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Effect of Acoustical Engine Enclosures On Truck Cooling System Performance

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16. Abstract

The airflow through a truck engine compartment consists of a complex system of series and parallel flow paths. This system can be modeled in terms of a network of equivalent ductwork elements. The detailed description of engine compartment airflow provided by this model can be used as a means of predicting the effect of acoustical engine enclosures when the ductwork element parameters are altered. The initial implementation of this model will require the performance of a number of component and truck wind tunnel tests; this data should be obtained to permit the use of the model. In-service monitoring of either engine or gearbox temperature performance requires the recording 8 to 10 parameters plus clock time for an adequate description of vehicle behavior. This monitoring should be initiated at the initial vehicle startup and continued throughout the day, including periods of engine shutdown. A total daily test period of approximately 10 hours is expected. The sampling data rate of 1/min for each parameter is sufficient for monitoring purposes. This monitoring requirement can be meet by a relatively inexpensive data logger plus suitable transducer signal conditioning.

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EFFECT OF ACOUSTICAL ENGINE ENCLOSURES ON TRUCK COOLING SYSTEM PERFORMANCE

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PREPARED UNDER: CONTRACT NO. 68-01-6154, TASK ORDER T8
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SUMMARY

Quieting internal combustion engine noise from highway vehicles is currently accomplished by means of acoustical engine enclosures. These enclosures work by restricting the noise around engines from escaping outside the vehicle. Unfortunately, these enclosures also have the effect of restricting the cooling airflow traveling through the engine compartment. The purpose of this study is to describe engine compartment airflow and the effect of acoustical enclosures, and to identify instrumentation which can be used to monitor truck cooling system performance.

The airflow through a truck engine compartment consists of a complex system of series and parallel flow paths. This flow enters through the grill and radiator, is boosted by the engine cooling fan, and is then split into three parallel paths: one passing beneath the engine; another to the sides of the vehicle, exiting through the wheel wells; and the third passing by the engine, exiting beneath and/or behind the truck cab. Through each of these paths a series of pressure drops are experienced as a result of various airflow path elements. These pressure drops must be counter-balanced by the pressure rises obtained from the aerodynamic ram air pressure and the engine cooling fan.

The engine compartment airflow can be modeled in terms of a system of equivalent duct-work elements. The pressure changes due to these elements can be quantified by a combination of duct design equations and empirical measurements on trucks and truck components. The detailed description of

engine compartment airflow provided by this model can be used as a means of predicting the effect of acoustical engine enclosures when the duct-work element parameters are altered. Relatively little information has been published to date describing in detail engine compartment airflow. Therefore, the initial implementation of this model will require the performance of a number of component and truck wind tunnel tests.

In-service monitoring of either engine or gearbox temperature performance will require the recording of eight to ten parameters plus clock time for adequate description of vehicle behavior. This monitoring should consist of time-series recording of parameters originated at the initial vehicle start up and continue throughout the day, including periods of engine shutdown. A total daily test period of approximately 10 hours is expected. The data sampling rate of 1/min for each parameter is sufficient for monitoring purposes. This monitoring requirement can be met by a relatively inexpensive data logger plus suitable transducer signal conditioning. This device will permit data recording for the daily test period without vehicle driver attention.

This study recommends that airflow model element data be obtained to permit use of the model. This data to be obtained includes: the volumetric airflow through the engine compartment airflow paths, and the path and exit resistances for each path; the fan performance of commercially available fans, measured using consistent test procedures; and the effect of inlet (shroud) and outlet (diffuser) configurations using consistent test procedures -- including high ram air, low fan speed conditions. This study also recommends that the airflow model should be implemented into a computer program which can be then used to perform parametric studies which would quantify engine enclosure effects and permit enclosure optimization.

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NOMENCLATURE

A	cross-sectional area
C	units constant
D	diameter
F	fan
K	flow resistance coefficient
L	equivalent duct length
Q	airflow volume
R	per cent open area
U_T	fan tip speed
Re	Reynold's number, $\frac{vD}{\nu}$
f	friction factor
k	resistance factor
p	pressure
t	grill thickness
v	velocity
Δ	change between two points
ϕ	fan flow coefficient, $\frac{Q}{A_F U_T}$
ψ	fan pressure coefficient, $\left(\frac{\Delta p}{\frac{\rho U_T^2}{2}}\right)$
ϵ	duct roughness dimension
ρ	mass density
ν	kinematic viscosity

I. INTRODUCTION

The primary component sources of noise on a truck are the exhaust system, engine block, engine cooling fan, and intake. The exhaust and intake systems are quieted relatively easily by means of improved silencing components. Current practice for reducing fan noise exposures is to use thermostatically controlled fans (although this approach does not quiet the fan when it is operating). Engine noise -- noise radiated by the engine block due to the combustion explosion and piston slap excitations transmitted through the engine block -- is currently reduced by enclosing the engine or otherwise sealing the engine compartment.

Acoustical engine enclosures typically consist of: bottom (belly pan) covers below the engine and transmission, side panels or flaps to obstruct noise propagation through the wheel wells, and -- particularly for cab-over-engine (COE) configurations -- covers behind the cab above the engine and transmission. Typically the panels and some interior surfaces of the engine compartments are lined with sound absorptive material to reduce reverberation within the engine compartment. Essentially, the enclosure acts as a duct. It is open at the front end for the radiator air flow and at the rear for the airflow exit and drive shaft penetration.

The engine enclosure tends to restrict, thus reduce the cooling airflow through the radiator and around the engine and gearbox. The reduced airflow affects vehicle cooling both with and without fan operation. Fan-assisted cooling may require a higher performance fan and/or radiator to meet cooling system design specifications. Higher performance fans are more expensive, consume more fuel, and/or larger--resulting in layout problems.

Cooling normally performed without fan assistance is reduced. The consequent additional fan usage results in increased fuel consumption and fan noise. Furthermore, engine and gearbox cooling normally accomplished by heat transfer directly from the engine block and gearbox casing is also affected.

The purpose of this report is to describe and discuss engine compartment airflow, establish the framework of a semi-analytical model describing this airflow, review the effect of acoustical engine enclosures on this airflow, and identify and specify additional data required for such a model to be measured. In addition, instrumentation which can be used to monitor truck cooling system performance during in-use operation is specified.

