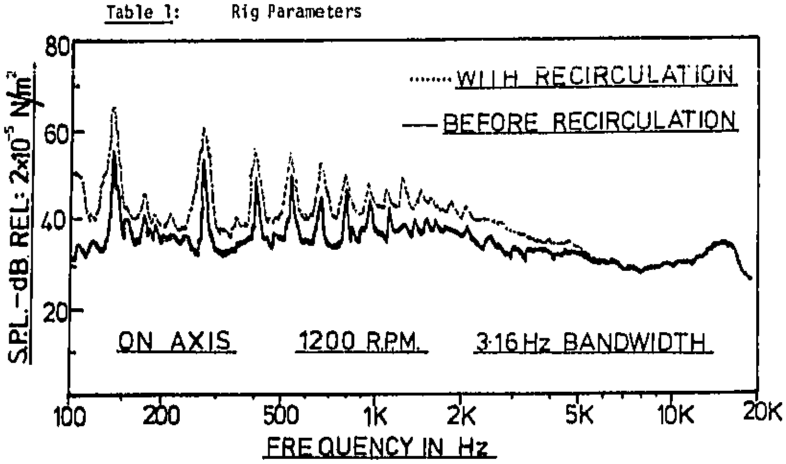

Figure 1: Fan Noise Spectra

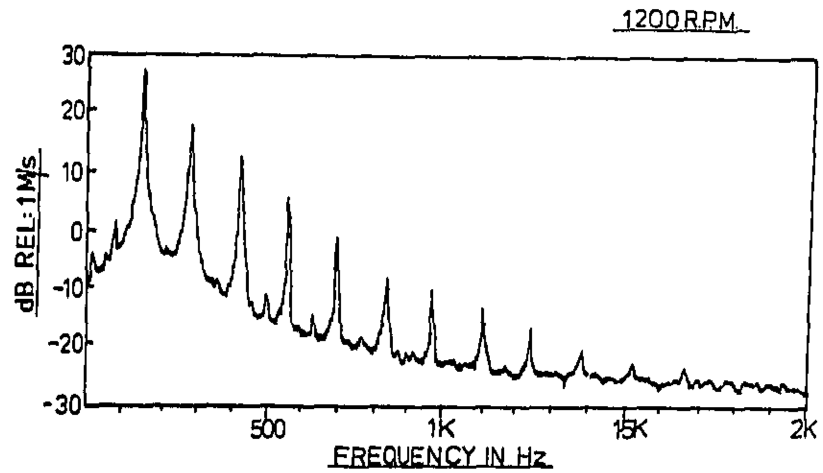

Figure 2: Velocity Spectrum - Stationary Hot Wire

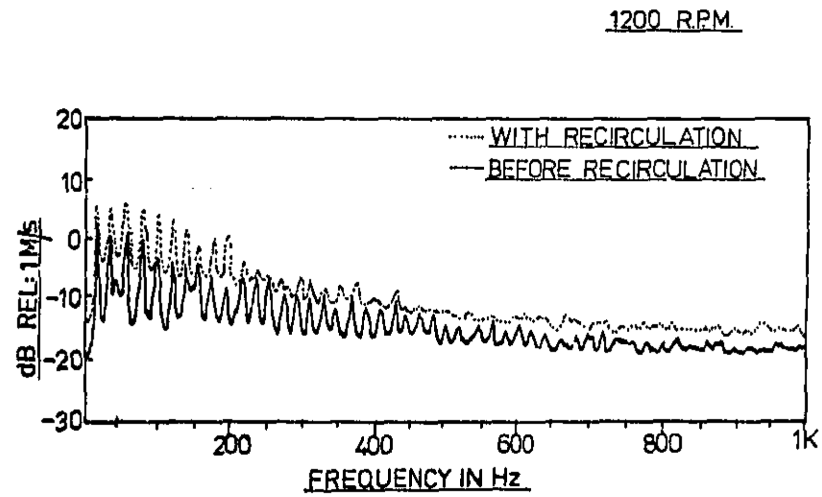

Figure 3: Velocity Spectrum - Rotating Hot Wire

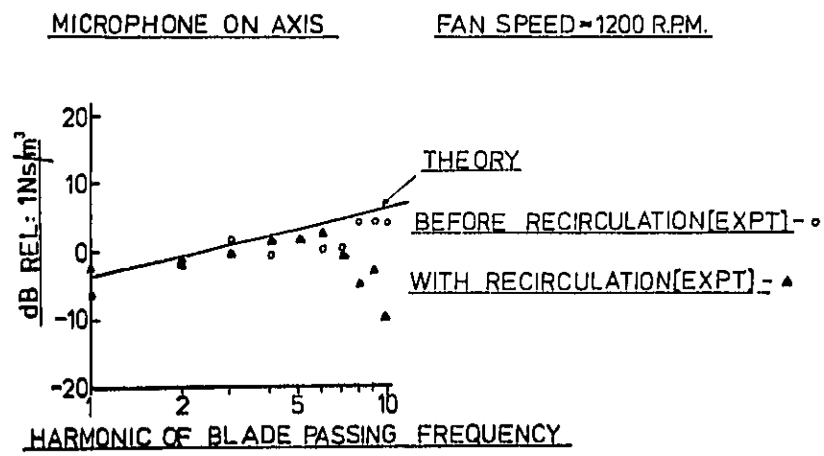

Figure 4: The Aero-Acoustic Transfer Function

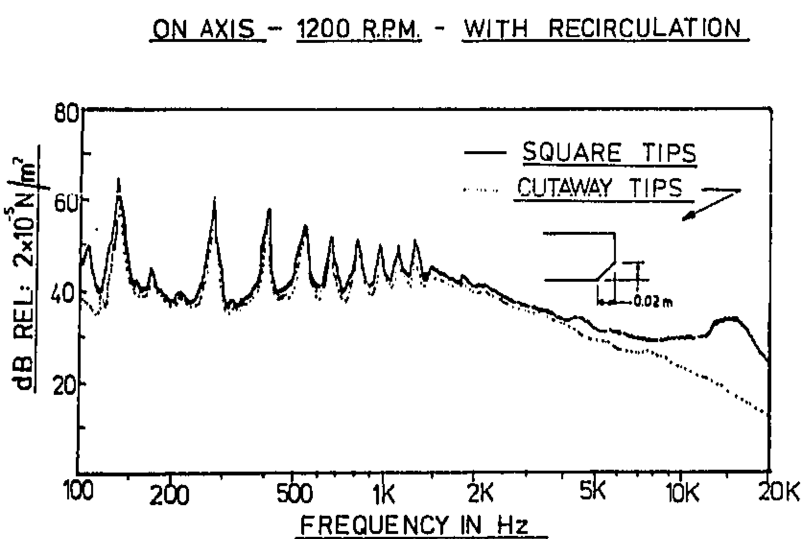

Figure 5: The Effect of Tip Shape on Fan Noise

NOISE SOURCE DISTRIBUTION IN SUBSONIC JETS

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SUMMARY

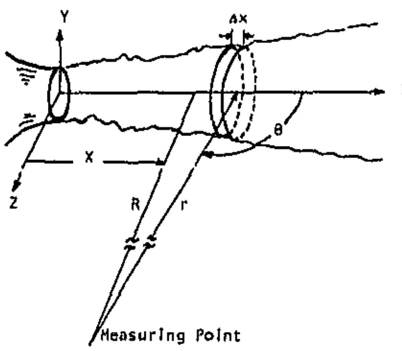
The feasibility of investigating the noise source distribution in subsonic jets quantitatively by means of a directional microphone system is presented together with some preliminary experimental results to substantiate the merit of such a technique.

INTRODUCTION

Before a rational approach to the jet-noise suppression problem can be taken, the nature and location of the important noise sources must be clearly understood.

Generally speaking, the past attempts addressed to this problem can be grouped into six categories: (1) theoretical prediction based on dimensional arguments together with some pertinent parameters of jet turbulence (Refs. 1,2,3,4,5), (2) near field pressure measurements (Refs. 6,7,8), (3) extrapolation from equal intensity contour survey (Refs. 9, 10), (4) shadow technique by physically blocking off the sound coming from certain portions of the jet (Refs. 11,12), (5) cross-correlation between two points in the acoustic field (Ref. 13) or between the noise source in the jet and the acoustic field (Refs. 14,15,16), and (6) imaging technique using a rigid plane surface (Ref. 17). However, these approaches are either too qualitative or too involved. A simpler method based on the imaging principle of a spherical concave reflector is presented here. This technique was first used by F. R. Grosche (Ref. 18) for some qualitative measurements in subsonic jets.

ANALYSIS



Let us model the jet acoustically as a line distribution of noise sources having strength equivalent to that of the elementary "slices" of the jet at the corresponding location. The justification follows from the fact that the reflector being used has a window function that is wider than the jet cross-sectional dimensions. The filtered mean-square acoustic pressure measured by the reflector located at (X,0,R) can be written as:

$$\frac{\overline{p^2}}{\rho c}(X,0,R;f) = \frac{\overline{E^2}(X,0,R;f)S^2}{B\rho c} = G(f)\int_{-\infty}^{\infty}\frac{\tilde{\Delta}(X,\xi;f)\,W'(\xi,\theta;f)}{4\pi r^2}\,d\xi \qquad (1)$$

where $\overline{E^2}$ = mean-square value of the voltage output from the microphone
S = sensitivity of the microphone
B = bandwith of the filter, being constant 50 Hz for this case
ρ = ambient density
C = ambient speed of sound
$\tilde{\Delta}$ = normalized window function of the reflector; $\tilde{\Delta}(0;f) = 1$
G = gain factor of the reflector = $\Delta(0;f)$
W' = noise power/unit length radiating in direction θ
over-bar denotes time average.

Implicit in this formulation is that sources located in the segment Δx are uncorrelated with neighboring ones within the resolution of the measurement. This assumption is supported by recent results (Ref. 19) and by the integration of our own results (Fig. 4).

Approximating r by R, Eq. 1 becomes

$$A\overline{E^2}(X;f) = \int_{-\infty}^{\infty} \tilde{\Delta}'(X-\xi;f)W'(\xi,\theta;f)\,d\xi \qquad \text{where } A = \frac{4\pi R^2 S^2}{B\rho C G(f)} \tag{2}$$

and we approximate the integral as a convolution because of the small angular displacements involved. To recover $W'(X,\theta;f)$, Fourier transform techniques can be used. It can be shown that

$$W'(X,\theta;f) = \frac{A}{2\pi}\int_{-\infty}^{\infty} \frac{\overline{\mathcal{E}^2}(\eta;f)}{\tilde{\mathbf{\Delta}}(\eta;f)} \exp(i\eta X)\,d\eta \tag{3}$$

where $\overline{\mathcal{E}^2}$ and $\tilde{\mathbf{\Delta}}$ represent the Fourier transform of $\overline{E^2}$ and $\tilde{\Delta}'$. The Fourier transform and its inverse were done numerically.

EXPERIMENT AND RESULTS

Based on a parametric study of the behavior of both spherical and parabolic reflectors (Ref. 20), a 3' diameter spherical reflector with a focal length of $26\frac{1}{2}$'' was chosen for the directional microphone system. The measurements were taken in an anechoic chamber of both a 1'' and 2'' diameter jet operating at subsonic Mach numbers up to 1. The reflector was located at a distance R of 82'' from the jet axis. Rather than moving the reflector parallel to the jet axis to obtain $\overline{p^2}(X,0,R;f)$, it was placed at two fixed locations, 4 and 9 diameters downstream and was rotated to scan the different portions of the jet. The window function and the gain factor $G(f)$ were calibrated from a point source.

Fig. 1 shows some typical measured source strength distributions in terms of the microphone output voltage and Fig. 2 depicts the normalized window function at two typical frequencies. Fig. 3 shows some typical results of $W'(X,90^\circ;f)/4\pi R^2$ as derived from Eq. 3.

DISCUSSION

Fig. 3 clearly shows that the higher frequency noise is generated by sources located closer to the nozzle exit as expected. If Eq. 3 is integrated with respect to X, one obtains

$$W_t(90^\circ;f) = \int_{-\infty}^{\infty} W'(X,90^\circ;f)\,dX = \frac{4\pi R^2 S^2\,\overline{\mathcal{E}^2}(0;f)}{B\rho C G(f)\,\tilde{\mathbf{\Delta}}(0;f)} \tag{4}$$

Note that

$$\frac{W_t(90^\circ;f)\,\rho C}{4\pi R^2} = \Phi_{90^\circ}(f) \tag{5}$$

is the power spectral density of the acoustic pressure measured at 90° position of the total jet. This result compared well with the overall noise measurement obtained by a single non-directional microphone (Fig. 4). Furthermore, the integration of Eq. 5 with respect to frequency gives the mean-square acoustic pressure of the total jet measured at 90°. The total power estimate based on these results together with the semi-empirical theory of Ref. 21 for the 1'' and 2'' jet at $M = 1$ was within 2dB of predicted values.

A comparison of the 5 kHz measurements of Fig. 1 with Fig. 3 demonstrates a numerical improvement of the spatial resolution of the directional microphone system at the lower frequencies.

While certain numerical approximations were made in Eq. 2, these are merely a convenience to simplify the initial analysis of the data. These initial results are sufficiently convincing to demonstrate that the present approach is an accurate quantitative method for noise source distribution measurements.

ACKNOWLEDGMENT

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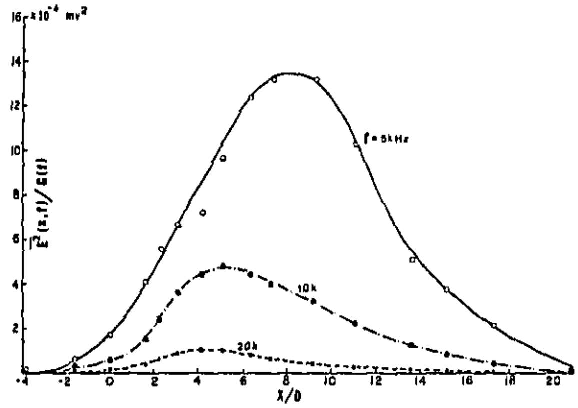

FIG. 1 MEASURED SOURCE STRENGTH DISTRIBUTION IN TERMS OF MICROPHONE OUTPUT VOLTAGE; D = 1", M = 1.