II. ENGINE COMPARTMENT AIRFLOW

The great majority of medium and heavy duty trucks are equipped with water-cooled engines with air-to-water heat exchangers or radiators. Air enters the front of the truck through the grill and passes through the radiator and fan. On leaving the fan, the heated air is largely unguided and swirls around inside the engine compartment. The air eventually leaves the engine compartment through openings to the exterior. A sketch of the cooling airflow through a typical conventional-cab truck is shown in Figure 2.1. This sketch is an elaboration on the single flow path of Rising.[1] Engine compartment openings combined into three groupsare indicated on the sketch: vertical airflow, lateral airflow, and longitudinal airflow. Vertical airflow exits the bottom of the engine compartment toward the road surface. Lateral airflow escapes through wheel wells or other side openings. Longitudinal airflow moves past the top and sides of the engine, past the transmission, and exits below or behind the cab. The figure shows no air recirculating from the fan outlet into the grill since most modern trucks are adequately baffled to avoid this problem with the minimum speed possible at full power. Although this sketch shows a conventional cab truck, the airflow would be no different in concept for a cab-over-engine truck.

AIRFLOW MODEL

The airflow through the engine compartment can be visualized as that of an equivalent duct system with the vehicle components modeled by duct-work

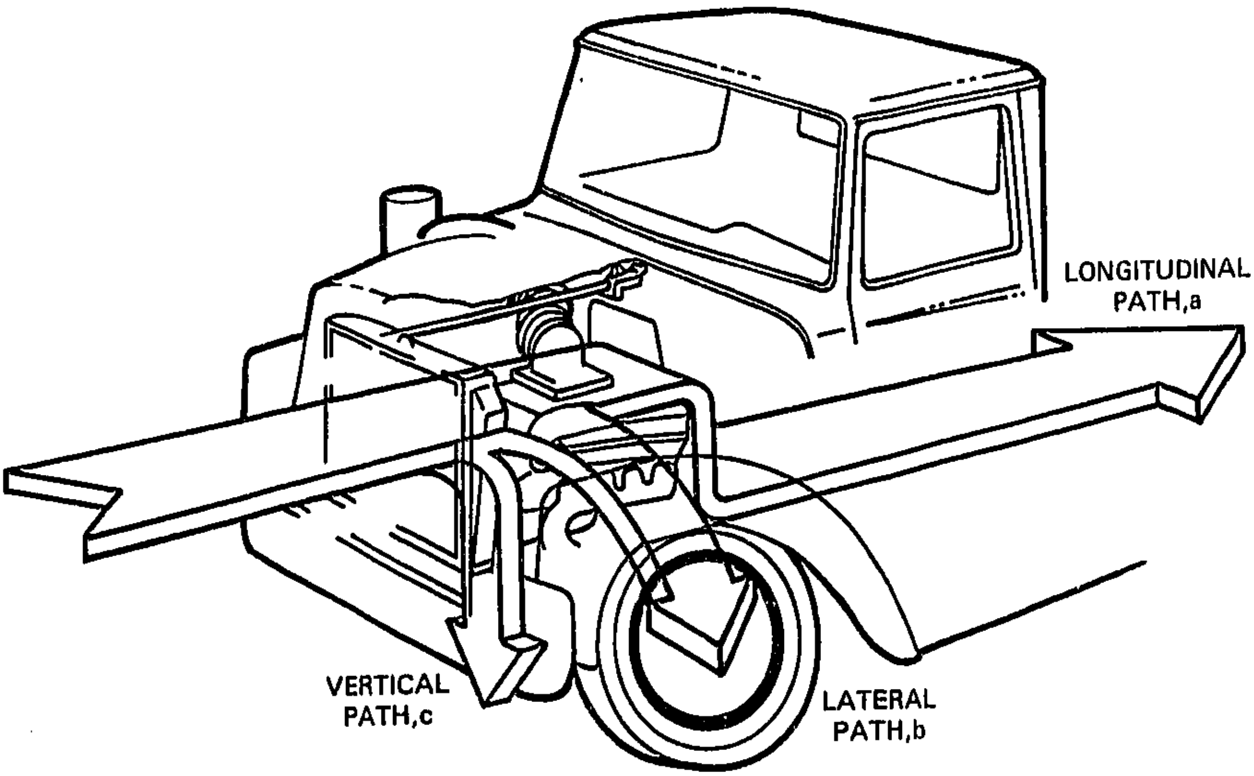


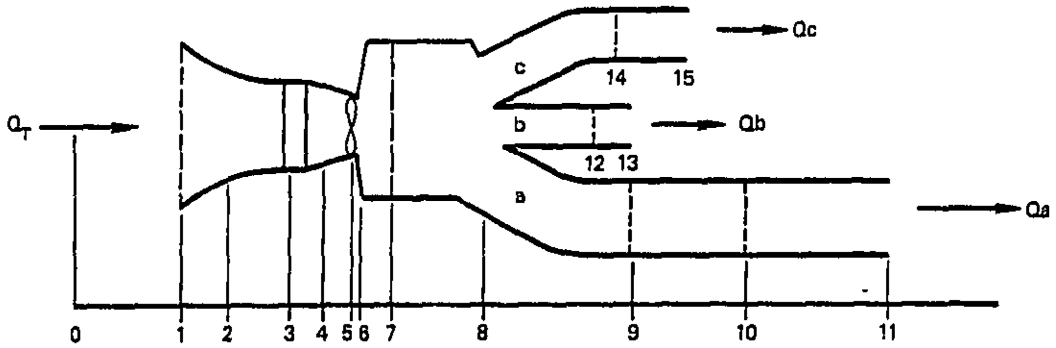
FIGURE 2.1. ENGINE COMPARTMENT FLOW PATHS

components. The flow paths of Figure 2.1 are elaborated upon in a schematic diagram of the cooling airflow given in Figure 2.2. In this diagram, each part of the cooling air path has been assigned a number. The ram air head occurs at Station 0 just upstream of the grill. Each piece of hardware downstream then has a higher sequential number. For example, the grill is assigned Station 1 with the grill-to-radiator ducting assigned Station 2. The multiple flow paths out of the engine compartment have been combined into the three groups identified earlier. Pieces which would exist in both conventional and acoustically treated vehicles are shown. For example, the longitudinal flow path (path a) includes an extra duct element with an added duct resistance representative of the transmission covers which may be part of a quieted design. Further, a quieted design is likely to limit airflow to only one of the three production vehicle flow paths shown. This can be described in the airflow model by capping the exit of the eliminated paths.

Several mathematical expressions can be constructed based upon the relationships shown in the model. Conveniently, the model can be thought of as representing a series of static pressure changes.

$$\Delta p_T = \sum_{i=1}^{n} \Delta p_i$$

Note that all the pressure changes are negative -- i.e., pressure drops; with the exception of stations 0 and 5, the ram air and engine fan pressure rises and that the sum of the pressure rises must equal the sum of the pressure drops. However, with a thermostatically controlled fan drive, the fan may also provide a pressure drop depending upon the driven speed of the fan and the available ram air head.



0 ram air
1 grill
2 radiator inlet
3 radiator
4 fan inlet
5 fan
6 fan outlet
7 engine front path
8 flow splitter, a-longitudinal, b-lateral, c-vertical
9 engine side
10 engine rear
11 longitudinal (rear) outlet
12 wheel well
13 lateral outlet
14 belly pan
15 vertical (bottom) outlet

FIGURE 2.2 ENGINE COMPARTMENT AIRFLOW MODEL

From conservation of mass,

$$Q_T = Q_a + Q_b + Q_c$$

and, since

$$Q = Av,$$

$$A_1 v_1 = A_3 v_3 = A_5 v_5, \text{ etc.}$$

Thus, ideally, the local airflow at any station is known if the effective duct cross-sectional area is known. However, in practice the engine compartment presents a very cluttered and complex flow path. As a result, local airflow velocities should most properly be determined by measurement.

The pressure drops can also be considered in terms of flow resistances,

$$K_i = 2 \Delta p_i \Big/ \rho_i v_i^2$$

where v_i is the local airflow velocity at Station i. This characterization is convenient since the value of resistance coefficient is independent of local flow velocity. The individual flow resistances which make up the total system resistance can be obtained in a number of ways. Some data is available from wind tunnel tests of individual components. Other data can be estimated from previous tests of geometrically similar components. Data on the performance of complex sections can only be obtained from full-scale tests on similar vehicles or even the vehicle in question. In terms of flow resistances, the engine compartment airflow is described as

$$K_T = K_1 + K_2 + K_3 + K_4 + K_5 + K_6 + K_7 + K_8 + \left[\frac{1}{K_9 + K_{10} + K_{11}} + \frac{1}{K_{12} + K_{13}} + \frac{1}{K_{14} + K_{15}} \right]^{-1}$$

where some of the resistances are in series and others are in parallel.

As stated previously, the pressure rise provided by the fan and ram air effect must match the pressure drops incurred by the rest of the system. This means that the engine fan must provide a specified pressure rise at the volumetric airflow required to maintain adequate engine cooling. This fan airflow performance is the operating point of the fan. Figure 2.3 shows the changes in performance from wind tunnel tests to an actual vehicle.[2] The actual fan operating point is not the intersection of wind tunnel radiator core resistance and fan performance curves, but must be corrected to include vehicle resistance from the grill and engine compartment and fan losses. The resistance of the grill, inlet, and engine compartment is equal in magnitude to the resistance of the core in this case. Modern heavy trucks may have total system resistance over four times radiator core resistance alone.[1]

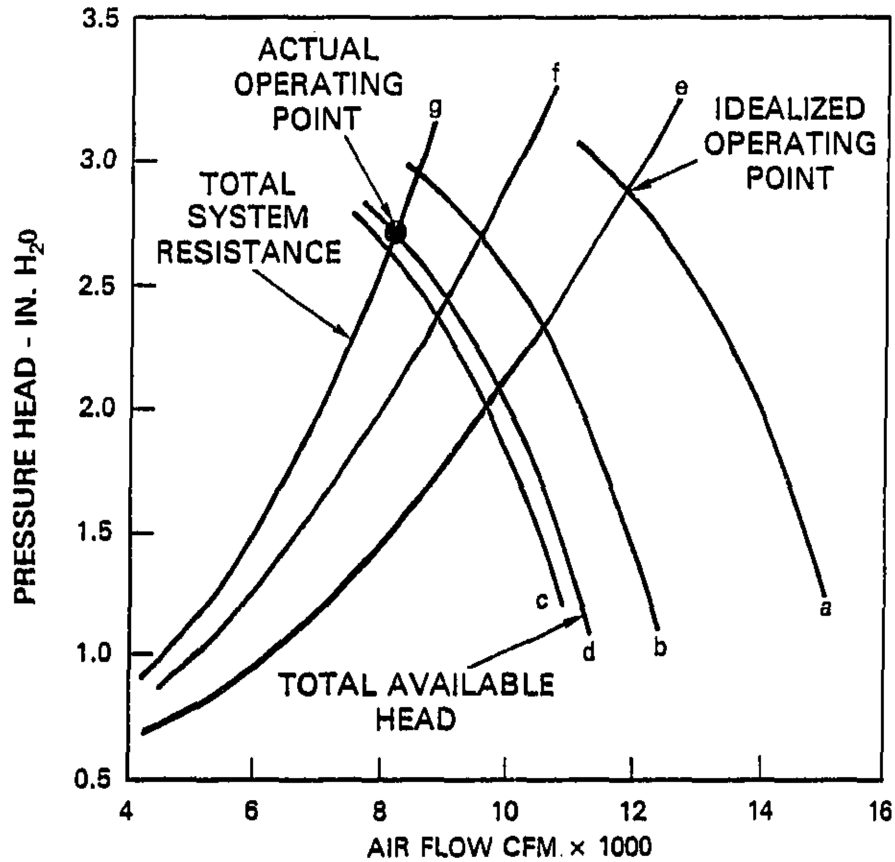
In the following paragraphs, the pressure changes at each station of the airflow model will be discussed. Each of these pressure change equations assumes that consistent units are used. For example, pressure in lb/ft^2 requires that density be in $slugs/ft^3$. If density is given in pounds $mass/ft^3$, it must be divided by the acceleration of gravity, 32.2 ft/sec^2 to convert this to $slugs/ft^3$. Velocity must then be in ft/sec. Pressure in lb/in^2 requires a constant of 144 in^2/ft^2 be added to the denominator of the right side of the equation. Pressure expressed as head ($in.H_2O$) requires a density correction between air and water of 0.075/62.4 as well as 12 in/ft. The pressure rise is divided by fluid weight density to get head.