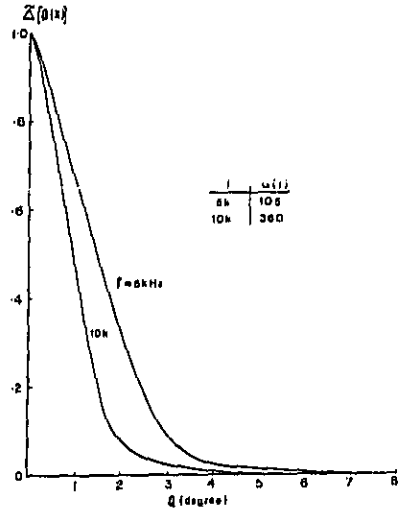

FIG. 2 TYPICAL WINDOW-FUNCTIONS OF THE REFLECTOR

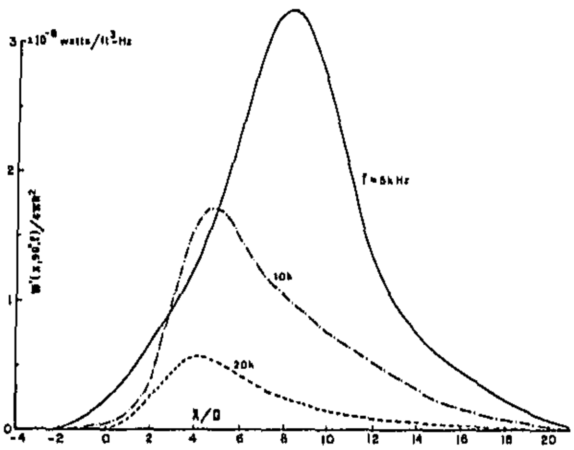

FIG. 3 ACTUAL SOURCE STRENGTH DISTRIBUTION IN SPECTRAL BANDS; D = 1", M = 1

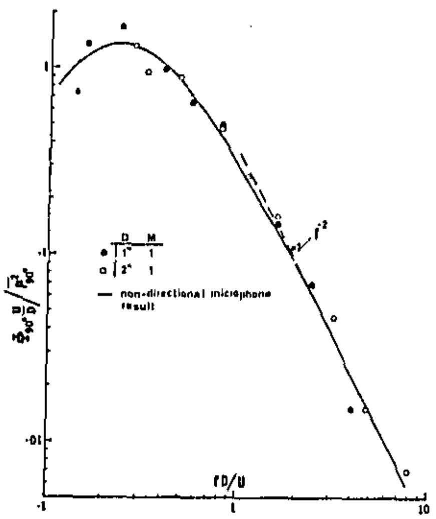

FIG. 4 COMPARISON OF THE NON-DIMENSIONAL PRESSURE SPECTRAL-DENSITY FUNCTIONS OBTAINED BY THE REFLECTOR METHOD AND THE SINGLE MICROPHONE METHOD

NOISE OF JETS DISCHARGING FROM A DUCT CONTAINING BLUFF BODIES*

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INTRODUCTION

The overall acoustic power output from the exhaust of jet engines is found not to scale with any simple power of jet velocity over a wide range of jet velocities.(1),(2) A narrow region may be found to fall close to an eighth power, but generally, the scaling appears to change almost continuously with jet exit velocity. This suggests that, if the dominant sources can be represented by various simple sources such as monopoles, dipoles and quadrupoles, their relative contributions to the noise output depend strongly on the jet velocity, and when added, the sum results in a complicated superposition. Attempts to disentangle the various effects resulting in the overall noise level and spatial distribution are being made by several investigators. Little attention has been given to the importance of noise sources inside the engine. The work leading to this paper focuses on the relative importance of sources internal to the engine as compared with the source region in the free jet.

Heat addition in a subsonic compressible flow is known to cause a reduction of both static and total pressure.(3) Non-steady heat addition would therefore cause static and total pressure fluctuations. The combustion process in a turbojet engine is a non-steady phenomenon in which the burning couples with the flow disturbances resulting in non-steady heat addition. It may be expected that this non-steady heat addition, since it causes both static and total pressure fluctuations, would result in increased turbulence and noise inside the engine. The work reported here is intended to show that disturbances in the flow inside the duct, such as that due to combustion, can result in a substantial increase in noise outside the duct. In order to demonstrate some of the noise producing characteristics of internal sources without introducing all of the many complexities associated with combustion, bluff cylindrical bodies are used to generate the internal noise. Moreover, bluff objects are always present in a jet engine duct system, and these can produce noise independent of the combustion process. The interaction of these bodies with the flow creates fluctuating forces in the flow resulting in dipole sources and generally increased turbulence, resulting in broadband noise with pole types to be determined. Combustion may be expected to generate monopole sources(4) or quadrupole sources(5) depending on the model chosen for analysis. This must be checked in controlled experiments. It is possible to show, by adaptation of Curle's(6) analysis, that the exhaust noise measured in the far field of a jet engine may have originated from a combination of sources inside and outside the engine, as illustrated in Fig. 1.

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PRESENT EXPERIMENTS

In the experiments reported here, noise originating from a ducted flow, exhausting as a jet into an anechoic chamber with and without bluff bodies inside the duct, is being studied. The bluff bodies act as a source of unsteady force fluctuations as well as turbulence inside the duct, producing exit plane unsteadiness.

A schematic drawing of the flow system is shown in Fig. 2. A 1/16 inch diameter cylinder is mounted 12 inches upstream from the exit of a 1 inch diameter duct, representing a blockage of only 8% of the flow area. The cylinder is mounted either vertically or horizontally. Microphone traverses are taken in the horizontal plane of the jet at a radial distance of 51 inches. Two B&K 1/8 inch microphones located at 1 and 10 inches from the duct exit are flush mounted on the duct in order to sense the internal noise. In the results reported here, the axis of each microphone is perpendicular to the cylinder mounted in the duct. Signals from duct microphones are used to determine the spectrum of sound and turbulence fields in the duct and for cross-correlation with signal from the far-field microphone. Auto- and cross-correlation functions are determined using a SAICOR-42 Correlator and Probability Analyzer. The output of the correlator is then fed to an analog-to-digital converter and the spectral (or cross-spectral) density function is determined via a Fast Fourier Transform Program.

RESULTS AND DISCUSSION

Directivity patterns, in a horizontal plane of measurements, of the overall sound pressure level for (1) the clean jet (no bluff body inside the duct), (2) the jet with cylinder mounted horizontally, and (3) the jet with cylinder mounted vertically, were determined for jet exit velocity ranging from 500 to 900 ft/sec. A typical directivity pattern is shown in Fig. 3 for a jet exit velocity of 810 ft/sec. The vertical strut produced an increase of as much as 17 dB in the region between 60° and 90° from the jet axis, but only a little change near the axis. The horizontal strut produced similar effects, but with an increase up to 11 dB over the clean jet in the region between 60° and 90° from the jet axis. The effect of strut orientation and jet velocity on the overall sound pressure level in the far field at a microphone position of 90° to the jet axis is shown in Fig. 4. As expected, the clean jet data follow very closely the U^8 law predicted by Lighthill. The effect of the vertical strut at this angular position is dominant over the entire range, with a maximum of 25 dB above the sound pressure level of the clean jet alone at a jet exit velocity of 700 ft/sec. The sound pressure level for the jet with horizontal strut progressively increases with jet velocity, but the increase over the sound pressure level of the clean jet is a maximum between 600-700 ft/sec.

A 1/3-octave band spectrum of the sound pressure level in the far field was obtained for jet velocity ranging from 500 to 900 ft/sec. Typical comparison between the spectrum of the noise from the clean jet, jet with horizontal strut, and jet with vertical strut is shown in Fig. 5 for a jet velocity of 800 ft/sec. and microphone angular position of 75°. Little difference between the spectra can be observed below 2000 Hz. At higher frequencies, the effect of the strut seems to add both broad band and discrete tones to the spectrum of noise. A dominant peak is clearly identifiable at about 25 KHz in the spectrum of the noise with vertical strut. It should be mentioned here that the spectra of the noise signals from the duct microphones for horizontal and vertical struts were the same under all running conditions. The differences in the far field noise spectra for the cases of horizontal and vertical struts are due to the differences in orientation of the dipole sources inside the duct relative to the plane of measurements. When the strut in the duct is in the plane of measurements, only axial dipole sources contribute to the far field sound. When the strut is perpendicular to the plane of measurements, both axial and lateral dipole sources contribute to the far field noise. The far field directivity pattern in each case, however, depends also on the exciting frequency, flow Mach number, and duct modes excited by the strut.

The strong effect of the lateral dipole sources as evidenced Figs. 4 and 5 indicates that the contribution of the flutuating lift on the strut can not be neglected, as suggested by Gordon(7) in a related study.

For the case of a vertical strut the decrease in far-field sound pressure with jet velocity above 800 ft/sec, Fig. 4, was also observed for all angular positions between 30° and 90°. This puzzling trend may be explained by considering the magnitude of the unsteady lift force on the cylinder. The oscillating lift force per unit length of the strut is proportional to $\frac{1}{2}\rho V_j^2 d_r$. The constant of proportionality (lift coefficient), found experimentally by various investigators[8], lies in the range between 0.5 and 2.0. The actual value of the lift coefficient in each experiment depends, among other factors, on the Reynolds Number and on the degree of turbulence of the upstream flow. (The Reynolds Number in these tests is between 1 and 2×10^4.) In this Reynolds Number range, it is known that, the higher the degree of upstream turbulence, the lower the value of the lift coefficient. In the flow system used in this study, it is expected that the degree of turbulence inside the system upstream of the strut increases with jet exit velocity. Therefore, it seems plausible that above 800 ft/sec, the effect of increased turbulence upstream of the strut more than compensates for the effect of increased jet velocity on the V_j^2 factor. The net result would be a reduction in the amplitude of the unsteady lift force on the strut and hence a reduction in the radiated sound pressure level in the far field. This behavior is not exhibited for the case of the horizontal strut because the lateral unsteady force in that case does not contribute to the far-field sound pressure level.

Auto-correlation and spectral density functions for the far field noise at an angular position of 75° were obtained for the case of jet with vertical strut over the entire range of jet exit velocities studied here. For each jet exit velocity the spectrum has a predominant peak about a frequency given by $f = 0.17\frac{V_j}{d_r}$, where V_j is the jet exit velocity and d_r is the strut diameter. A Strouhal number of 0.17 reported here is in agreement with the accepted value of about 0.2 previously reported in the literature.

Typical cross-correlation and cross-spectral density functions between the far field noise signal at angular position of 75° and the noise signal from the microphone in the duct located 1 inch upstream of duct exit are shown in Figs. 6 and 7 respectively. The cross-correlation diagram has a peak at a time delay between 3.7 and 3.8 milliseconds. This time delay is very close to the time taken by sound waves to travel between nozzle exit and far-field microphone, which is 3.76 milliseconds. This result indicates that, in addition to the aeolion tone, broad band noise generated inside the duct is also radiated to the far field. The cross-spectral density diagram, Fig. 7, shows the dominant peak frequency at about 23 KHz and other spectral components on both sides of the peak frequency. Again, this frequency corresponds to a Strouhal number of 0.17 based on the strut diameter and jet exit velocity. In addition to the peak at this frequency, some of the spectral density diagrams obtained in this study indicated existence of a second-harmonic component but at a much lower spectral density level.

CONCLUSIONS

Relatively small size bluff bodies in flow ducts can produce large increase (up to 25 dB in this study) in noise level over a wide range of jet velocities. The effect of the bluff body is to change both the directivity pattern and the spectrum of the noise measured in the far field. Aeolion tones are produced due to the shedding of vortices downstream of the bluff body. For a cylindrical body the frequency of the tones corresponds to a Strouhal number of 0.17 based on body diameter and jet velocity. The relative contributions of the axial and lateral dipoles acting on the bluff body depend on the orientation of the bluff body relative to the meridian plane of measurements. More noise raidates to the far field when the axis of the bluff body is perpendicular to the plane of measurements than when the axis is in the plane of measurements. With the present understanding of the mechanisms and characteristics of noise generation due to flow-bluff body interaction in ducts more complex experiments involving combustion processes may be conducted with the possibility of identifying and evaluating the relative contribution of each of the internal noise sources.

ACKNOWLEDGMENTS

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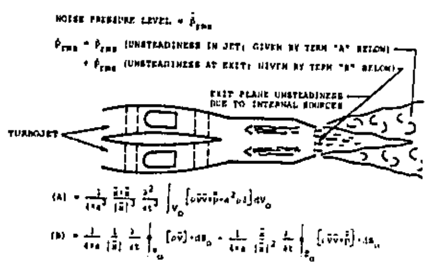

Figure 1 Sources of Noise in a Jet Engine Exhaust

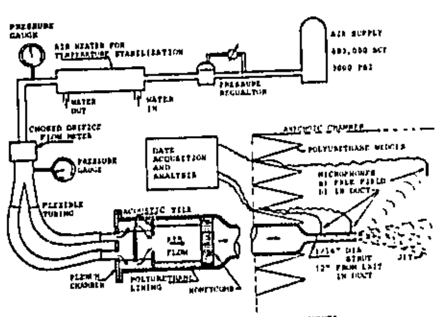

FIGURE 2 SCHEMATIC OF FACILITY FOR COLD FLOW NOISE EXPERIMENTS

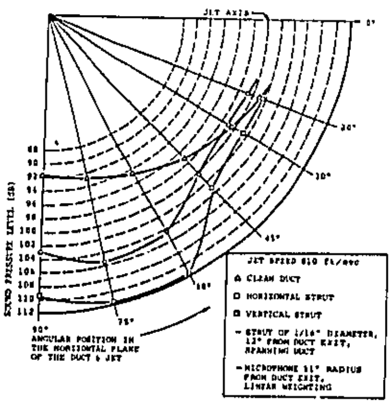

FIGURE 3 EFFECT OF INSERTED STRUT ON JET NOISE: INCREASE IN SOUND PRESSURE LEVEL AND SHIFT IN DIRECTIVITY

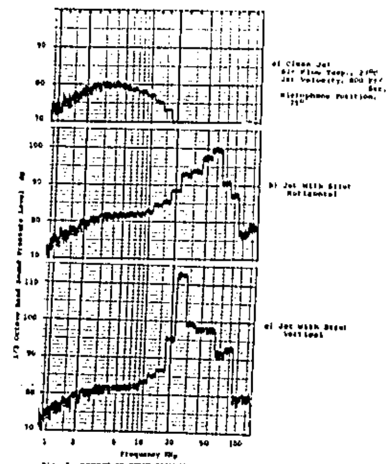

Fig. 5 EFFECT OF STRUT ORIENTATION ON 1/3 OCTAVE BAND SPECTRA IN THE FAR FIELD.

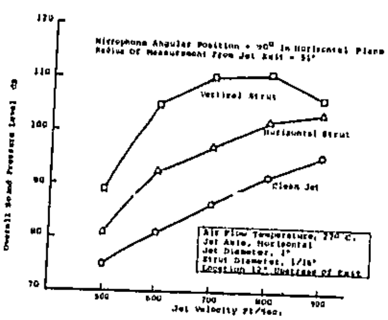

Fig. 4 EFFECT OF STRUT ORIENTATION ON SOUND PRESSURE LEVEL MEASURED IN THE FAR FIELD

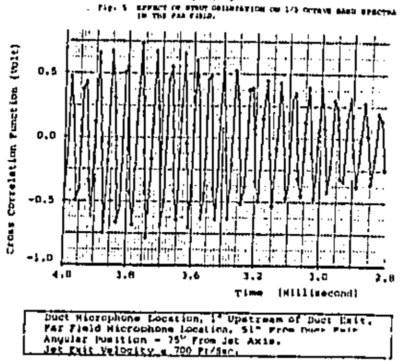

Fig. 6 CROSS CORRELATION FUNCTION BETWEEN SIGNALS FROM DUCT MOUNTED MICROPHONE AND FAR FIELD MICROPHONE

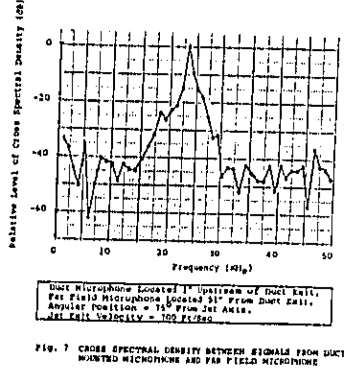

Fig. 7 CROSS SPECTRAL DENSITY BETWEEN SIGNALS FROM DUCT MOUNTED MICROPHONE AND FAR FIELD MICROPHONE

EFFECT OF SPANWISE CIRCULATION ON COMPRESSOR NOISE GENERATION

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ABSTRACT

A theoretical method for determining fan rotor noise from the blade design characteristics is presented. The method uses a three-dimensional flow model in a cascade and combines the work of Scholz with the Prandtl lifting line theory to obtain the integral equation for circulation distribution. The effect of the casing enclosing the cascade is introduced using the well-known method of images. Comparisons of the present method with limited experimental results show good agreement. The effects of gap width, hub-tip ratio, number of blades, rpm, camber, taper angle, and twist angle on sound power level are presented. It is shown that for a given design there is a critical twist angle which will produce minimum noise.