Ram Air Pressure -- Station 0. Pressure increase due to difference between velocity of vehicle and flow-path velocity,

$$\Delta p_0 = \rho_0 v_0^2 / 2$$

Grill -- Station 1. Pressure drop due to grill open area, thickness, and inlet and outlet geometry,

$$\Delta p_1 = - \left[K_{1a} + (0.45/R + 0.024t)\ K_{1b}\right] \rho_1 v_1^2 / 2$$



a	Fan curve in wind tunnel, 0.125 in. tip clearance
b	Fan curve corrected for effect of radiator and engine front
c	Fan curve corrected for radiator, engine front and shroud with 1 in. tip clearance
d	Ram air added to fan curve
e	Radiator resistance, 3 in. core, 11 fins/in.
f	Radiator plus grill and entrance losses
g	Radiator with grill, entrance, and engine compartment losses

FIGURE 2.3 SYSTEM CHART

where,

Δp_1 = grill pressure drop

K_{1a} = grill entry factor = 0.5 for a flush grill

K_{1b} = discharge factor $\propto$ (unobstructed duct length downstream of grill)$^{-1}$

= 2.75, for 6 in. duct

R = percent open area

t = grill thickness

v_1 = velocity upstream of grill

Radiator Inlet -- Station 2. Pressure drop due to radiator inlet transition (see p. 14, Ref. 3), a function of the transition piece geometry,

$$\Delta p_2 = -(K_{2a} + K_{2b})\, \rho_2\, v_2^2/2$$

where,

K_{2a} = 0.95, for pipe opening

K_{2a} = 0.87, for bulkhead opening

K_{2a} = 0.45, for 5° cone with 30 percent area reduction

K_{2a} = 0.10, for bellmouth opening

K_{2b} = L/39D duct length

Radiator — Station 3. Pressure drop due to radiator, a function of core type and size and fin density (i.e., number of fins/in.),

$$\Delta p_3 = -K_3\, \rho_3\, v_3^2/2$$

K_3 = f(core type, fins density, core thickness) from manufacturer's wind tunnel test.

Examples of manufacturer's radiator curves are given in Figures 2.4.

Fan Inlet -- Station 4. Pressure drop due to radiator-to-fan transition (p. 13, Ref. 3)

$$\Delta p_4 = -K_4 \rho_5 V_5^2/2$$

K_4 = f(radiator-to-fan distance, shroud type)
= 0.05, for a box shroud

Discussion of the shroud flow resistance is meaningful only when the fan is not operating. When the fan is operating, a properly designed shroud greatly increases fan efficiency, thus increasing fan pressure. The benefit of the shroud is a function not only of the shroud design itself (i.e., fan blade tip-to-shroud distance, fan coverage by shroud, and shroud type -- box, ring, or venturi) but is also dependent upon the fan design. A more sophisticated fan (e.g., one with twisted blades with air foil sections) will exhibit a greater performance improvement with a well designed shroud than a relatively crude fan.[4] For the purpose of the airflow model when the fan is operating, the effect of the shroud can be considered a pressure rise, $\Delta \bar{p}_4$, or --more typically-- can be included in the fan performance, ϕ and ψ, with $\Delta \bar{p}_4$=0.

Fan -- Station 5. Pressure change due to fan,

$$\Delta p_5 = \frac{\rho_5 C Q_T^2}{d^4} \quad \frac{\psi}{\phi^2}$$

where,

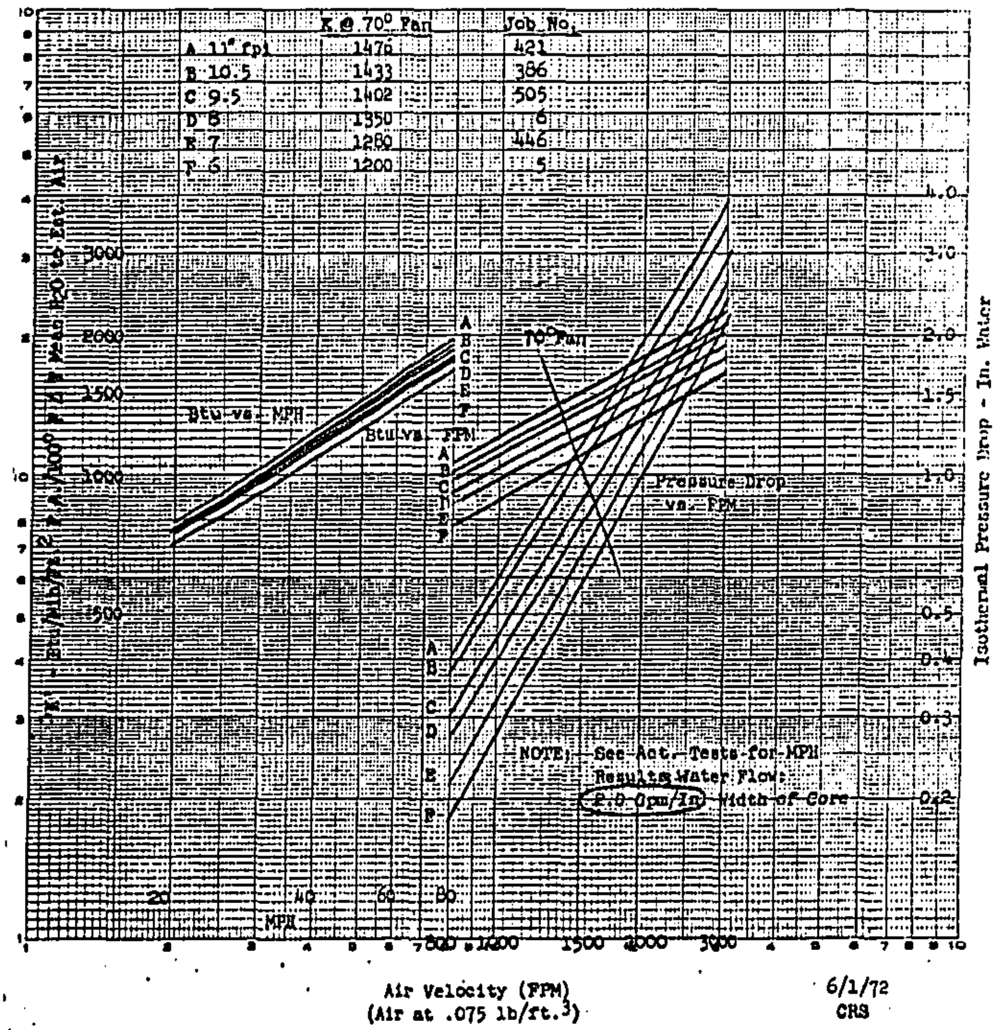
C = dimensional constant, e.g., $C = 0.895$ in.H_2O-in.^{4}-min.^{2}/slugs-ft^3
Q_T = required volumetric airflow,
d = fan diameter,
ψ = fan pressure coefficient, and
ϕ = fan flow coefficient.

The fan must be selected to meet operating point with ϕ, and ψ from dimensionless fan curves. Note that ψ may be negative for declutched fans at high vehicle speeds.

Fan Outlet - Station 6. Pressure drop due to fan discharge into plenum (i.e., engine compartment),

THE G & O MFG. CO.
NEW HAVEN, CONNECTICUT

2.333 T-9 Performance Curves
(4-Row)

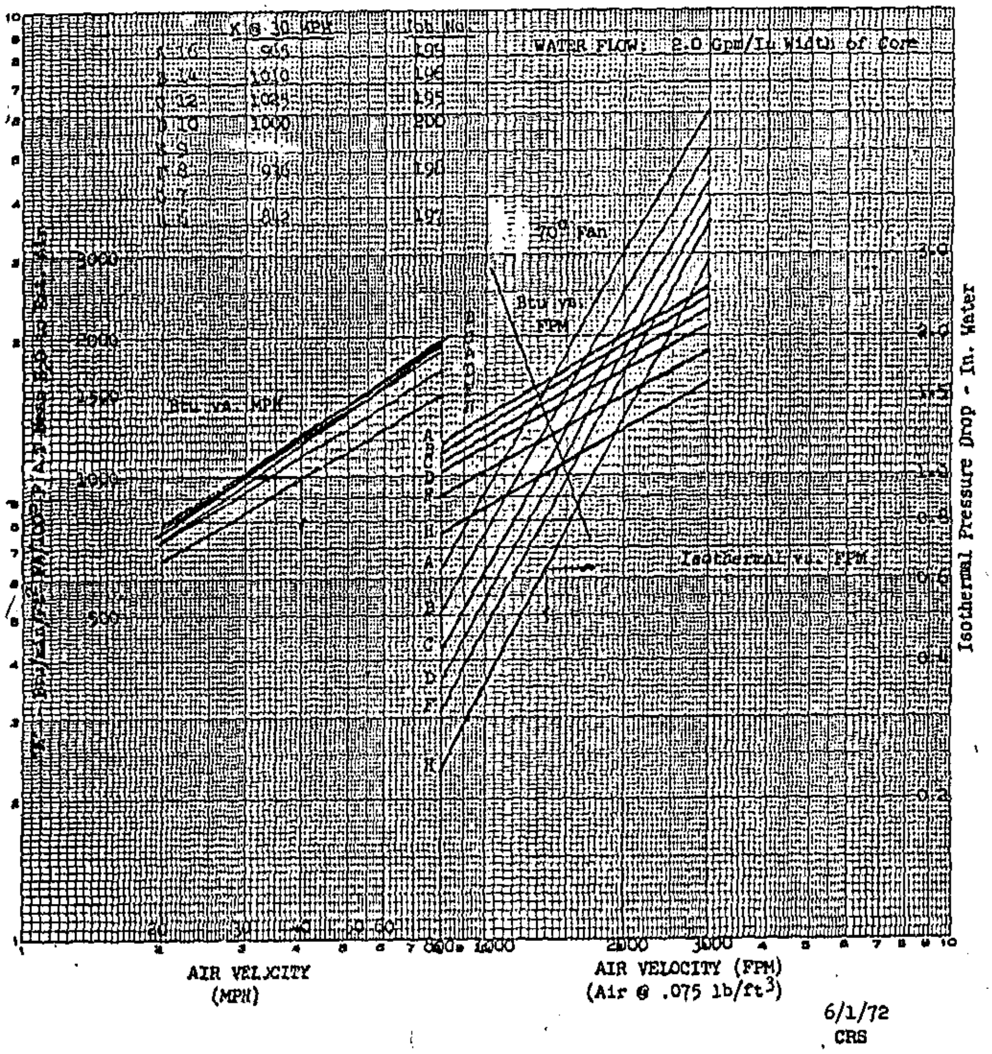


EXAMPLES OF MANUFACTURER RADIATOR PERFORMANCE CURVES

FIGURE 2.4a. FORD TYPE -- LOUVERED PLATE FINS

THE G & O MFG. CO.
NEW HAVEN, CONNECTICUT

2 5/8 VTR (4-Rows)



EXAMPLES OF MANUFACTURER RADIATOR PERFORMANCE CURVES

FIGURE 2.4b. GENERAL MOTORS TYPE -- SIDEFLOW -- LOUVERED SERPENTINE FINS

$$\Delta p_6 = - K_6 \rho_5 v_5^2/2$$

$K_6 \cong 1$ typical in current trucks

$K_6 = 0\text{-}1$ with a diffuser

$K_6 = 0.3$, for a 30° expansion angle to doubled cross-sectional area (p. 13, Ref. 3)

A diffuser will recover some of the fan velocity head and reduce fan outlet pressure drop.

Engine Front Path — Station 7. Pressure drop due to engine front-mounted equipment,

$$\Delta p_7 = - K_7 \rho_7 v_7^2/2$$

K_7 is a function of equipment size, shape, and distance from fan and must be determined experimentally.

Flow Splitter — Station 8. Pressure drop due to flow splitting,

$$\Delta p_8 = - K_8 \rho_8 v_8^2/2$$

K_8 is a function of flow path geometry and must be measured experimentally.