INTRODUCTION

During the last decade, research on fan compressor noise has brought a number of advances in the design of high performance turbofan engines that are quieter than their predecessors. Notable changes (for example on the JT9D engine) include the elimination of the inlet guide vanes, reduction from two to a single fan stage, low blade tip speed, optimum blade-to-vane spacing, and optimum number of blades and vanes. It appears that engines in future propulsion units will continue to increase in bypass ratio, thus reducing the primary jet exhaust noise. Another noise reduction feature in the future will be the increase in spacing between fan rotor and exit guide vanes. This means that one of the remaining problems will be the fan rotor noise.

The fan rotor noise differs from propeller noise principally in the number of blades, hub-tip ratio and the fact that the fan rotor is ducted, yet the classical theories developed for the propeller noise [1,2] have formed the basis of fan rotor noise studies. Of the three main rotor noise generation mechanisms (thickness noise, lift noise, and vortex noise) the dominating noise source appears to be the lift noise [3] which is caused by the thrust and torque distribution over the blades. Since the lift on a blade is a function of the steady circulation, a detailed knowledge of the circulation distribution is essential in finding the noise generated by a rotor. Methods published to date for the determination of rotor noise use the two-dimensional circulation and do not account for the spanwise circulation. For example, in a recent paper by Hulse and Large [4], the concepts developed earlier by Garrick and Watkins [2] in connection with propeller noise were used to predict the compressor noise field. In their mathematical model, the radial circulation distribution degenerated into a modified constant. In the work of Benzakein and Kazin [5,6,7] the two-dimensional circulation is applied at three points: hub, tip, and midspan. Although this may represent an improvement over the use of a modified constant, this curve-fitting method cannot be used to find any of the three-dimensional effects on rotor noise. The purpose of the present paper is to use a recently developed method of calculating the spanwise circulation [8] to obtain some of the three-dimensional effects on rotor noise.

ANALYSIS

Calculation of Noise Generation

The purpose of this section is to present a brief outline of the method used to obtain the noise generation and to relate it to the circulation. The approach here follows that of Benzakein [5] and Sowers [9] with some modifications to apply it to the present analysis. Only the case of sound propagation in a duct with the medium at zero velocity will be considered. The introduction of a moving medium is a simple extension and has been done by many authors [5,9,10,11]. Here it does not contribute to the analysis.

With the assumption that the classical linear acoustic wave equation holds, the solution for an annulus consisting of rigid walls and with no end reflections is (see Figure 1):

$$P = \left(\frac{1}{2}\rho a^2\right) \frac{M_o M_t}{2\sqrt{2\pi}} \frac{t}{R_o} \sum_{n=KN}^{\infty} \sum_{m=1}^{\infty} \frac{A_{nm}}{K} E \frac{R_n(\lambda_{nm} r)}{\eta_{nm}} \tag{1}$$

where

$$R_n(\lambda_{nm} r) = Y'_n(\lambda_{nm}) J_n(\lambda_{nm} r) - J'_n(\lambda_{nm}) Y_n(\lambda_{nm} r) \tag{2}$$

$$\eta_{nm} = \frac{1}{2\lambda^2_{nm}} [(\lambda^2_{nm} - n^2) R^2_n (\lambda_{nm}) - (\lambda^2_{nm} \nu^2 - n^2) R^2_n (\lambda_{nm} \nu)] \tag{3}$$

$$A_{nm} = \int_\nu^1 r\, f(r)\, R_n(\lambda_{nm} r)\, dr \tag{4}$$

$$f(r) = \frac{\Gamma(r)}{U_o t} \tag{5}$$

and where P = sound pressure, ρ = density, a = velocity of sound, t = blade spacing,
M_o = reference Mach number upstream of rotor (function of mass flow),
M_t = tip Mach number = wR_o/a, w = angular velocity of compressor stage,
K = integer, N = number of blades, ν = hub/tip ratio, Γ = circulation, and
U_o = reference velocity.

The λ_{nm} are calculated from the Eigenvalue equation

$$Y'_n(\lambda_{nm}\nu)\, J'_n(\lambda_{nm}) - Y'_n(\lambda_{nm})\, J'_n(\lambda_{nm}\nu) = 0 \tag{6}$$

The value of $\mu^2 = n^2 M_t^2 - \lambda^2_{nm}$ is first tested. If $\mu^2 < 0$, cutoff occurs and

$$E = e^{-n\sqrt{-\mu^2}\, z}$$

where z = duct length. If $\mu^2 > 0$, then $E = \cos(n\mu z)$.

Here the circulation is slightly modified from that of References 5,9 in order to account for the continuity in the radial direction. For parallel blades, t is a constant, and for radial blades, t is a function of the radius. The reference Mach number directly upstream of the rotor, M_o, is associated with the mass flow through the rotor. However, since it is assumed that the noise is propagating in a stationary medium, this assumption is valid for flow Mach numbers in the duct of up to 0.5 or so.

The dimensionless sound pressure is $\bar{P} = P/(\frac{1}{2}\rho a^2)$ (7)

and the sound power level is $W = \frac{\pi}{4} R_o^2 \rho a^3 \int_\nu^1 \bar{P}^2 r\, dr$ (8)

which in db re 10^{-12} watts is given by $10 \log_{10} W/10^{-12}$ watts (9)

Finally, the sound pressure level is given by $SPL = 20 \log_{10} \bar{P}/P_o$ (10)

Calculation of the Spanwise Circulation

The mathematical and physical details of obtaining the spanwise circulation are described in detail in Reference 8. Essentially the method combines the work of Scholz [12], which extends Glauert's theory of thin airfoils to profiles of cascades, with the Prandtl lifting line theory to obtain the integral equation for the steady spanwise circulation in a cascade.

It is shown in Reference 8 that in order to find the steady spanwise circulation it is necessary to evaluate

$$\sum_{j=1}^{\infty} b_{2j-1} \left[\sin(2j-1)\psi + (2j-1)\frac{e^{-i\beta}t}{2\pi L} \tanh \frac{\pi e^{i\beta}c}{2t}\, \mu(\psi)\right]$$
$$= \frac{U_\infty}{U_o} e^{-i\beta} \left\{-\frac{2}{\pi} \tanh \frac{\pi e^{i\beta}c}{2t} \int_0^\pi \frac{dy}{dx}\, d\phi + \left(1 - \tanh \frac{\pi e^{i\beta}c}{2t}\right) \sum_{n=1}^{\infty} a_n \frac{[1+(-1)^n]}{2} \tanh^n \frac{\pi e^{i\beta}c}{4t}\right\} \tag{11}$$

where

$$\mu(\psi) = \frac{\pi \sin(2j-1)\psi}{\sin\psi} + \int_0^{\frac{\pi}{2}} \frac{\cos(2j-1)\psi' d\psi'}{\cos\psi' + \frac{\xi}{L}} + 2\sum_{n=1}^{\infty} \left\{ \int_0^\pi \frac{(\cos\psi' - \cos\psi)\cos(2j-1)\psi' d\psi'}{(\cos\psi' - \cos\psi)^2 + \frac{m^2t^2}{L^2}\cos^2\beta}\right.$$
$$\left. + \int_0^{\frac{\pi}{2}} \frac{(\cos\psi' + \frac{\xi}{L})\cos(2j-1)\psi' d\psi'}{(\cos\psi' + \frac{\xi}{L})^2 + \frac{m^2t^2}{L^2}\cos^2\beta} \right\}$$

and

$$\cos\phi = -\frac{\tanh \frac{\pi x e^{i\beta}}{t}}{\tanh \frac{\pi c e^{i\beta}}{2t}}, \qquad a_n = \frac{4}{\pi}\int_0^\pi \frac{dy}{dx} \cos n\phi\, d\phi, \qquad \frac{\xi}{L} = -\left(2 + 2\frac{\delta}{L} + \cos\psi\right)$$

where β = stagger angle, ϕ = transformation angle for the two-dimensional case, ψ = transformation angle for the three-dimensional case, L = blade length, c = blade chord, U_∞ = free stream velocity, m = number of blade spacings, x = coordinate in the chordwise direction, ξ = coordinate in the spanwise direction, $y = y(x)$ defines the blade profile shape, and δ = blade tip clearance.

After the constants b_1, b_3, etc. are evaluated in equation (11), the circulation is

$$\frac{\Gamma}{U_o t} = \sum_{j=1} b_{2j-1} \sin(2j-1)\psi$$

<u>Numerical Computation</u>

A computer program is available from the authors which will calculate the sound power level or sound pressure level from the following input information:

N, ν, a, M_o, z, wL/U_o (dimensionless angular velocity), α, β_o, Φ, δ/L, d/c, $\Delta C/L$, C_r/L,

where C_r = root chord, ΔC = root chord − tip chord, d = maximum camber, Φ = twist angle, α = angle of attack, and β_o = root stagger angle.

After the input information has been completed, the functions β, c/t, t/L, U_∞/U_o and dy/dx are evaluated for different values of ξ. Since

$\cos\psi = -\xi/L$ and $C/L = C_r/L - \frac{\Delta C}{L}|\cos\psi|$, then, for radial blades

$$\frac{U_\infty}{U_o} = \sqrt{1 + \frac{w^2L^2}{U_o^2}\left(\frac{\nu}{1-\nu} + |\cos\psi|\right)^2} \tag{12}$$

$$\alpha = \tan^{-1}\left[\left(\frac{\nu}{1-\nu} + |\cos\psi|\right)\frac{wL}{U_o}\right] \tag{13}$$

$$t/L = 2\left(\frac{\nu}{1-\nu} + |\cos\psi|\right) \sin\frac{\pi}{N} \tag{14}$$

$(C/L)\,/\,(t/L) \;=\; C/t$

$\beta \;=\; \beta_o + \Phi\,|\cos\psi| \qquad \frac{\pi}{2} \leq \psi \leq \pi$

Also $\quad dy/dx \;=\; y'$ [due to camber] + tan $(\beta-\alpha)$[due to stagger]

For parallel blades, $U_\infty/U_o = 1$, $\beta = \beta_o$, t/L = given constant, and C/t = given constant.

For the case of a parabolic camber line symmetric about its mid-section,

$$\sum_{n=1}^{\infty} \frac{a_n[1-(-1)^n]}{2} \tanh^n \frac{\pi a^{i\beta_c}}{4t} \;=\; 0 \qquad (15)$$

For simplicity and purposes of comparison and illustration, this is the case being studied here in more detail. The coefficients $a_1, a_2, \ldots a_n$ can be calculated with the computer program also.

COMPARISON AND DISCUSSION OF RESULTS

Figure 2 compares the present analysis with some experimental results as well as with the theoretical method presented in Reference 9. The input information for this comparison is

$\nu = 0.33$, $C_r/L = 0.375$ camber = 0, taper = 0, $\Phi = 0$, $N = 8$, $\beta = 30°$,

$\delta/L = 0.04$, $R_o = 1/2$ ft, $\rho = 0.0027$ slug/ft , $a = 1100$ ft/sec, and

$M_o = 0.325$ (estimate).

For convenience in studying the effect of various parameters on noise generation and/or reduction, the fan with the above input information was used as the standard compressor. For the standard case, the value of wL/U_o was taken to be 0.774, and the standard sound power level was 109.9 db re 10^{-12} watts. Figures 3 to 6 show the effect of changing the taper, twist angle, number of blades, and hub/tip ratio. The camber as well as the tip clearance were also calculated; however, for small values of these parameters, there was no significant influence on the sound power level. It can be seen that both the hub/tip ratio and the twist angle are very important parameters in order to achieve minimum noise. Particularly, it appears that for each fan rotor, there is a value for the twist angle for which the noise generated is at a minimum.

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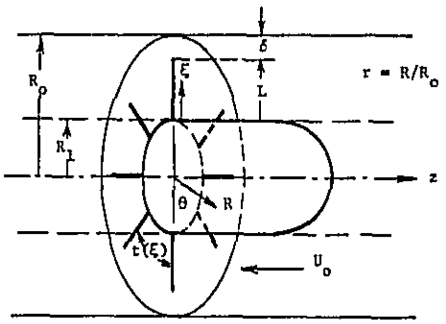

Fig. 1 Coordinate system and definition of symbols

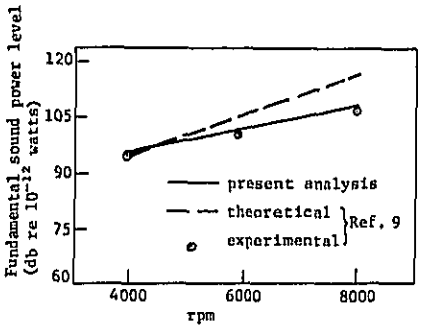

Fig. 2 The effect of rpm on rotor noise

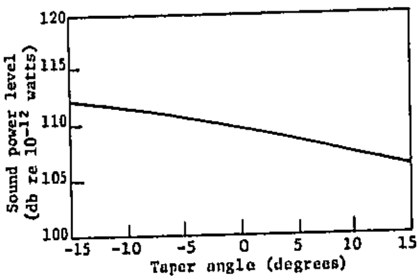

Fig. 3 The effect of taper angle on rotor noise

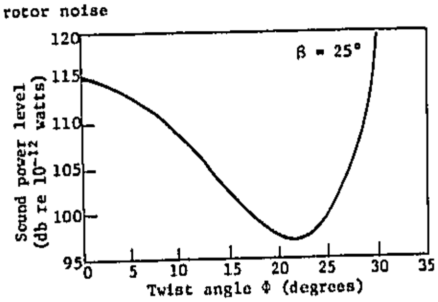

Fig. 4 The effect of twist angle on rotor noise

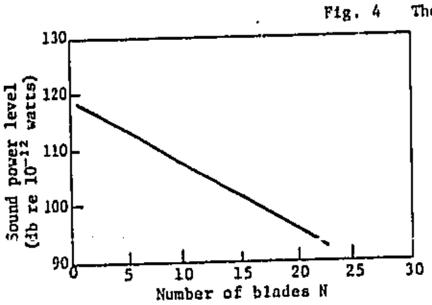

Fig. 5 Effect of number of blades on rotor noise

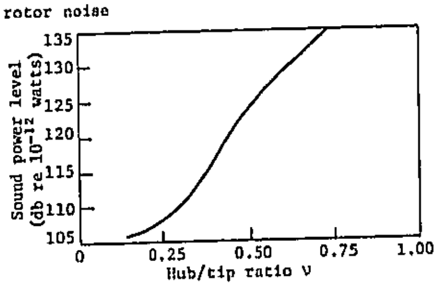

Fig. 6 Effect of hub/tip ratio on rotor noise

RECENT STUDIES OF FAN NOISE GENERATION AND REDUCTION

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INTRODUCTION

A two year comprehensive program on fan and compressor noise was undertaken by Pratt & Whitney Aircraft under contract to the Federal Aviation Administration. The primary objective of the contract was the development of an accurate fan noise prediction system which would enable the selection of proper acoustical design choices for future engines. Other objectives were the development of a deeper understanding of noise generating mechanisms and the demonstration of noise reduction on a modern high tip speed fan vehicle.

To accomplish these objectives a variety of analytical tools, test rigs and full scale engine data were incorporated into a coordinated program. This paper presents a brief summary of the work that was performed and selected results of a few of the experimental programs conducted under this contract.