Engine Side — Station 9. Pressure drop due to engine side resistance,

$$\Delta p_9 = - K_9 \rho_9 v_9^2/2$$

$$K_9 = \frac{fL_9}{D_9}$$

where,

L_g = path length past engine
D_g = effective diameter of "duct" passing the engine
f = friction factor = $f(R_e, \varepsilon/D_g)$
$R_e = v_g D_g/\nu$

ε = equivalent roughness dimension
ν = kinematic viscosity

Practically, K_g must be determined experimentally. "Cleaning" the flow path by placing wiring and tubing in trays or fairing engine-mounted accessories could reduce this resistance in an engine compartment.

Engine Rear Path -- Station 10. "Duct" losses,

$$\Delta p_{10} = - K_{10}\, \rho_{10}\, v_{10}^2/2$$

$$K_{10} = k_{10} \frac{L_{10}}{D_{10}}$$

k_{10} = 0.02 - 0.05
= 0.025, for smooth ducting
L_{10} = path length
D_{10} = effective diameter of "duct"

Longitudinal (Rear) Outlet -- Station 11. Pressure loss due to outlet expansion,

$$\Delta p_{11} = - K_{11}\, \rho_{11}\, v_{11}^2/2$$

where,

K_{11} = 1.17, for open pipe
K_{11} = 1.17, for bulkhead opening
K_{11} = 0.8, for cone with 10° expansion angle and 10 percent diameter increase

Wheel Well -- Station 12. Pressure drop through the wheel well and side openings,

$$\Delta p_{12} = - K_{12} \rho_{12} v_{12}^2 / 2$$

K_{12} = function (contraction rate) = 0.6, for a sharp-edged orifice Practically, K_{12} dictated by tire clearance, springs, and local geometry and must be measured experimentally. A velocity traverse of the cross section leading to the wheel well and a traverse at the wheel well edge would define this loss. Acoustic panels blocking lateral flow would make this resistance infinite.

Lateral Outlet — Station 13. Wheel outlet loss,

$$\Delta p_{13} = - K_{13} \rho_{13} v_{13}^2 / 2$$

where:

v_{13} = average wheel outlet velocity.

K_{13} = the loss of the velocity pressure plus transition loss and is a function of outlet geometry similar to K_{11}.

Belly Pan -- Station 14. Pressure drop past the belly pan,

$$\Delta p_{14} = - K_{14} \rho_{14} v_{14}^2 / 2$$

K_{14} is similar to K_{12} and must be measured experimentally. If a tight-fitting acoustical belly pan blocks all flow, the resistance is effectively infinite.

Vertical (Bottom) Outlet — Station 15. Bottom outlet loss,

$$\Delta p_{15} = - K_{15} \rho_{15} v_{15}^2 / 2$$

where, v_{15} = average bottom outlet velocity, and K_{15} is similar to K_{11} and K_{13}; this loss would include velocity pressure and transition loss.

EFFECTS OF ENGINE ENCLOSURES

For an engine enclosure to be effective it must obstruct the propagation of engine noise to sound receivers around the truck. Since sound propagates efficiently through even relatively small openings of an enclosure, the ideal noise control enclosure is perfectly air tight! Clearly, engine enclosures are not perfect acoustical enclosures; however, to achieve minimally acceptable enclosure quieting performance, some restriction of the engine cooling airflow is expected.

Evaluation of engine enclosure effects on airflow is essentially speculative at this time since virtually no detailed quantification of engine compartment airflow has been performed. In general, cooling performance is expected to suffer somewhat as a result of the enclosure restriction. However, some benefits may be derived by the rationalization of the flow through the engine compartment — by the elimination of the parallel paths — and potential improvement of the air flow discharge. In the following paragraphs, the enclosure effects will be briefly discussed with respect to the engine compartment airflow model. The airflow element resistances through to the fan discharge (Station 6) are expected to be essentially unchanged by the installation of the acoustical enclosure itself — although additional cooling capacity may be necessitated to compensate for enclosure effects.

<u>Engine Front Path—Station 7</u>. The enclosure panels may add clutter to the flow path resulting in increased resistance at this station. However, the "clean up" of accessories, wires, and so forth may provide less flow obstruction and decreased resistance.

<u>Flow Splitter—Station 8</u>. The enclosure is likely to block the lateral and vertical flow paths (paths b and c) eliminating the flow splitting loss and directing the entire airflow through the longitudinal path (path a). If the lateral and vertical paths had been carrying significant portions of the airflow, the flow velocities in the longitudinal path will increase significantly to carry the increased volumetric flow. This will result in a substantial increase in pressure drop at each of the downstream stations through the path a since $\Delta p \propto v^2$.

Engine Side--Station 9. In addition to the increased pressure drop resultant from increased flow velocities, the path resistance is likely to increase as a result of the probable decrease in the effective duct diameter due to the enclosure. However, some benefit may be obtained in reducing the friction factor due to the "cleaning" of the engine compartment by placement of wires in trays and so forth.

Engine Rear Path--Station 10. Since the enclosure is likely to constrain the flow over a longer longitudinal distance, path resistance is likely to increase as a result of the increased duct length as well as a likely decrease in the effective duct diameter.

Rear Outlet--Station 11. The enclosure is likely to have two effects at this station. One effect is the more abrupt opening of the flow path which is likely to result in higher resistance. The other effect is the alteration of the effective location on the path outlet which may have favorable or unfavorable effects depending upon the pressure region to which the duct effectively terminates.

The lateral and vertical paths are likely to be obstructed by the enclosure panels, and, therefore, have essentially infinite resistance and zero flow.

ADDITIONAL DATA REQUIREMENTS

The usefulness of the airflow model depends upon the availability of data to quantify the pressure changes. Relatively little work has been done quantifying even the total engine compartment airflow resistance of highway vehicles. Of all the airflow elements, the radiator is best quantified with available performance data, and the grill performance is relatively readily calculated. Fan performance data, however, while always available to the fan manufacturers, is frequently considered proprietary information. The primary data requirements for the implementation of the airflow model are outlined in Table 2.1. In general, these requirements must be satisfied by a combination of component wind tunnel testing and/or in situ (i.e., in a truck) wind tunnel

TABLE 2.1 AIRFLOW MODEL PARAMETERS REQUIRING QUANTIFICATION

AIRFLOW MODEL ELEMENT	STATION	PARAMETER	COMMENTS
Fan Inlet	4	K_4, radiator-to-fan transition resistance	Measure pressure drop at flow with fan not operating —in wind tunnel. Measure pressure rise benefits with operating fans— in wind tunnel.
Fan	5	φ, ψ, fan performance	Measure under controlled conditions for commercially available fans over a range including negative
Fan Outlet	6	K_6, fan discharge resistance	Measure with and without operating fans in wind tunnel referenced to constant cross-section duct.
Engine Front Path	7	K_7, path resistance	Measure in truck — in wind tunnel.
Flow Splitter	8	Q_a, Q_b, Q_c, flow distribution	Determine flow distribution by measuring average flow velocities at path exits of truck in wind tunnel and/or block path and observe changes.