DEEPER UNDERSTANDING OF NOISE GENERATION MECHANISMS

Future reductions in fan noise must be sought by exploring the possibilities of absorbing sound with inlet and discharge liners and by reducing it at the source through fan design changes. Since the high-bypass-ratio powerplants now entering service already incorporate noise-reducing design features evolved through roughly a decade of concentrated acoustic research, further source reductions will be increasingly difficult. Only through development of significantly deeper understanding of the noise-generating processes involved, is there a reasonable chance of learning how to evolve quieter designs. Similarly, the accuracy and range of applicability of fan noise prediction systems can be established with confidence only when the basic physical processes are clarified.

Theoretical and experimental programs were conducted under this contract to advance the understanding of basic fan noise mechanisms. Work was accomplished in each of the three types of fan noise; discrete-tone, broadband, and combination tone noise.

Interaction noise theory of discrete-tone noise generation has been developed and refined over the past ten years. Portions of this theory were used in the design phase of current high-bypass-ratio turbofan engines. Although lower discrete-tone levels were attained in these engines, this type of noise was not completely eliminated from the radiated noise spectra. An experimental program utilizing a 28-inch diameter single-stage fan rig was undertaken to correlate theoretical vs. measured discrete noise. Wakes were generated by 34 rods, equally spaced circumferentially around the annulus, one inch in front of the rotor blades. Rods were used to minimize distortion of the sound field, while still providing wakes that created an unsteady lift on the blades. This represented a compromise between theoretical simplification and real engine conditions. Figure 1 shows experimental verification of the cutoff phenomenon predicted theoretically. The blade passing frequency tone rises rapidly above the residual noise level at the cutoff speed. This is the rotor speed at which the interaction pressure pattern sweeps the outer duct wall at supersonic speed, and the noise propagates unattenuated in the duct. In a reference configuration with rods removed, inlet duct flow distortion was identified as a source producing interaction modes similar to those produced by wake interaction with the rotor.

There has long been a need for instrumentation to measure the unsteady pressure distributions on airfoil surfaces in the fan, since these fluctuations are the primary source of internal aerodynamic noise. The state of the art has progressed so that suitable miniature transducers are now available and methods for installing them in rigs and engines have been developed. Surface pressure transducers were installed on the suction and pressure surfaces of both rotor blades and downstream stator vanes, and fluctuating pressures were successfully measured. Figure 2 shows the fan rotor instrumented with miniature pressure transducers. High sensitivity to unsteady phenomena is indicated in the spectrum. Exit guide vane passing frequency is clearly seen even though the rotor blades were two chords upstream of the vane. With the development of this capability, the measured forces on blade and vane surfaces may be used to calculate discrete noise levels within the duct and to aid in development of unsteady blade aerodynamic theory and theories of broadband as well as discrete noise.

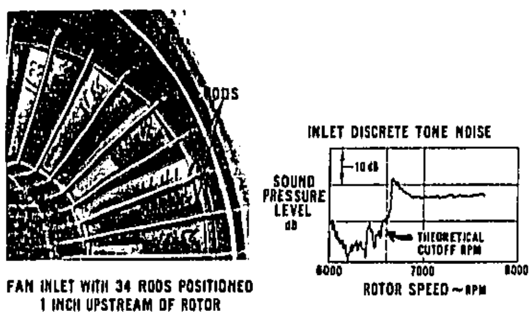


Figure 1 Demonstration of Cutoff Theory - 28 Inch Fan Inlet Rod Test

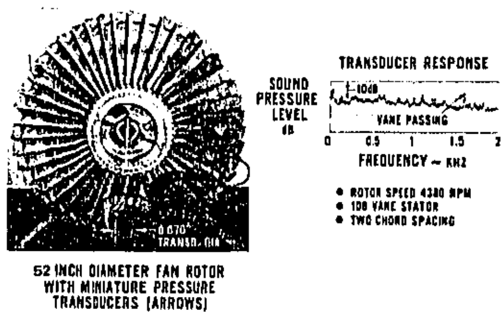


Figure 2 Miniature Surface Pressure Transducers On Fan Rotor Blades

An understanding of the mechanisms of broadband noise generation is essential to the development of more effective means for predicting and controlling this type of fan noise. Several sources of noise have been identified such as turbulent flow over blades and vanes, boundary layers and noise due to scrubbing of air over duct walls. In a fan it is not easy to identify the various sources of noise and to assess the relative contribution of each to the overall spectrum. It is necessary to conduct more simple experiments to study a limited number of noise mechanisms, one at a time if possible, in a controlled environment.

Experimental work was carried out with an isolated airfoil immersed in the flow downstream of a model nozzle as shown in Figure 3. The airfoil was subjected to both smooth and turbulent flow. The resulting sound generated was studied as a function of all variables that were thought to be important, including blade area, air speed, the intensity and spectrum of turbulence in the stream, thickness-to-chord ratio of the airfoil, and angle of attack. (Refs. 1, 2).

Another kind of noise appears when current turbofans operate at high power. Combination tone noise, or multiple pure tone noise as it is sometimes called, is generated by the blade tip shock waves at supersonic fan inlet relative tip Mach numbers, and manifests itself in the noise spectrum as a series of discrete tones at integral multiples of fan rotational speed. This noise is produced because the slight differences in blade tip leading edge shock waves, resulting from small blade-to-blade manufacturing tolerances, are accentuated by nonlinear behavior as the shock waves propagate forward of the fan. The blade-to-blade periodicity of the pressure field very close to the fan is thus lost, and is replaced by an irregular pattern rotating with the fan.

Figure 3 Broadband Noise Research Studies — Isolated Airfoil

A two dimensional shock field was set up experimentally by placing a coaxial sleeve in the inlet of a 28-inch diameter fan rig. This sleeve and the rig outer wall produced a narrow annulus duct upstream of the fan within which radially uniform shocks were generated. Probing of this duct allowed detailed information to be obtained on shock wave propagation. A mathematical model based on statistical analysis was derived that predicted the expected distribution of power among the combination tones for certain types of blade geometry. (Refs. 1,3).

THREE FAN PROGRAM

Noise is generated in fans and compressors by several aerodynamic processes resulting from the flow through blade and vane rows. As a complete mathematical description of the details of the fan flow has eluded the aerodynamicist, it is not difficult to understand that the prediction of noise radiation from these marginally understood aerodynamic processes presents a formidable challenge.

In view of the virtual impossibility of calculating fan noise generation from first principles of aerodynamics and acoustics, empirical methods have been used extensively in the development of current prediction procedures for each type of fan noise. Data for incorporation into these empirical procedures were gathered from a series of tests on various size fan vehicles. Test programs were conducted which provided data over a wider range of fan operating parameters than previously was available, thus supplying critical input for the empirical procedures. Noise amplitude and spectral distribution were related to predictable or measurable fan performance parameters and design features.

Current fan design technology allows a flexibility in the design fan tip speed, while maintaining approximately the same diameter, flow per unit area, pressure ratio and fan efficiency. Accordingly, three fans having design point tip speeds of 1100, 1430, and 1600 ft/sec were designed and fabricated.

The low and high speed fans bracketed the 1430 ft/sec baseline fan, which is representative of fans used in current high bypass ratio turbofan engines. One fan blade from each cascade is shown in Figure 4. All fans incorporated known favorable acoustical design features such as no inlet guide vanes, proper selection of rotor and stator vane numbers and wide rotor-stator separation. Tests of these fans provided data which were used to separate the effects of tip speed and blade loading. Figure 5 shows the operating regions of each fan in terms of pressure ratio and fan tip speed. The testing of these fans also allowed a direct assessment of the effects of fan tip speed on noise generation, and the establishment of whether a low tip speed fan having relatively high blade loading had a noise advantage over a fan having a higher tip speed and lighter loading.

These fans, approximately 52 inches in diameter, were tested on a single stage fan rig, driven by a JT3D turbine, which in turn is powered by the exhaust from a JT3C turbojet engine. Suitable sound muffling techniques have been employed to keep the noise generated by the drive engine and free turbine exhaust from contaminating the fan noise field. A photograph of the rig is shown in Figure 6. The rig is located outdoors in a large cleared area especially prepared for obtaining far field noise data. Eighteen microphones permanently positioned along an arc 150 feet from the fan inlet at intervals no greater than 10 degrees provide far field noise measurements from 0 to 135 degrees from the inlet centerline.

Results of these tests are summarized in Figure 7, in which tone corrected PNdB levels at the angles of maximum inlet and fan discharge noise along a 200 foot sideline are plotted against percent design speed of each fan. It is seen that the low tip speed fan is between 1 and 3 PNdB less noisy than the two higher speed fans over the entire fan operating range. Significantly larger differences between the low speed and the other two fans are noted at the inlet. The maximum difference occurs in the cutback thrust region, and is primarily attributable to the fact that no combination tone noise is generated by the low tip speed fan.

Differences in spectral content of the three fans are evident in Figure 8, which are typical inlet quadrant spectra for the cutback operating condition of each fan. Combination tones are dominating the high speed fan noise level, and are starting to emerge in the baseline fan, the noise level of which is still controlled by the blade-passing frequency noise. At the subsonic relative tip speed of the low speed fan combination tones are not generated, and both the blade-passing tone and broadband noise levels are lower than those of the baseline fan, again due to its lower tip speed. Thus, we see that three fans, developing approximately the same total thrust, exhibit very different noise signatures.

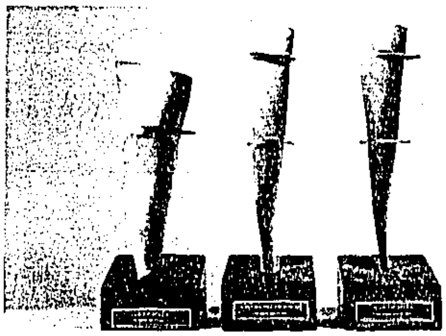


Figure 4 Three Fan Blades

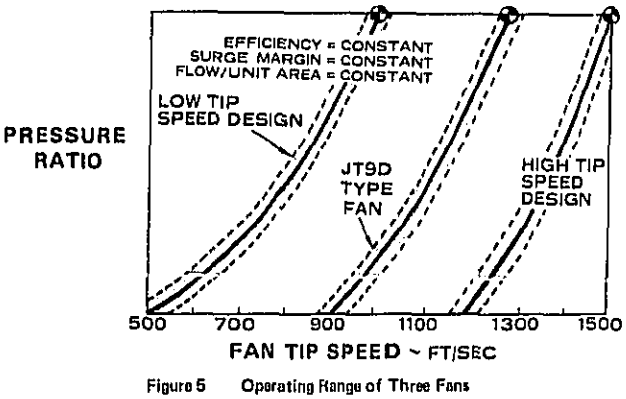


Figure 5 Operating Range of Three Fans

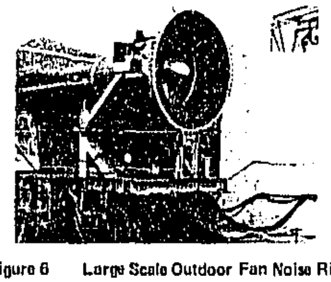


Figure 6 Large Scale Outdoor Fan Noise Rig

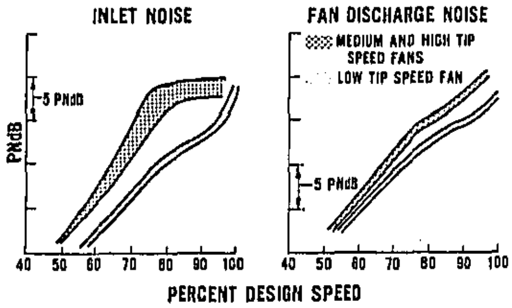


Figure 7 FAA Sponsored Fan Noise Tests - Max Tone Corrected PNdB vs. Percent Design Speed

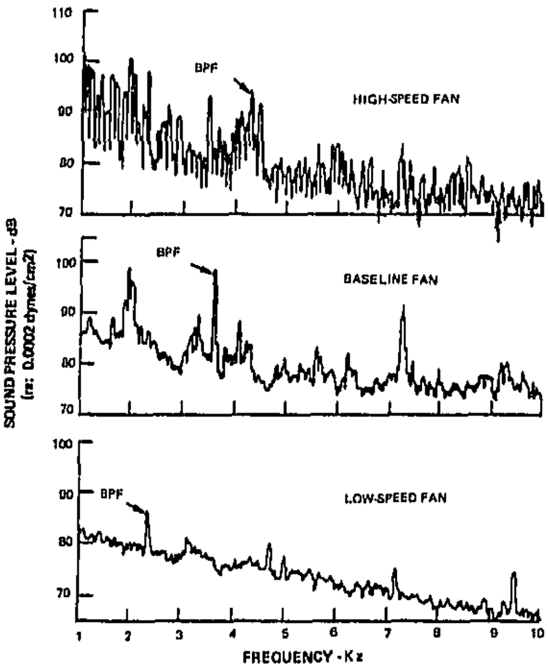


Figure 8 Inlet Spectra Comparison - 78 Percent Design Speed

Fan broadband noise was known to be heavily dependent on both fan tip speed and blade loading. The relative importance of these two factors, however, was impossible to separate from tests of an individual fan because flow and pressure rise, two parameters which are related to loading, also are interrelated with tip speed. Results of tests of the three fans provided uniquely useful data to investigate fan loading effects on noise because substantially different levels of loading were achieved at a given fan tip speed. These data were used to establish empirically the effects of loading on fan noise.

The variation of broadband noise with fan tip relative Mach number from the three fans tested is illustrated in Figure 9 for the 60° inlet angle and the 2500 Hz center frequency 1/3 octave band. Each fan was tested with both an undersize and oversize area nozzle as well as with a baseline nozzle. Typical measured levels for each nozzle are shown in the figure. Also shown in the figure are shaded bands which contain the bulk of the data points. In general, broadband noise levels measured with the undersized nozzles are near the tops of the bands, and levels measured with the oversized nozzles are near the bottom of the bands. Similar plots were generated for different 1/3 octave bands in both inlet and fan discharge quadrants. General trends were the same as those shown in Figure 9.

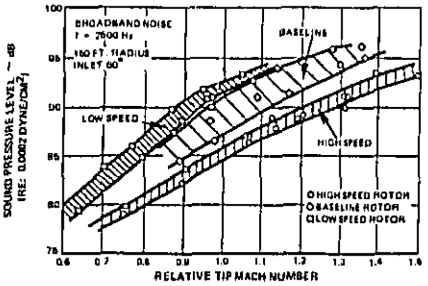


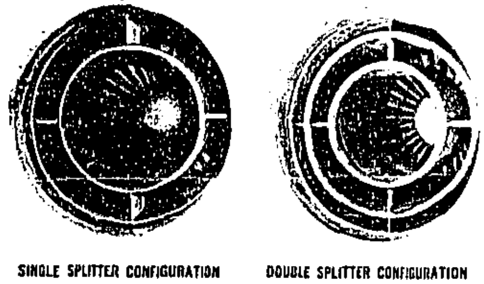
Figure 9 Broadband Noise From the Three Fans

From an examination of this figure, it is clear that factors affecting broadband noise generation other than tip speed are involved. If tip Mach numbers were the controlling factor, all bands would overlap. However, the data are separated into distinct bands, with the highest levels at a given tip Mach number being measured from the low tip speed fan and the lowest levels being measured from the high tip speed fan. At a given Mach number, the low-tip-speed fan was substantially more highly loaded than was the high tip speed fan, it being closer to its design tip speed, resulting in the production of more thrust. Detailed analysis of these data enabled quantitative effects on broadband noise of both fan speed and blade loading to be determined.

NOISE REDUCTION PROGRAMS

An objective of this contract was the demonstration of noise reduction on a fan capable of supersonic tip speed. Those experimental programs conducted under the contract which fit into this category are discussed here. Both the 28-inch diameter rig and the 52-inch diameter fan test vehicle were used for these programs.

The use of treated annular rings in a fan inlet allows more sound absorbing material to be incorporated with no increase in inlet length. The rings provide for more effective use of the treatment because of the existence of opposing surfaces. Single ring and double ring configurations, shown in Figure 10, were compared with a clean inlet configuration. As seen in Figure 11 large discrete noise attenuations are observed for both single and double treated rings. Only moderate reductions were obtained for broadband noise due to noise generated by the rings.



SINGLE SPLITTER CONFIGURATION DOUBLE SPLITTER CONFIGURATION

Figure 10 Treated Inlet Splitter Ring Configurations - 28 Inch Fan Rig, No Inlet Guide Vanes

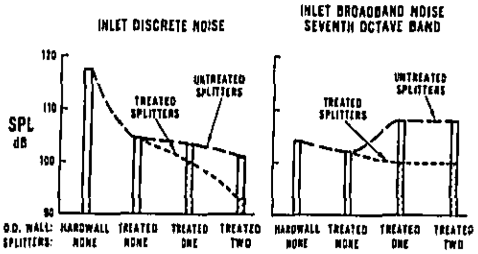


Figure 11 Effect of Treated Inlet Splitter Rings On Discrete and Broadband Noise, 28 Inch Fan Rig - 9000 RPM

The effect of a sound absorbing liner on combination tone noise in the large scale fan rig inlet with no rings is illustrated in the spectra shown in Figure 12. The upper spectrum shows a far field noise signature from a fan operating at a supersonic tip speed with no inlet liner. It shows the characteristic plurality of tones associated with combination tone noise.

The lower spectrum shows the far field noise signature from the same fan at the same tip speed, but with 46 inches of sound absorbing liner in the inlet. It can be seen that the tones are reduced by as much as 20 dB and the final spectrum shape is quite different.

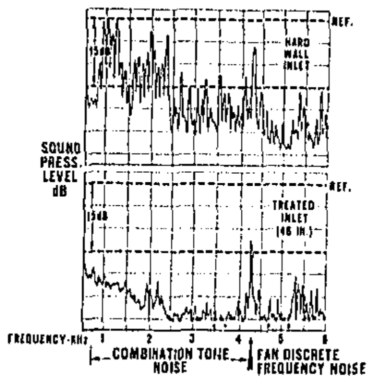


Figure 12 Effects of Sound Absorbing Inlet Liner on Combination Tone Noise, 52 Inch Dia. Fan Research Rig

The final noise reduction program to be discussed evaluated the effects on noise generation of fan exit guide vane designs at selected locations behind the fan rotor. Results shown in Figure 13 illustrate the effect of a 76 vane stator design compared to a 108 vane stator at similar locations behind a 46 blade fan.

Test results for the 76 stator design (which does not consider noise theory) show that improvements can result from increased spacing. The configuration in which the number of vanes was selected on the basis of acoustic theory (108 stators) shows a large reduction in noise relative to the 76 vane stator. This is a result of the blade/vane interaction noise decaying inside the duct rather than propagating. As a result there is essentially no change in noise for the 108 vane configuration as spacing is changed within the limits shown.

The noise reduction concepts tested proved to be successful for one or more types of fan generated noise. Some of the noise reduction features have been employed in current high bypass ratio turbofan engines. Others may be used in conjunction with each other, and, with further development, may be incorporated in designs of future turbofan engines.

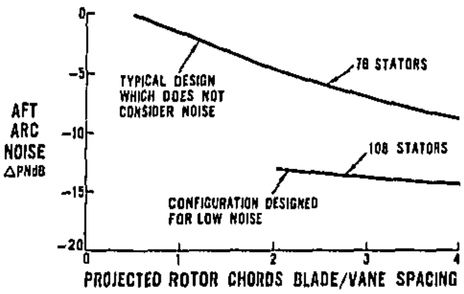


Figure 13 Effect of Exit Guide Vane Design on Fan Noise

APPLICATION TO FAN NOISE PREDICTION

The three separate types of fan noises (discrete tones, broadband, and combination tone) appeared to be mutually independent. Different correlations of noise and performance parameters were explored using mathematical models and noise data. A tabulation of the design and performance factors most correlatable with the measured noise data is presented in Figure 14 for each type of fan noise. It was found that the primary variables were relative Mach number, the Mach number of the air that the airfoil sees approaching it, the blade loading and the duct and blade geometry.

The three individual correlations were finally combined into an integrated overall fan noise prediction computer program for single stage guide vaneless fans having design tip speeds between 1100 and 1600 ft/sec, fan pressure ratios between 1.35 and 1.6, fan gap/chord ratios from 0.5 to 1.1 and fan diameters from 30 to 100 inches. The information provided under this contract is applicable to a wide range of current and future fan designs.

	DISCRETE TONE	BROADBAND	COMBINATION TONE
AMPLITUDE	ROTOR FIELD CUTOFF RATIO (~ REL. TIP MACH NO.)	REL. TIP MACH. NO.	REL. TIP MACH NO.
	FAN PRESSURE RATIO	BLADE LOADING (AERO. DIFFUSION FACTOR)	NO. OF PROPAGATING SHOCKS
	FAN DIAMETER	FAN DIAMETER	
		FAN BYPASS RATIO	
SPECTRUM	NO. OF BLADES	REDUCED FREQUENCY $\left(\frac{f' - f_c}{U}\right)$	STATISTICAL MODEL
	FAN RPM		BLADE NONUNIFORMITIES
		FAN DIAMETER	ENTRANCE REGION CURVATURE

Figure 14 Design and Performance Parameters

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EXTRANEOUS MODES IN SOUND ABSORBENT DUCTS

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INTRODUCTION

In this paper, a brief account of some of the recent developments in the theory of transmission and attenuation of sound in lined ducts is presented. These developments are felt to be significant in the analysis of optimization of sound attenuation. This analysis could also be relevant for the analysis of instabilities in the sound transmission in such ducts.

Attention has been focused in this paper on the modes which are termed here as the extraneous modes. In most cases, these modes would be generated with a much higher modal density than the conventional modes. In the case of the non-uniform wall admittance along the length of the duct, the boundary condition at the wall is satisfied at each point by a number of such modes acting together. Even when the impedance is uniform, the presence of fluctuating vorticity, a single such mode could be present. For, although the mode does not produce enough acoustic velocity in the direction normal to the walls, the balance is provided by the vorticity component of the fluctuating velocity. There is some evidence to suggest that a combination of both non-uniformity and vorticity could well result into instabilities.

In what follows, a brief account of the existing theory of transmission and attenuation of sound in absorbent ducts, relevant to the later sections, is presented. This is followed by an illustrative example to demonstrate how the extraneous modes get generated in the case of the non-uniform wall impedance. A discussion of the implications of such a result is included.

DEVELOPMENT OF THE THEORY OF TRANSMISSION AND ATTENUATION OF SOUND IN DUCTS

The theory of attenuation of sound in ducts has received contributions from many researchers from Kirchkoff and Rayleigh, onward. This theory becomes a natural follow up of the theory of transmission of sound in rigid walled ducts. About a decade ago, Tyler and Sofrin (1) and Morfey (2) had analyzed the problem of transmission of sound in rigid ducts, and its radiation from the end of the duct. These studies involved analyzing the sound field into component eigenfunctions or modes. The transmission of each mode was studied separately and the field at any point could simply be evaluated by adding their contributions together. Even the energies could be calculated separately, for these modes were orthogonal and were not coupled to one another. The analysis of sound transmission in rigid duct has recently been considerably modified by Doak (3) and also by Lansing (4). Lansing has shown that an accurate analysis of a finite length duct would require use of more complex sets of modes, ((m,n,p) modes instead of (m,n) modes). There is, indeed, an epistemological similarity in between the analysis of that paper and the analysis that is to follow.

It was shown (Refs. 5,6) that when the duct is lined with a moderately absorbent material, the sound transmission inside it could be analyzed as a perturbation of the sound field in a rigid walled duct. The eigenvalues were now complex, yet the coupling in between the modes was negligible. Rice (7) and Vaidya (8) extended this analysis to the entire range of impedance. Although this involved modes which were non-orthogonal, the eigenvalues of these modes had a one-to-one correspondence with those of the rigid walled ducts. When the viscosity effects at the walls or the shear boundary layer effects are accounted for in terms of fluctuating vorticity effects, altering the 'effective' boundary conditions at the walls, a similar analysis results. Here too, there remains a one-to-one correspondence with the hard wall eigenvalues.

This is no longer so, once axially non-uniform boundary conditions are considered. Such non-uniformities are inevitable in many cases, either through nonlinear behavior of lining materials or axial growth of the boundary layer. What is of a stronger interest is the evidence that such non-uniformities might indeed be desirable in some cases. It could be predicted that the liner of the future would be designed to have large non-uniformities along its length to achieve maximum attenuation with a given length and weight of the lining.

In what follows, a preliminary analysis of sound transmission with non-uniform boundary conditions is presented. At first, sound transmission in a hollow circular duct is discussed. This is followed by indications of modifications which are to be expected in presence of vorticity and in the case of non-uniform wall impedance.

ATTENUATION OF SOUND IN A CYLINDRICAL DUCT

Consider an infinitely long cylindrical duct of radius 'a' with its axis coincident to the axis of x, having a locally reacting wall, of normal acoustic impedance z_t.

To analyze the sound fields in such a duct, under idealized conditions, such as absence of viscosity and heat conduction effects and no mean flow, solutions of the wave equation

$$\nabla^2 p + k^2 p = 0 \tag{1}$$

with appropriate boundary conditions are required. In this equation, 'p' is the pressure phasor, the total acoustic pressure being $pe^{-i\omega t}$, and k is the wave number, $k = \omega/c$ where c is the speed of sound of the fluid in the duct.

The solutions have been obtained in the following form (for example, Ref. 5):

$$p = p_o e^{i\alpha kx} J_m(\mu r) e^{im\theta}, \qquad \mu^2 = k^2(1-\alpha^2) . \tag{2}$$

Values of μ, the transverse component of the wave number, depend upon the boundary condition. It is shown by equating the radial component of velocity with the radial velocity at the walls that values of μ could be obtained by the solution of the transcendental equation

$$\frac{zJ_m'(z)}{J_m(z)} = \eta , \tag{3}$$

where

$$\frac{i\rho\omega a}{z_t} \equiv \eta , \tag{4}$$

and

$$\mu a \equiv z . \tag{5}$$

$\eta = o$, corresponds to the perfectly rigid walled duct. In this case, (and in the case of pressure release duct), except for the well known cut-off phenomenon, there is no attenuation. (μa) assumes a set of values, $\beta_o, \beta_1, \beta_2 \ldots$. $\eta = \infty$, corresponds to the perfectly pressure release walled duct and the solutions for (μa) could be expressed as $\Lambda_o, \Lambda_1, \Lambda_2 \ldots$. Both β's and Λ's are wholly real and moreover, $\beta_n < \Lambda_n < \beta_{n+1} < \Lambda_{n+1} \ldots$, this means that these two sets are interlaced with one another.

η in general would be different from either o or ∞ and could well be complex. In any case, a number of methods could be used to solve the equation (3). What is of an essential importance to this paper is the result, that once such a solution is obtained, say $z = \mu a = z_1 + iz_2$, there exists one, and only one, solution such that $\beta_n < z_1 < \Lambda_n$. Thus, although the eigenvalues of a uniformly absorbent duct are altered, it is possible to establish a one-to-one correspondence of the absorbent duct eigenvalues to those of the rigid duct eigenvalues. Thus, the complex domain of eigenvalues is subdivided into regions within which the solutions of the equation are single valued. Further discussion of this point would be included in Reference (9).

It was pointed out by Cremer (10) that viscosity (and heat conduction) near the walls creates a certain amount of fluctuating vorticity. This vorticity alters the effective boundary condition for the acoustic field. This would therefore alter the eigenvalues of each modes. The case of flow with shear is much more complex. Mohring (11) has suggested that sound transmission under these conditions would have to be analyzed in terms of continuous modes rather than discrete modes.

EXTRANEOUS MODES AND THEIR AMPLITUDES

Consider now, a hollow cylindrical duct, similar to the one in the previous section except that it has a wall impedance which varies with x, such that

$$z_t = z_{t,o} e^{-iqkx} \tag{6}$$

where 'q' could well be complex quantity. Equation (3), then, changes to

$$\frac{zJ_m'(z)}{J_m(z)} = \eta_o e^{iqkx} \tag{7}$$

or $\frac{zJ'(z)}{J(z)} = \eta e^{iqkx}$, if the subscripts 'm' and 'o' were to be suppressed.

Solutions of the wave equation could be obtained in this form

$$p = e^{im\theta} [J(k^*_{\bar{o}} r) e^{i\alpha_{\bar{o}} kx} + b_1 J(k^*_{\bar{1}} r) e^{i\alpha_{\bar{1}} kx} + b_2 J(k^*_{\bar{2}} r) e^{i\alpha_{\bar{2}} kx} \tag{8}$$

where

$$J'(k^*_{\bar{o}} a) = 0 , \tag{9}$$

$$\alpha_{\bar{o}} = \sqrt{k^2 - k^{*\,2}_{\bar{o}}} , \tag{10}$$

$$\alpha_{\bar{n}} = \alpha_o + \bar{n}\, q , \tag{11}$$

$$k^*_{\bar{n}} = k\sqrt{1 - \alpha_{\bar{n}}^2} \tag{12}$$

b's are the coefficients through which the extraneous modes are linked to the rigid wall modes;

$$b_1 = \frac{\eta_o\, J(k^*_{\bar{o}} a)}{(k^*_{\bar{1}} a) J'(k^*_{\bar{1}} a)} \tag{13}$$

and

$$b_2 = \frac{\eta_o\, J(k^*_{\bar{1}} a)}{(k^*_{\bar{2}} a) J'(k^*_{\bar{2}} a)} b_1, \text{ etc.} \tag{14}$$

The expansion depends upon the condition that $b_{\bar{n}} e^{i\alpha_{\bar{n}} kz} \to o$, as $\bar{n} \to \infty$. (15)

It should be noted that the subscript '$\bar{n}$' here, is different from 'n' of the conventional hard wall analysis. Thus, for a given 'm' there is a set of solutions of $J'(k^*_o a)$, giving rise to $k^*_{\bar{o}}$'s which are the same as the k^*_{mn}'s in the hard wall analysis notation. Each of these gives rise to a further set of $k^*_{\bar{n}}$'s, etc.

There is an alternative expansion possible, where the form of equation (8) is retained; but

$$J(k^*_{\bar{o}} a) = 0 , \tag{16}$$

$$\alpha_o = \sqrt{k^2 - k^{*2}_{\bar{o}}} \tag{17}$$

$$\alpha_{\bar{n}} = \alpha_0 - \bar{n}q \tag{18}$$

and

$$k^*_{\bar{n}} = k\sqrt{1 - \alpha_{\bar{n}}^2} \tag{19}$$

Further,

$$b_1 = \frac{(k^*_{\bar{0}}a)J'(k^*_{\bar{0}}a)}{\eta_0 J(k^*_{\bar{1}}a)}\,, \tag{20}$$

$$b_2 = \frac{(k^*_{\bar{1}}a)J'(k^*_{\bar{1}}a)}{\eta_0 J(k^*_{\bar{2}}a)}\, b_1, \text{ etc.} \tag{21}$$

Once again the same condition is valid for convergence. For any given situation, only one of these expansions would meet the convergence condition.