TABLE 2.1, Continued

Engine Side	9	K_9, path resistance	Measure in truck — in wind tunnel.
Engine Rear	10	K_{10}, path resistance	Measure in truck — in wind tunnel.
Longitudinal Outlet	11	K_{11}, exit resistance	Measure in truck — in wind tunnel.
Wheel well	12	K_{12}, path resistance	Measure in truck — in wind tunnel.
Lateral Outlet	13	K_{13}, exit resistance	Measure in truck — in wind tunnel.
Belly Pan	14	K_{14}, path resistance	Measure in truck — in wind tunnel.
Vertical Outlet	15	K_{15}, exit resistance	Measure in truck — in wind tunnel.

testing. In many cases, the flow element testing must be replicated for both an unquieted and quieted vehicle using the acoustical engine enclosure, although as experience and knowledge are gained, enclosure effects should be quantifiable a priori.

III. IN-SERVICE TEMPERATURE MONITORING INSTRUMENTATION

Trucks fitted with acoustical engine enclosures are typically evaluated in wind tunnel tests to assure cooling performance has not been degraded. These tests consider engine cooling performance but generally are not concerned with gearbox heat rejection. Alternatively, engine and gearbox cooling performance can be evaluated by monitoring the appropriate parameters during the actual in-service use of the vehicle.

Engine heat rejection is a function of engine speed and load. This waste heat is a known fraction of the chemical energy of the fuel burned by the engine. Thus, it can be quantified by means of the engine fuel consumption for the engine speed.

Gearbox heat generation is a result of two mechanisms. The primary mechanism is the frictional heat generation are to the gear meshing -- a function of the input torque transmitted through the gears and the number of gear meshes required for the desired driveshaft speed. The secondary heat generation mechanism is the churning of the gearbox lubricant by the gears and is a function of rotational speed of the gears.[5] The input torque transmitted through the transmission can be related to engine power and engine speed, thus can also be quantified by the measurement of fuel consumption.

RECORDED PARAMETERS AND TRANSDUCERS

Evaluation of engine and/or gearbox cooling performance can be performed by the sampling of the following parameters at regular intervals during vehicle operation:

- For both engine and gearbox cooling evaluations:
  - time of day
  - ambient air temperature
  - engine speed
  - engine fuel consumption

- For engine cooling evaluations:
  - engine coolant temperature(s)
  - crankcase oil temperature
  - engine block surface temperature(s)
  - fan speed.

- For gearbox cooling evaluations:
  - gear
  - gearbox oil temperature
  - gearbox casing temperature(s)

Most of these parameters can be sensed by means of fairly simple transducers and/or instruments. Engine speed can be monitored by accessing the output of the existing electronic engine tachometer. Fan speed requires the installation of a special fan tachometer with an electronic output. Temperature measurements of any type can be monitored by means of either thermocouple or thermistor installations. Gearing can be monitored by means of shift lever-actuated contacts adding or subtracting resistances in an electronic circuit, such that a voltage stepwise proportional to gear is produced.

Two fuel measurement devices are available with accuracy necessary for the purpose of quantifying engine power output. These instruments are manufactured by Fluidyne Instrumentation and VDO-ARGO Instruments, Inc. The

Fluidyne instruments are more accurate, measuring in units of 10^{-3} gal, but expensive, about $4,000. The VDO-ARGO unit costs about $800-$1000 but is considerably less accurate, measuring in units of 10^{-1} gal. It is possible to trade cost for accuracy in both types of instruments. For example, Fluidyne has a system that does not include temperature compensation which is somewhat less expensive than their best system. The VDO-ARGO system is probably capable of producing greater accuracy than is found in their mass-produced device, so it may be possible to customize the installation to achieve greater accuracy at somewhat greater cost.

One matter that should be made clear is that the fuel system of a diesel truck will have to be modified for accurate measurement of fuel consumption. The modification has to be done because the amount of return fuel flow from the injectors has to be taken into account. The details of the modifications are specific to the type of injection system, thus will vary with engine manufacture.

DATA RECORDING DEVICES

A recording system or data logger of the type needed to monitor in-service performance of trucks comprises three components: signal conditioning, a processing device, and a recorder. Signal conditioning is a matter of changing the raw input from a sensor into a form that can be assimilated by the processor. The processor contains programs that determine both the procedure for sampling the input from the sensors (rate of sampling, duration of sample, and order in which the sensors are sampled) and the procedure for recording the sampled measurement. The recorder is the device used to record the measurements. The three components are usually combined into a single piece of equipment for purposes of field measurements.

The data logging system must be capable of recording time-series data. It is not difficult to find inexpensive processors that collect counts of events by category. The counts can be displayed in a histogram so that one might be able to display, say, miles driven in each gear and miles driven while the transmission casing was in various ranges of temperatures. However, this type of data neglects the crucial importance of the dynamic relationships among the various aspects of system performance. For example, the question of whether a temperature exceeded 200° F after ten minutes or ten hours of operation in a given condition would not be answered by counts in categories.

The data logger must also be capable of operating in a relatively confined area of a truck cab for a relatively long period of time with almost no attention or maintenance devoted to it. This specification eliminates from consideration a large number of devices that are suitable only for use in laboratories and other relatively genteel environments.

Specification of a System

Signal conditioning requirements are a function of the particular sensors used. In general, signal conditioning for analog inputs will involve changing resistances so that input voltages will match the range required by the recording device, but conditioning for digital signals may be somewhat more complicated.