Considerable modifications are to be expected in this analysis when effects such as those due to the finite duct length and vorticity are considered. When the dependence of wall impedance on length is non-exponential, an even more complex picture would emerge. In all such cases the modal generation function, suggested by Vaidya (12) might turn out to be a more useful concept than the concept of conventional modes.

It should be noted that some of the extraneous modes would have a tendency to grow in amplitude along the length of the duct. Normally, this would invalidate the radiation condition and such modes would have to be rejected. However, when there exist mechanisms of energy transfer, (say from the flow energy or through the active absorbents) to acoustic energy, this would not be so.

Even such a preliminary analysis would seem to suggest the need for further experimental and theoretical investigation of the non-uniform wall impedance problem.

CONCLUSIONS

It has been argued that an analysis of non-uniform wall impedance would be of significant importance for optimization of attenuation in flow ducts. A technique is presented for such an analysis when the impedance has a specific dependence on the length. A number of additional modes, termed as extraneous modes, get generated under such conditions. It is also expected that such modes might explain the instabilities which are observed in flow ducts lined with porous materials.

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DEVELOPMENT OF A SONIC INLET FOR JET AIRCRAFT

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INTRODUCTION

The increasing demand for quieter propulsion units in future aircrafts (particularly STOL and VTOL) requires a reevaluation of our present method of attempting to reduce inlet noise by means of acoustical treatment. Short of a new breakthrough in the design of a quiet propulsion unit, it appears that the sonic inlet can be an attractive alternative to other available methods of inlet noise attenuation. The principle of the choked inlet involving the acceleration of the inlet air to sonic velocities to reduce inlet noise is well known. But because of the numerous aerodynamic and design problems, no design for subsonic aircrafts exists so far which could be termed flyable from a practical standpoint. Although the choked inlet may be impractical for purposes of retrofit, it may be possible to incorporate it into the initial design as part of a new power plant. Particularly for the STOL aircraft where takeoff and landing requires near-maximum thrust, the choked inlet is especially competitive compared with other methods of inlet noise reduction, since here the amount of area variation to maintain choked operation need not be large.

One of the early papers on the subject of the choked inlet was by Sobel and Welliver [1] who tested the choked inlet mainly for its acoustic response with a bellmouth and a long inlet. Except for overall pressure recovery, very little aerodynamic data was published in their paper. Since then, many proposals and publications have appeared on the sonic inlet [2-10]. However, some of these papers deal only with measurements to determine the acoustical effectiveness of the choked inlet.

The problems of the choked inlet as well as some of the possible methods to maintain choked flow are shown in Figure 1. This figure presents the requirements, methods, problems, and the basic approach to a solution for the choked inlet, but the divisions into these categories are of course superficial since they are all interrelated; however, it is convenient to divide and identify each problem separately. Presently the available experimental results on the aerodynamic performance of the choked inlet are largely random and uncorrelated, perhaps with the exception of the recently published documents by Boeing [11]. However, it seems unlikely that a successful choked inlet can be developed with special experimental tests conducted for specific engines under restricted conditions. Basic to the lack of progress on the sonic inlet is the lack of a systematic theoretical analysis.

Theoretical Considerations

Theoretically, choking occurs when the inlet air is accelerated to sonic velocity, but what constitutes choking in an actual inlet is a vital and perhaps controversial point. With a fluid at rest, it can be assumed that the waves emitted by the noise source in the duct travel out of the duct with little or no back reflection; however, as the fluid accelerates, some of the waves are convected or reflected from the mouth of the inlet until sonic velocity is reached. At this point, the flow velocity equals the wave velocity and noise can no longer propagate forward. In an actual inlet the three-dimensional flow field interacts with noise propagation in a rather complex manner. Thus an accurate theoretical model of interaction of flow with the noise is difficult to obtain.

In 1960, a theoretical analysis by Slutsky [12] showed that the reflection ratio in a duct with flow is insignificant at $M < 0.5$ and goes to infinity at $M = 1$. In 1967, Smith and House [13] arrived at the empirical expression

$$- \text{db} = \log_{10}\left(\frac{1}{1-M_n}\right) \tag{1}$$

from their test data to relate the reduction of broadband noise due to duct flow Mach number. Like the reflection ratio, this equation predicts very little noise attenuation until a Mach number of 0.5 is reached, and substantial noise reduction is not achieved until the flow in the duct is near Mach one. In Slutsky's solution it was shown that the reflection ratio is frequency-dependent. Also in some recent experiments [14] it was found that equation (1) should be modified to

$$- \text{db} = \log_{10}\left(\frac{1}{1-M_n}\right)^{x_f} \tag{2}$$

where x_f accounts for the influence of frequency.

Although numerous tests have been conducted to determine the effect of Mach number on sound propagation in inlets, the basis on which these results are plotted show such a wide scatter that it is difficult to determine the exact trend. For example, engine test data from NASA [4] and Rolls-Royce [13] show rather modest noise reduction even at choke, while the data of Pratt and Whitney [13] and Boeing [5,11] show substantial reduction at throat centerline Mach numbers of 0.7 or 0.8. These optimistic results are due to the use of the centerline Mach number parameter (which is the minimum Mach number at a given cross section near the throat). Also the Boeing tests used boundary layer control. It will be seen later that the extent of noise attenuation and the shape of the noise attenuation curve depend on the pressure recovery. With boundary layer control the pressure recovery is of course much higher.

The present program was initiated in 1969 to study theoretically the aerodynamics of the choked inlet (steady state and transient effects) and to provide guidance to an experimental program. The details of the theoretical work are presented in References 15 and 16; this work is still in progress. A summary of the theoretical work and recent experimental results are given in this paper. Tests presently being conducted on an automatic control system and the analog simulation of this system will be the subject of a later report.

AERODYNAMIC STUDY

The theoretical work centered around finding means of achieving steady flow with high pressure recovery. The study consisted of two parts, steady state and transient analysis. The purpose of the steady state analysis was to calculate the amount of boundary layer control needed to prevent separation, and the transient analysis considered the problem of stability as well as the inlet response to upstream and downstream disturbances during choked operation. A transient study is necessary for the selection of a control system. This study, presented in Reference 16, considered only the acoustic instabilities and upstream and downstream disturbances during choked operation but did not take instabilities due to shock-boundary layer interaction into account. The steady state analysis showed that for short inlets, injection and/or vortex generators should be used to minimize separation during choked operation. For example, in order to keep the length/diameter ratio of the inlet approximately the same as in the present-day inlets, the injection mass required to control separation is about 5 percent of the primary air at 90 percent pressure recovery. The study indicates that with a moderate amount of injection, short inlets can be made to choke.

The transient analysis shows conflicting requirements: near Mach one, disturbances such as a gust or thrust variation require a large damping time. On the other hand, boundary layer separation and instabilities due to shock-boundary layer interaction (buffeting) increase with increasing shock strength. Figure 2 which combines the analytical solution of Reference 16 with Osborne's empirical criterion for two-dimensional airfoils shows that for weak shocks there is very little damping of disturbances; for stronger shocks, the damping increases but so does the danger of shock-boundary layer interaction instabilities. Thus if choking requires operating the inlet at Mach numbers greater than one, a reasonable operating point would be at $M = 1.15$, depending on the control system to be used. This problem can be minimized if choking can be achieved at Mach numbers of around 0.8. Unfortunately most inlets have to be designed with radii of curvature at the throat which will lead to rather large variations in Mach number from centerline to wall. For example, in the tests conducted by Boeing [5] where the maximum centerline Mach number reached 0.8, the wall velocity at the same vertical plane was supersonic; thus partial shocks and consequent instabilities can exist.

EXPERIMENTAL STUDY

Since it has been shown both theoretically [15] and experimentally [5] that boundary control by injection is effective in maintaining high pressure recovery and stable flow during choked operation, the present experimental program explored some other methods of boundary layer control as well. Since a number of potential problems can be avoided if choking can be in fact achieved at Mach numbers less than one, this condition was studied further experimentally. It should be mentioned that tests conducted with a series of two-dimensional models showed that no sharp drop in pressure recovery occurs when the throat Mach number approaches one.

Experimental Setup

Figure 3a is a photograph of the experimental setup and Figure 3b is a schematic of the apparatus. The primary air mass flow was between 1.8 and 2.7 lb/sec. Except for the test section, the whole experimental setup including the silencer was acoustically wrapped to reduce background noise. Sound measurements were made with a one-inch condenser-type microphone, a constant-percentage bandwidth analyzer and a graphic recorder. The bandwidth selectivity switch was set at 29 percent which gives a bandwidth of between 1/2 and 1/3 octave. This equipment is shown in Figure 3a.

Four different inlets were tested. Figures 4a, 4b, 4c are drawings of these models, and Figure 4d indicates the area distribution of the models in the choked and unchoked positions. Test models 1, 2 and 3 used the same cowl, whereas test model 4 had its own cowl and a centerplate. Model 1 was first tested without vortex generators, then with vortex generators and finally with ambient air injected over the vortex generators. The objective here was to inject a slight amount of ambient air over the vortex generators to promote mixing and reduce separation and instabilities. Also the design of the injection slot is very important in boundary layer control; however in the presence of vortex generators, the design becomes less crucial. Model 1 with its modifications will be referred to as 1, 1a or 1b, respectively. Model 2 was designed to give a slightly larger pressure gradient than 1. Model 3 was designed with a more rapid rise in pressure gradient downstream of the throat than the other models; here vortex generators did not improve the pressure recovery. Model 4 was designed to increase the pressure recovery by use of an annular ring since in two-dimensional tests, the introduction of a centerplate substantially improved the pressure recovery. All inlets were designed to give an approximate ratio of lenth to maximum inner diameter (L/D) of 1.6 (the typical L/D of the Boeing 707 or DC-8 aircraft is around 0.9). All tests were conducted with flight lips at static conditions. Since all experiments were run with the ejector steam valve wide open to maintain steady flow and also a steady noise level, the centerbody was translated to various positions by pressurizing the cylinder until the desired flow field was obtained. The noise source used was simply the ejector noise and a resonant source at 600 Hz. The upstream microphone was set 6 inches from the inlet and a microphone downstream of the test section was used to check downstream noise.

Results and Discussion

Figure 5 shows the overall SPL reduction as a result of increasing Mach number. This Mach number is the average maximum axial Mach number as determined by pressure readings on the cowl and centerbody and considers the aerodynamic rahter than the geometric throat. Test model 4 is not included here because its geometry and hence the aerodynamic data differed somewhat from the other models. In models 2 and 3 there does not appear to be a sharp drop in noise level near Mach one. These two models are less efficient and of an aerodynamically poorer design than model 1. They seem to give a higher noise reduction at lower Mach numbers, but do not give the large noise reduction near Mach one. Figures 6a, 6b, 6c and 6d are noise spectra of inlet models 1, 1a, 2 and 3.

Figure 7 shows the pressure recovery and attenuation for the various inlets. This graph essentially presents the cost versus cure for these inlets with flight lips. It is interesting to note that the noise reduction ability of the inlets is related to the pressure recovery. Test model 1 increased in overall pressure recovery when the vortex generators were added and also increased slightly in pressure recovery when the slot was opened to allow the ambient air to inject over the vortex generators. This increase is very consistent and particularly noticeable at lower Mach numbers. Worth noting is also that the vortex generators on model 3 did not help because of the initial poor design. Model 4 with the centerplate appeared to be a better inlet at lower Mach numbers but deteriorated as choked conditions were approached. Besides giving high pressure recovery, model 1b also gave steady flow, particularly when sonic velocity was approached. For models 2 and 3, the flow fluctuated when choked conditions were approached, and small upstream disturbances caused some pressure oscillations. When a fan with a velocity of 100 ft/sec was used to blow upstream of the inlet at various angles, model 1b showed very little pressure unsteadiness, and rapidly moving the centerbody did not cause large pressure fluctuations.

SUMMARY

From these results, the following qualitative conclusions can be drawn within the scope and limitations of the experiments:

1. Boundary layer control should be used in inlets of reasonable length to obtain high pressure recovery and low distortion.
2. For well-designed inlets (with a high pressure recovery), substantial noise reduction can be achieved at average maximum axial Mach numbers of less than one.
3. High pressure recovery appears to be a prerequisite to large noise reduction.
4. Vortex generators must be properly designed; they are not effective when placed in a poorly designed diffuser.
5. Transient upstream disturbances of 100 ft/sec at various angles of attack did not make the flow fluctuate for the inlet with vortex generators and a slot gap (model 1b). Pressure fluctuations were observed in the other models during choked flow.

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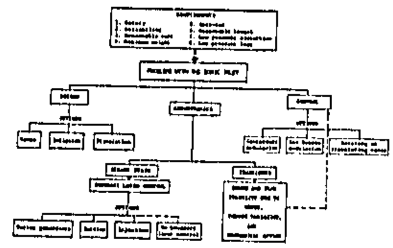


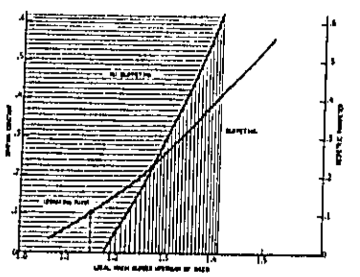


Fig. 3a Experimental setup

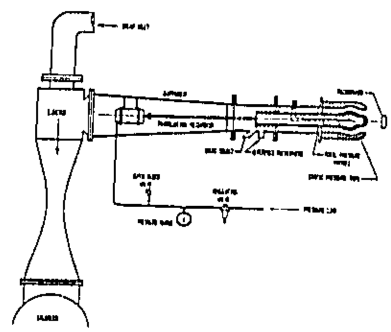
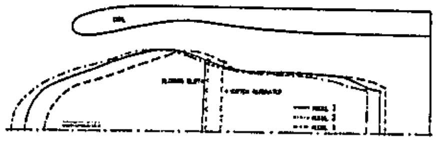
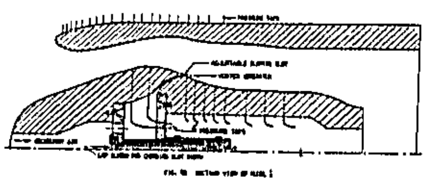
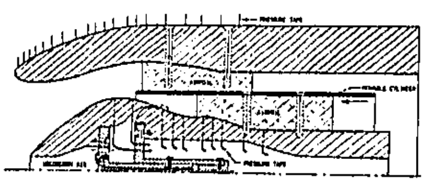


Fig. 3b Schematic of experimental apparatus







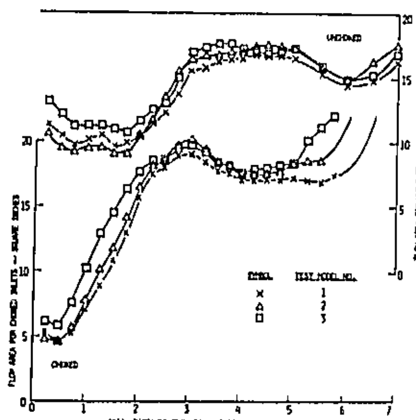


FIG. 4b AREA DISTRIBUTION OF MODELS 1, 2, 3 IN CHOKED AND UNCHOKED POSITION

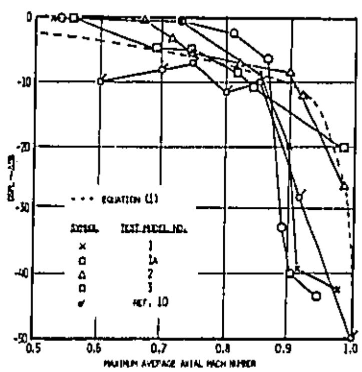


FIG. 5 OVERALL SOUND PRESSURE LEVEL REDUCTION AS A RESULT OF INCREASING AXIAL MACH NUMBER

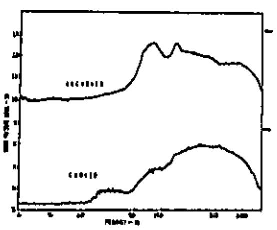


Fig. 6a Model 1 OSPL

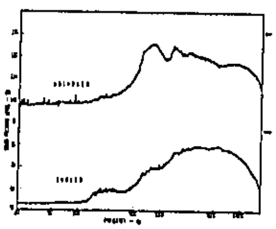


Fig. 6b Model 1a OSPL

Figs. 6a-6d Noise spectra of various models

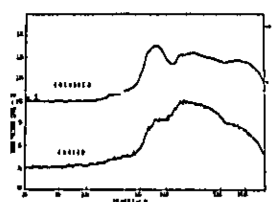


Fig. 6c Model 2 OSPL

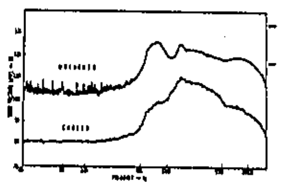


Fig. 6d Model 3 OSPL

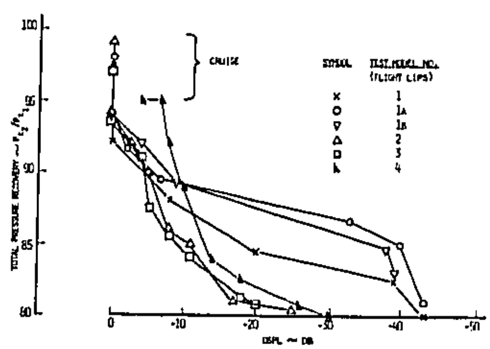


FIG. 7 REDUCTION OF OVERALL SOUND PRESSURE LEVEL VERSUS PRESSURE RECOVERY

COMBUSTION NOISE PREDICTION TECHNIQUES FOR SMALL GAS TURBINE ENGINES

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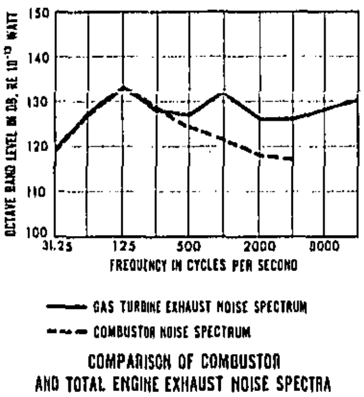
INTRODUCTION

One of the most significant sources of noise from small turboshaft engines and auxiliary power units is the combustion process. The need is often encountered in practice to establish a guideline for the design of a quiet engine. Two approaches are presented in this paper for the use of noise factors in predicting combustion noise. The temperature, velocity, and density gradients that exist in the high-speed flow cause a characteristic low-pitched roar. This is further compounded by interaction with the blades of the turbine section. As a result, turboshaft engine exhaust contains a large amount of acoustical energy even before it emerges. This noise is considerably different from that which is produced by turbojet and low-bypass-ratio turbofan engines, where the predominant source of noise is the turbulent interaction between the moving air jet and the relatively quiescent atmosphere.

Gas turbine engine combustion noise generation is a highly complex phenomenon. Its measurement and analysis are further complicated by the presence of one or more turbine stages through which the acoustic fronts must pass. To clearly identify the contribution of combustion to the noise-generation process, measurements were made of the acoustic spectrum emitted by the combustor alone and with the same combustor installed on a gas turbine engine (Figure 1). The engine used in this analysis featured a straight-through annular combustor. Other engine/combustor combinations have shown similar results.

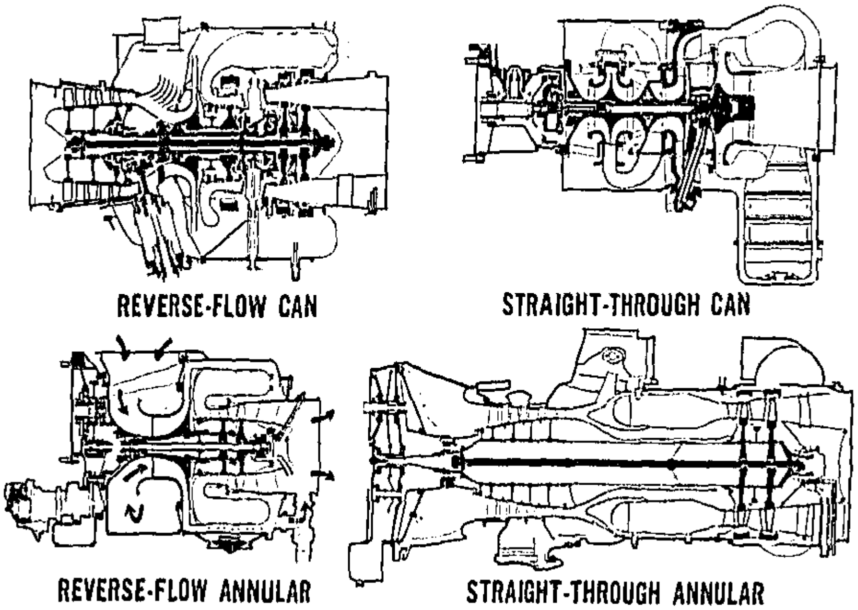


COMPARISON OF COMBUSTOR AND TOTAL ENGINE EXHAUST NOISE SPECTRA (STRAIGHT-THROUGH ANNULAR COMBUSTOR ENGINE, NO LOAD CONDITION)

FIGURE 1

As can be seen from the spectra of Figure 1, combustion noise produces a characteristic low-frequency hump. Because the size and weight of most sound-attenuating devices are inversely proportional to the frequency to which they are tuned, such low frequencies are both the most difficult and the most expensive to remove in an aircraft installation. Thus, an understanding of the engine parameters that control the generation of the combustion noise is important to both the prediction of such noise and its eventual elimination.

Two approaches have been used in this study. First, an empirical evaluation of potential noise factors affecting exhaust noise was conducted. A parameter uniting the combustor inlet temperature (T_3), the combustor discharge velocity (V_d), and the equivalent discharge diameter of the combustor (D_d) was found. Second, a similar expression was derived dimensionally and compared with data from both combustor rig and engine tests. Four types of gas turbine engines were used in evaluating the potential noise factors. They are categorized by the type of combustor used: reverse-flow, straight-through, can, or annular. Outlines of these four types of engines are shown in Figure 2.



CROSS-SECTIONAL VIEWS OF TYPICAL AUXILIARY POWER UNITS DIFFERENTIATED BY COMBUSTION SYSTEM TYPE

FIGURE 2

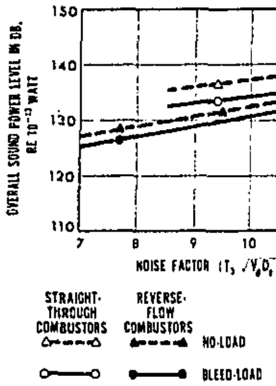
CORRELATION WITH AN EMPIRICAL NOISE FACTOR

A number of engineering parameters are normally used to describe the physical characteristics of the combustor and of the combustion process contained therein. While these parameters themselves are not very descriptive of the detailed flow, heat transfer, or chemical conditions that prevail from point to point within the combustor, they are relatively easy to measure, and they provide the development engineer with information concerning the performance of his system. It is from these readily available parameters that several potential noise factors were constructed and empirically compared to measured sound levels from different engines.

Among the flow conditions considered were the inlet air mass flow, pressure and temperature, and the discharge gas temperature. Two velocities were considered: the reference (inlet) velocity, and the gas discharge velocity. In addition, various characteristic combustor dimensions such as liner length, maximum diameter, and discharge diameter were examined. For annular combustors, the annulus thickness and the equivalent discharge diameter based on the liner discharge area were used. Other measurable or calculable quantities, such as orifice diameter and velocity, were considered briefly.

Obviously, there are many combinations of all these parameters. However, several relatively simple noise factors appeared to correlate with the measured acoustic data.

The combination of $T_3 \sqrt{V_d D_e}$ proved to be the most consistent and the most reliable in predicting gas turbine exhaust noise; where T_3 is the combustor inlet temperature, V_d the combustor discharge velocity, and D_e is the effective diameter of the combustor. In addition, this parameter is in general agreement with the form of a "noise factor" reported in Reference 1. With this parameter used as the independent variable, Figure 3 graphically relates the measured noise levels for several engines. Every datum point represents the average maximum noise level of an engine series. One is from a propulsion engine, while others are from auxiliary power units. For auxiliary power units there are generally two limiting data points indicated; they represent two completely different modes of operation (no load and full bleed load, for example).



EXHAUST OVERALL SOUND POWER LEVEL AS A FUNCTION OF AN EMPIRICAL NOISE FACTOR

FIGURE 3

The acoustical data were taken from eight different engines in accordance with the standard free-field method of Reference 2. Figure 4 shows a typical measurement-test setup. The individual sound pressure level measurements were converted to sound power levels (referenced to 10^{13} watt) by the aforementioned standard.

In most cases, the no-load condition is noisier than the full-bleed load condition. The eight engines appear to give consistent results, since Figure 3 shows good agreement with the arbitrarily drawn curves. Since these curves are representative of the trend of the exhaust-noise output of various engines, similar analyses can be made to predict the noise from other engines.

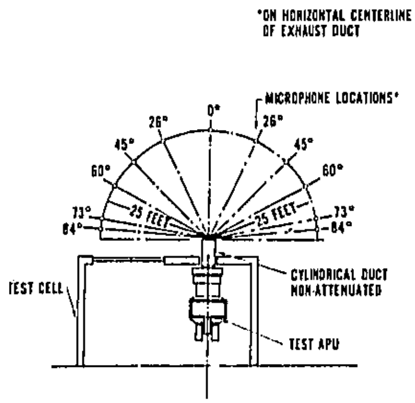
DERIVATION OF A MODIFIED NOISE FACTOR

In order to closely correlate the atmospheric combustor tests with the actual measured engine combustor noise output, a modified factor for the prediction of combustion noise was derived by dimensional analysis. This work is a follow-up of the previously obtained result. The method was based on the energy output through the use of the Buckingham π-theorem. The result came surprisingly close to the empirical noise factor described earlier, but with some additional generalized considerations.

An attempt was made to predict engine noise power output from the following independent variables:

Variable	Symbol
Acoustical power output, lb-ft per sec*	P
Fuel-flow rate, lb per sec	$\dot{\omega}_f$
Airflow rate, lb per sec	$\dot{\omega}_a$
Temperature,** °R	T
Pressure, lb per sq ft	p
Physical dimension ft	D
Combustion energy release rate, lb-ft per sec	W

*For convenience, the generally used watt is converted into the equivalent lb-ft per sec.
**The dimensional analysis does not distinguish from inlet or exhaust temperature; hence, there is no specific designation made at this time. However, later in the derivation a proper choice is allowed.



TYPICAL EXHAUST ACOUSTICAL TEST SETUP
(MICROPHONE POSITIONS)

FIGURE 4

Combustion acoustic efficiency generally varies from one combustor or condition to anther. A functional dependence of the acoustic power output with the combustion energy release; and, in turn, with these basic parameters can be expressed in the following form:

$$P = f\ (W) = f\ (\dot{\omega}_f,\ \dot{\omega}_a,\ T,\ p,\ D) \tag{1}$$

By applying the Buckingham π-theorem, the following expression can be assumed:

$$W = C\ \left(\dot{\omega}_f\right)^{\alpha}\left(\dot{\omega}_a\right)^{\beta}(T)^{\gamma}(p)^{\xi}(D)^{\lambda} \tag{2}$$

where C is a proportional constant and α, β, ε, and λ are arbitrary exponential constants to be determined by the experiments.

By substituting dimensions into the above expressions,

$$\left(\frac{\text{lb-ft}}{\text{sec}}\right) \sim \left(\frac{\text{lb}}{\text{sec}}\right)^{\alpha}\left(\frac{\text{lb}}{\text{sec}}\right)^{\beta}\ (°\text{R})^{\gamma}\left(\frac{\text{lb}}{\text{ft}^2}\right)^{\xi}(\text{ft})^{\lambda} \tag{3}$$

Notice that in the right-hand side of the above expression there is a dimension of (°R) to the γth power which is not directly related to the dimensions on the left-hand side. Hence, $\gamma = 0$ implies that temperature itself, at this moment, is not a basic variable. It may well be obtained later as a derived quantity. In work described earlier in this paper and elsewhere, the importance of such a factor has been revealed. Thus, justification of the unit is needed before formally proceeding with the dimensional analysis. A transformation of dimensions will demonstrate the need for inclusion of a temperature factor in the analysis.

If a derived quantity ($\dot{\omega}_f \Delta H$) were introduced, this item would represent the energy release rate of the combustion, where ΔH would be the heat released by the fuel. Expressing the combustion efficiency as η and temperature increase due to combustion, $T_4 - T_3$, as ΔT, we arrive at the correlation as

$$\left(\dot{\omega}_f \Delta H\right) = \frac{1}{\eta} \left(\dot{\omega}_f C_p \Delta T\right) \left(1 + \frac{1}{f}\right) \tag{4}$$

where C_p is the specific heat at constant pressure and f is the fuel/air ratio. The dimensions for this quantity generally are Btu per sec; it can be converted into lb-ft per sec by a proportional constant. Thus, Equation (2) can be rewritten as

$$W = C \left[\dot{\omega}_f \Delta T \left(1 + \frac{1}{f}\right)\right]^{\alpha} (P)^{\beta} \left(\dot{\omega}_a\right)^{\gamma} (D)^{\xi} \tag{5}$$

where for simplicity, the coefficient C is the resultant constant, representing all the necessary proportional constants--e.g., unit-conversion constants, specific heat, and efficiency.

After collecting the exponentials and solving the equations for the exponentials, the acoustical power output can be expressed as

$$W = C \left(\dot{\omega}_f \Delta T\right)^{\alpha} \left(1 + \frac{1}{f}\right)^{\alpha} \left(\dot{\omega}_a\right)^{1-\alpha} (D)^{1-\alpha} \tag{6}$$

With use of the equation of state for the gas, and the foregoing equation further reduced into an expression compatible with the empirical factor, it then becomes

$$W = C[(T_4-T_3)^{\alpha} \; V_d \; (D_e)^{3-\alpha}] (1 + f)^{\alpha} \; P_4 T_4^{-1} \tag{7}$$

where C and α are quantities to be obtained from experiments. However, for the information available at present, useful reference can be made to the previously obtained noise factor. A choice of $\alpha = 2$ can be easily observed; then Equation (7) becomes

$$W = C[(T_4-T_3)^2 \; V_d \; D_e] \; (1 + f)^2 \; P_4 T_4^{-1} \tag{8}$$

or it can be expressed as

$$W = CF^2 \tag{9}$$

where the modified noise factor becomes

$$F = (T_4 - T_3) \sqrt{V_d D_e} \; (1 + f) \sqrt{\frac{P_4}{T_4}} \tag{10}$$

Further expanding Equation (1) into a series, and substituting Equation (10) into it,

$$P = a_o + a_1 \; F^2 + a_2 \; F^4 + a_3 \; F^6 + \cdots \tag{11}$$

In other words, the acoustic power output P can be expressed in an even series of the noise factor F with appropriate coefficients, a_o, a_1, a_2, a_3, etc, to be determined experimentally.