The qualities that must be fulfilled by the processor can be fairly well specified in advance. For temperature monitoring purposes, the processor must be capable of sampling the sensor input at a rate of one sample per minute. (One sample consists of one ensemble of measurements from each transducer.) This sampling rate is sufficient to ensure that the recorded data contain sufficient resolution to track the onset and course of meaningful changes in the operating variables of a truck. Since the temperatures of interest vary slowly, greater detail in the time domain would be unnecessary and would waste recording capacity. The sampling rate of one per minute is not a demanding specification for the processors of today.

The previously discussed list of variables to be recorded provides another specification for the processor. Since recording clock time is a standard feature on data loggers, the processor must be able to handle about 8 channels plus elapsed time with sufficient information capacity to furnish the accuracy required by the study. A generous estimate of the information requirement is 100 bits per sample.

The specifications for the recorder follow directly from specifications for the processor and the operating characteristics of trucks. The recorder must be capable of recording data for at least a full day of operation without requiring attention from the driver. In addition, the driver must be able to remove recorded data from the recorder to mail it in preaddressed envelopes. These specifications eliminate all types of recorders except the tape cassette (a disk recorder would be too fragile). The capacity of the tape C_T, can be estimated by:

$$C_T > crt$$

where c = bits/sample
r = sampling rate, and
t = test period.

For a full day sample, including continued sampling during meal breaks, approximately 10 hours of data will need to be recorded. Based upon the previously discussed values of c = 100 bits/sample and r = 1/min, the capacity of one side of the tape must exceed 60k bits, well within the capacity of a 300 ft Philips tape cassette.

Feasible Systems

The data loggers listed in Table 3.1 meet the specifications given in the preceding section, and they are listed in order of increasing cost and increasing certainty of adequate performance. To begin with the highest certainty of successful application, the RCA/Vehicle Monitoring System (VMS) has received the most rigorous testing of all the devices listed. However, the capabilities and cost of the VMS far exceed those needed within the requirements discussed herein. Both the EDMAC and Westinghouse devices have seen use in automotive applications. Their cost, at least in the context of this type of device, could be considered moderate. The EDMAC device might receive a slightly higher ranking with respect to certainty of application because the company's experience in automotive applications is more recent. The EDMAC Model 4600 was designed specifically for this purpose. The A.D. Data Systems device is a relatively inexpensive candidate.

TABLE 3.1

FEASIBLE DATA LOGGER SYSTEMS

MANUFACTURER	MODEL	APPROXIMATE COST	ADVANTAGES	DISADVANTAGES	COMMENTS
A.D. Data Systems, Inc.	ML-10A Weatherproof	$ 3,000	Low cost.	Untried in automotive applications.	Signal conditioning requirements and "debugging" the system may increase the total cost of this device considerably.
Westinghouse	DDS-100	$15,000	Cost includes some signal conditioning.	Manufacturer may be "rusty" on automotive applications, relatively high cost.	Goodyear used this device for experimental tests of racing cars under controlled conditions.
EDMAC Associates, Inc.	Model 4600	$15,000[1]	Manufacturer has current experience in automotive applications.	Relatively high cost.	This device currently being used in a NHTSA study of bus operation.
RCA	Vehicle Monitoring System (VMS)	$80,000[2]	Manufacturer has extensive experience in automotive applications.	Device excessively elaborate and expensive for planned study.	A 10,000 mile test of this device at Aberdeen Proving Ground has been reported.

1 Price paid by Heavy Vehicle Division of the National Highway Safety Administration of twenty-seven units. This price includes development costs.

2 This price includes recurring hardware cost only.

Its price is appealing, but the problems that might arise in adapting it for automotive use are difficult to anticipate accurately. Thus, it is impossible to determine whether the total cost of installing the device and ridding it of defects might be more than the cost of the EDMAC or Westinghouse devices, the cost of which includes at least a modicum of engineering support.

Recommended Device

Given the amount of information at hand, it is not possible to recommend a device unconditionally. The A.D. Data Systems ML-10A is the least expensive system which satisfies the expected test requirements. While it does not contain built-in signal conditioning, a signal conditioning option is available at reasonable cost. Consequently, this system is the first choice for this application. If the ML-10A proves unsuitable, the EDMAC Model 4600 with greater capability and cost is the second choice.

IV. CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

Engine Compartment Airflow

The airflow through a truck engine compartment consists of a complex system of series and parallel flow paths. This flow enters through the grill and radiator, is boosted by the engine cooling fan, and is then split into three parallel paths: one passing beneath the engine; another to the sides of the vehicle, exiting through the wheel wells; and the third passing by the engine, exiting beneath and/or behind the truck cab. Through each of these paths, a series of pressure drops are experienced as a result of various airflow path elements. These pressure drops must be counter-balanced by the pressure rises obtained from the aerodynamic ram air pressure and the engine cooling fan.

The engine compartment airflow can be modeled in terms of a system of equivalent duct-work elements. The pressure changes due to these elements can be quantified by a combination of duct design equations and empirical measurements on trucks and truck components. The detailed description of engine compartment airflow provided by this model can be used as a means of predicting the effect of acoustical engine enclosures when the duct-work element parameters are altered.

Relatively little information has been published to date describing in detail engine compartment airflow. Therefore, the initial implementation of this model will require the performance of a number of component and truck wind tunnel tests.

In-Service Temperature Monitoring

In-service monitoring of either engine or gearbox temperature performance will require the recording of eight to ten parameters plus clock time for adequate description of vehicle behavior. This monitoring should consist of time-series recording of parameters initiated at the initial vehicle start up and continue throughout the day, including periods of engine shutdown. A total daily test period of approximately 10 hours is expected. The data sampling rate of 1/min for each parameter is sufficient for monitoring pruposes. This monitoring requirement can be met by an AD Data Systems, ML-10A data logger plus suitable transducer signal conditioning. This device will permit data recording for the daily test period without vehicle driver attention.

RECOMMENDATIONS

Airflow model element data must be obtained to permit use of the model. Specifically:

- Measure, by means of truck tests in wind tunnels, the volumetric airflow through the three airflow paths and the path and exit resistances for each path.

- Obtain, from fan manufacturer component wind tunnel tests, the fan performance of commercially available fans normalized to consistent conditions

- For each of the commercially available fans, measure the effect of inlet (shroud) and outlet (diffuser) configurations using consistent test procedure including high ram air, low fan speed conditions.

The airflow model should be implemented into a computer program which can be then used to perform parametric studies which would quantify engine enclosure effects and permit enclosure optimization.

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