From the acoustic data collected from engine tests, the third term of Equation (11) is dominant (Figure 5), and it reduces to

$$P \sim F^4 \tag{12}$$

while from the atmospheric rig combustor tests, it was found that (Figure 6)

$$P \sim F^2 \tag{13}$$

By using the definition of sound power level (PWL) re: 10^{-13} watts, Equations (12) and (13) can be expressed as

For engine combustion noise:

$$PWL = 40 \log_{10} F + B_1 \quad (14)$$

For air-rig combustor noise:

$$PWL = 20 \log_{10} F + B_2 \quad (15)$$

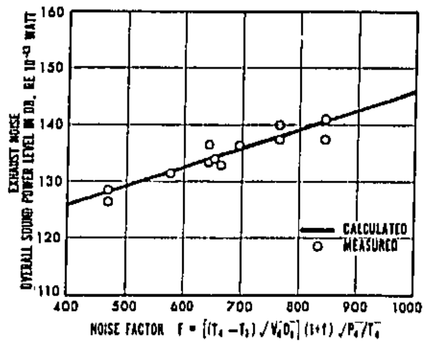
Constants in Equations (14 and (15) were found experimentally as:

$$B_1 = 23 \text{ and } B_2 = 81 \quad (16)$$

DISCUSSION

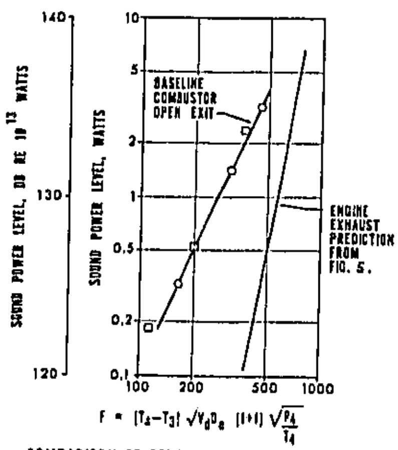
Equations (14) and (15) provide a method for the small gas turbine engine designer to predict the acoustical power level generated by a given design and, further, to predict the effects of changing one or more design parameters. This method has proven useful for several different families of small engines at AiResearch.

Because of limitations on the size of combustors used in the small gas turbine engine class, many of the characteristic dimensions of the combustors were similar. It was also noted that the peak frequencies of the combustion noise spectra also tended to remain fixed. From References 1 and 3, it might be concluded that the frequency as well as the amplitude of the combustor noise hump is related to one or more of these dimensions. It is also possible (References 1 and 4) to relate the sound output level to the flame speed and, thus, to the stoichiometric fuel/air ratio. This suggests a number of parameters, both chemical and physical, that might yet need to be considered. For instance, Powell (Reference 5) suggests that different fuels have differing thermal-to-acoustical conversion efficiencies.



PREDICTED AND CALCULATED EXHAUST NOISE SOUND POWER LEVELS

FIGURE 5



COMPARISON OF COMBUSTOR SOUND PRESSURE LEVELS WITH EXHAUST NOISE DATA.

FIGURE 6

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"ENGINEERING DESIGN CONSIDERATIONS IN THE NOISE CONTROL OF COMMERCIAL JET AIRCRAFT'S VENT AND DRAIN SYSTEMS."

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INTRODUCTION

With the sharp rise and increasing public interest in air travel as a preferred mode of modern day transporation, both the aircraft-manufacturers and commercial airlines alike have intensified their effort in building and operating spacious, comfortable and quiet jet airplanes. Because of the rapid growth in the number of commercial jet transports roaming the globe and the fantastic increase in the frequency of their operations from thousands of aiports during this past decade, "aircraft noise" has become one of the most serious environmental pollutant as it concerns the residential communities in the vicinity of these airports. Significant progress has been made by the noise control technology and the aircraft operating procedures in order to reduce the harmful effects of 'jet noise' to the people who are constantly exposed to it. Strict federal standards and noise ordinances have definitely improved the impact of jet noise to surrounding communities, near the airports. However, passengers flying in these new and advanced jet airplanes also need special attention in terms of quiet and comfortable cabins. 'Interior cabin noise and its control' has been given serious consideration by both the airlines and the aircraft manufacturers these past few years, mainly out of competition, but also reflecting the technological progress made in the state of art.

The object of this study was to identify various sources of interior cabin noise and then to apply noise-control technology to reduce the intensity of one of these sources, namely, the noise generated by jet aircraft's vent/drain/exhaust system. Unlike the air conditioning distribution duct noise (Reference 1) and the equipment noise (Reference 2) within the passenger cabin, the vent/drain/exhaust noise is difficult to identify and even more complex in nature to control.

DISCUSSION

The present study describes some of the engineering design considerations undertaken to control unacceptable and unsatisfactory noise levels prevalent in the cabin lavatories and toilet rooms of a modern jet airplane. The main source of the objectionable noise in these frequently occupied areas is generated by the operation of exhaust air-vent and water drain systems, including the toilet flush mechanisms. The stringent requirements of noise-level criteria within the passenger cabin made this work even more demanding and difficult.

A specially built effective acoustical muffler was designed for controlling exhaust and intake noise levels. The main features of the proposed muffler-design were two-fold, namely, that it was extremely effective in reducing medium and high-frequency (500 Hz to 4000 Hz) noise levels, thus bringing down the Speech Interference Level (SIL) by as much as 15 dB from the previously unacceptable value; and because of its compact size and simple design, the proposed acoustical muffler was easy to install, in addition to the fact that the fire-retardant acoustical lining material used in the muffler could be replaced without much difficulty. One of the major causes of prevailing high noise condition in the vent and drain system was a small restriction downstream of the existing cartridge, gives rise to a near sonic condition. In view of the complexities of the existing undesirable system configuration, each component was individually examined and tested, prior to the installation of an improved system, both mechanically and acoustically.

TEST PROCEDURE AND RESULTS

A mock-up of the existing vent/drain system was built with necessary components and connecting tubing (Figure 1) for acoustical evaluation in the laboratory. A test muffler was designed and tested with the original cartridge for its acoustic perfromance. A 1/2-inch thick open-pore polyurethane foam lining material was used inside the cartridge such that it covered the entire length of the cartridge upto its neck (Figure 2). The muffler was found to be very effective in reducing high frequency sound (as much as 10 dB overall) and hence the SIL range, but still not enough to meet the design specifications as listed below:

SPL in dB - RE 0.0002 dynes/cm^2

OCTAVE BAND (PREFERRED) CENTER FREQUENCY IN HERTZ

OVERALL	63	125	250	500	1000	2000	4000	8000
78	71	73	72	72	69	65	57	52

SIL ≈ 64 dB (Max.)

In view of the shortcomings of earlier design, it was necessary to redesign the original cartridge with its muffler, and change the dimensions of the connecting tubing without altering airflow pressure requirements. In addition to the newly designed cartridge and the muffler, the test set-up consisted of a standard drain mast (exhaust sink), three different kinds and thicknesses of acoustical panels for use as cabin walls housing the entire system, and different I.D.s of neoprene and flexible wire-reinforced connecting hoses. The airflow parameters used in the test simulated actual airplane design conditions for the lavatory vent/drain system. These were 16, 23 and 35 CFM for the front, middle, and aft lavatories respectively, while the corresponding values at the drain mast location were 75, 70, and 95 CFM respectively. Four differenct restrictor sizes were used in the side water-drain tube (Figure 5), ranging from 1/8-inch to 1/2-inch (with 1/4-inch increments). The three types of cabinet wall panels used were 1/4-inch and 1/2-inch thick Nomex honeycomb panels and a 1/2-inch thick Standard plywood panel to enclose the system in a cabinet that is representative of the actual airplane lavatory interior.

The sound measurements were recorded primarily at two different locations, one at 6" from the sink overflow end, and the other at 18" from the cartridge at 45°, with and without the panels installed. The test results are shown in the attached diagrams (Figures 3 & 4). The optimum acoustical system that met required noise level criteria is shown (Figure 3) along with its acoustical performance and dB attenuation.

CONCLUSION

It is clear from the test results of various system modifications that:

1) the noise levels obtained for each of the test configurations tested are unacceptable without the muffler in the cartridge, but do satisfy the acoustical requirements when the muffler is included as part of the system design,

2) either of the three wall panels is acceptable when used with the cartridge-muffler combination in the system,

3) an 1-inch I.D. neoprene tubing throughout the system with any of the resticitors ex-

cept 1/8-inch diameter not only meets design specifications but also satisfies aerodynamic requirements, and

4) either Y or T-connector with a minimum of 1-inch spacing from the cartridge-muffler system was found to be effective in reducing the lavatory vent/drain system noise, thus providing the required 'choking' condition at the drain mast @ 95 CFM airflow.

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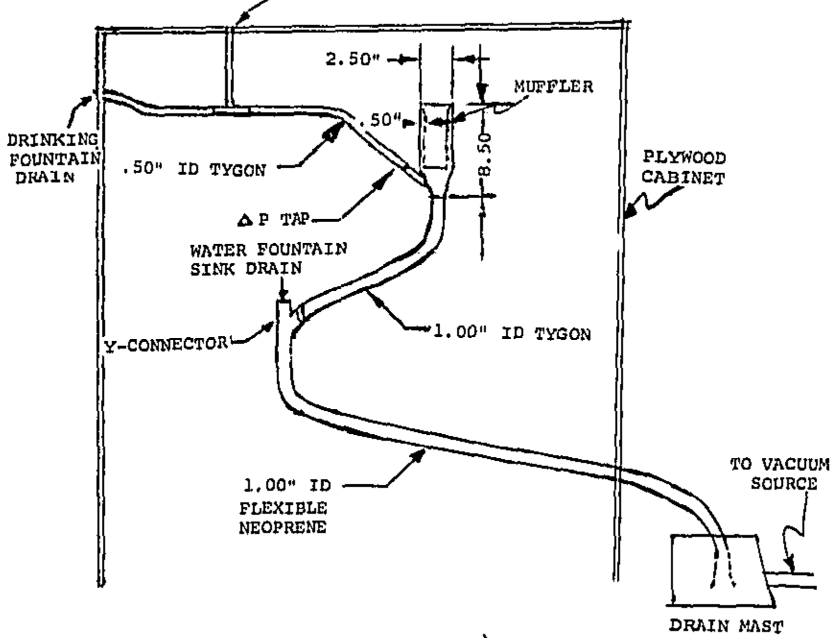
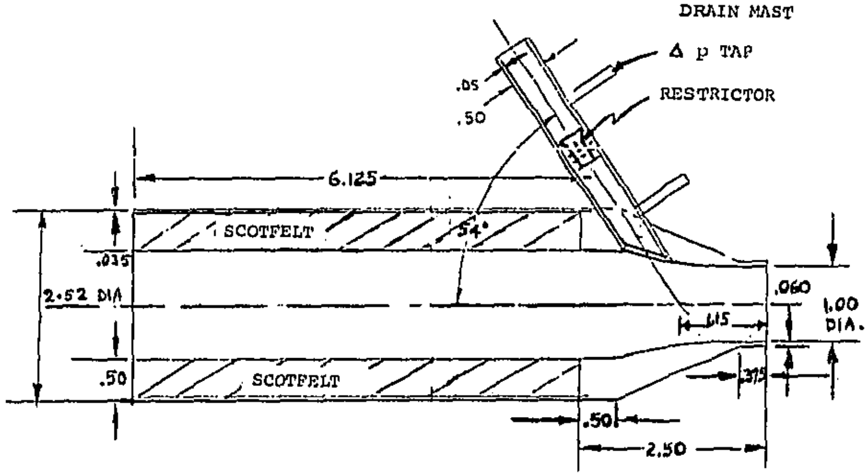
Figure 1-Schematic of the Test Set-up for Acoustical Evaluation of Vent/Drain System



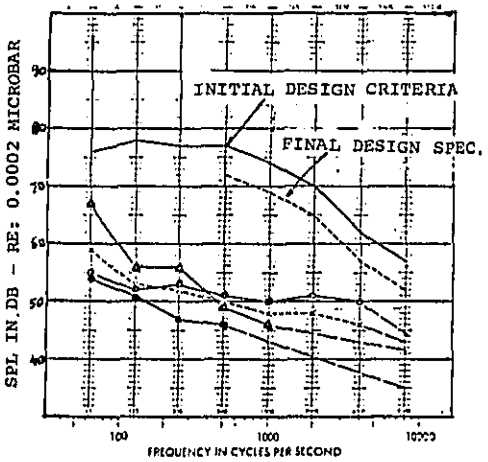


CARTRIDGE MUFFLER TEST SECTION DIMENSIONS

Figure 2-Cartridge/Muffler Test Section Geometry

Complete System Figure 3-Acoustical Performance of the Proposed Muffler in the Vent/Drain System.

Muffler in the Cartridge; 1/2" Thick Nomex Honeycomb Panel on; 3/8" Orifice; and Y-Connectors in the System

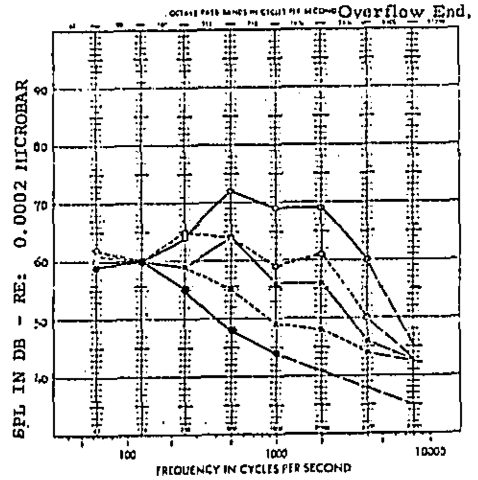


• Ambient
▲ 35 CFM; Mike 18" from cartridge
o 35 CFM; Mike 6" from cartridge
x 35 CFM; Mike 6" from the Water Fountain Opening

(Mid Toilets-35 CFM Max. Airflow Per Lavatory Unit)
Length of the Connecting Hose 20 Ft.

Figure 4- Acoustical Evaluation of the Exposed Vent/Drain System with the Proposed Muffler.

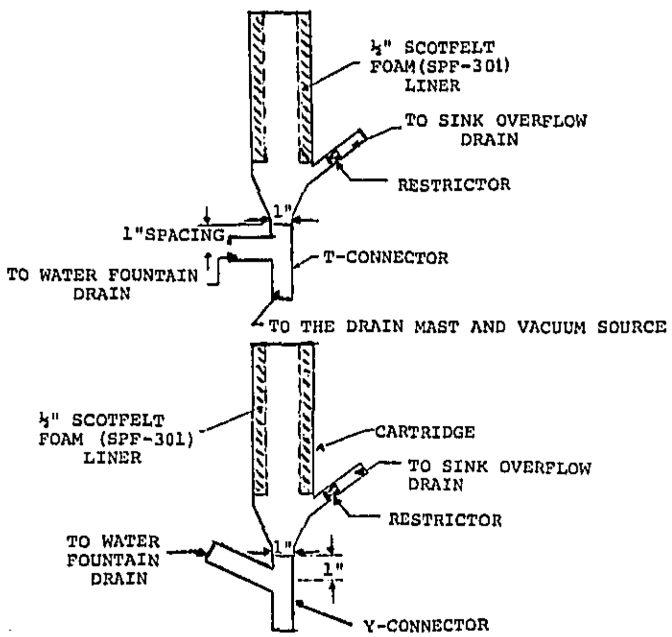
No Lid Cover (I. E. Cabinet Wall)-Mike 6" from the Overflow End.



Ambient
23 CFM-Muffler in cartridge
23 CFM-No Muffler in cartridge
35 CFM-No Muffler in cartridge
35 CFM-Muffler in cartridge

70 CFM at the Drain Mast; 1/8" Orifice in the 1/2" Side Tube (Water Drain)

Figure 5-Cartridge/Muffler Configuration with Related System Components



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