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Effect of Cooling System Design On Truck Noise

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16. Abstract

Improving cooling system performance, thus, reducing the fan aerodynamic requirement is a key to quieter fans. Relative to an existing gasoline-engined medium truck cooling system, increased top tank temperature permits an airflow reduction estimated to allow an 11 dB reduction in fan noise. Use of a multi-pass radiator in this configuration adds a further 1.5 dB reduction, as estimated in a non-optimized case. Fan aerodynamic and acoustical performance is most significantly improved by reducing blade tip-to-shroud clearance. Fan designs which include integral (rotating) shrouds provide the best aerodynamic performance and least noise. Alternatively, low tip clearance fans using fixed engine-mounted shrouds or radiator-mounted fans (driven by a flexible coupling), will also provide superior performance. Aerodynamic test data provided by manufacturers tends to use tip clearances much smaller than obtainable in practice. Standard test procedures are also unavailable for acoustical testing of engine cooling fans. Consequently, current production fan noise performance is almost completely undocumented. Aerodynamic and acoustical fan test procedures for vehicle applications which reasonably represent installed fans should be developed and performance data on commercially available fans be catalogued to permit rational fan selection and encourage fan development.

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CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

Quiet truck studies showed that the fan environment has a significant and perhaps dominant effect on fan noise generation. Thus, improving cooling system performance and consequently reducing the fan aerodynamic requirement is a key to quieter fans. Currently, cooling system components are specified primarily on the basis of minimum cost and not on the basis of maximum efficiency. If the relative values placed on cost control and heat transfer efficiency are changed, benefits can be derived not only from reduced noise, but also from reduced parasitic fuel consumption of the cooling system, an increasingly important factor in vehicle operating costs.

Quiet Vehicle Design

Radiator performance is primarily controlled by radiator face area, top tank temperature, and number of coolant passes. For a hypothetical radiator with constant face area and core thickness, increasing top tank temperature would permit an airflow requirement reduction which is estimated to allow a 11 dB reduction in fan noise. Use of a multipass radiator in this configuration would add a further 1.5 dB reduction, as estimated in a non-optimized case.

Fan aerodynamic performance is most significantly improved by reducing blade tip-to-shroud clearance. Test data provided by manufacturers tends to use tip clearances much smaller than obtainable in practice. Consequently, a test standard is required which would provide the vehicle designer a consistent data base. At the same time, a fan design which includes integral (rotating) shrouds will provide the best aerodynamic performance and least noise. Alternatively, low tip clearance fans using fixed engine-mounted shrouds or radiator-mounted fans (driven by a flexible coupling) will also provide superior aerodynamic and acoustical performance.

Quiet Fan Selection

Recent developments indicate that the understanding of fan noise generation mechanisms is now sufficient to develop analytical models which can be used to predict fan sound levels and optimize fan geometry for a given acoustic and aerodynamic performance. This understanding also underscores the need for an acoustical test procedure which will accurately describe fan noise generation characteristics (which are completely undocumented for current production fans). This information is urgently needed if the vehicle designer is to consider a quiet fan for his application.

RECOMMENDATIONS

From the previous conclusions, a number of actions are indicated which would result in quieter fans and vehicles.

General

Development of both aerodynamic and acoustical fan test procedures for vehicle applications should be fostered. These procedures would allow a common basis for the cataloging of the performance of commercially available fans and the comparison of new fan designs currently in research. With the availability of such measurement standards, a program of performance measurements should be undertaken to create a data base for the use of the vehicle designer.

A mathematical model describing fan noise generation which describes the major noise generation mechanism should be developed. This model should then be automated to permit the optimization of fan geometries for given vehicle applications.

Quiet Vehicle Demonstration Program

A quiet cooling system should be installed in a demonstration vehicle which utilizes a high top tank temperature. This can be accomplished by selecting a gasoline engine currently specified with high top tank temperature. The quieted vehicle should also utilize a multipass radiator of optimized configuration, radiator face area and core thickness. The fan for this vehicle should be an integral shroud fan or else be fitted with, preferably, a radiator-mounted fan with close tip clearance.

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I. INTRODUCTION

The engine cooling fan is a major component noise source in highway trucks as indicated in Table 1.1. This table, while specifically applicable to heavy trucks, is representative of medium trucks (or perhaps even understates the significance of the fan noise contribution). The data presented in Table 1.1 are based on work performed or reported during the early 1970s -- a watershed period in truck noise control. Since that time, however, the use of thermostatic speed-modulated fan drives has become commonplace, particularly in heavy trucks and increasingly in medium trucks. These fan drives were primarily installed to reduce the parasitic fuel consumption due to the fan but also have the benefit of reducing fan noise exposure. (The thermostatic speed modulation, while significantly reducing noise exposure -- the integration of sound level over time -- did not reduce the actual magnitude of the fan sound levels.) The use of thermostatic fan drives was further encouraged by the allowance in the EPA Medium and Heavy Truck New Product Noise Regulation for testing of the vehicles with fans inoperative when these devices are installed. Consequently, by the latter part of the 1970s, work on developing and implementing quieter truck fans appeared to come to a halt.

A number of lessons were learned from this earlier work. Perhaps the most salient realization was the relative insensitivity of noise generation to fan design. A number of these investigators found, when testing a variety of fan configurations in the test vehicles, no significant difference with respect to the original production fan was observed.[2,3,4] The fans evaluated included aerodynamically sophisticated designs with many blades and twisted, airfoil-section blade geometries. However, in one vehicle investigators found that all replacement fans tested in their vehicle performed between 4 and 10 dBA better than the original production installation.[5] This behavior may be explained by very poor fan installation in the production vehicle. Thus, when the production fan is reasonably well selected, relatively little additional noise control benefit can be obtained by fan substitution -- even when the production fan is a rudimentary stamped-steel configuration.

TABLE 1.1
HEAVY TRUCK COMPONENT NOISE SOURCES[1]
(Measured at 50 Ft. Per SAE J366a Acceleration Test)

VEHICLE TYPE	TOTAL VEHICLE LEVEL (dBA)	COMPONENT LEVEL (dBA)						
		BARE ENGINE	EXHAUST	FAN	INTAKE	TRANS-MISSION	TIRES	OTHER
DIESEL-ENGINED	87	82	82	81	74	75	68	75
GASOLINE-ENGINED	86	76	81	80	-	-	-	-

Investigators also observed that laboratory-measured results significantly overstated benefits of noise control modifications when installed in an actual vehicle.[2] This further underscores the significance of the fan environment on noise generation. This and the experience with fan substitution emphasize the need for reducing fan aerodynamic performance requirements as a means of quieting cooling systems. The use of a larger face area radiator is an obvious means of increasing cooling system performance. In one study the radiator face area was increased 67% and fan size increased 20% resulting in a halving of the fan speed requirement.[6] This configuration resulted in an approximately 15 dBA reduction in fan noise levels (although air-to-boil performance of the vehicle was approximately 5°F deficient).

Several investigators found that reducing fan tip-to-shroud clearance has a substantial benefit in improving fan performance and reducing fan noise.[2,5] For engine-mounted fans and radiator-mounted shrouds, tip clearance reductions down to approximately 0.75 in. were possible (from typical current practice of approximately 1.5 in.). Because of engine deflection on its mounts, greater tip clearance reduction required engine-mounted shrouds. This approach was implemented using a flexible shroud attached to the radiator.[2] Use of the engine-mounted shroud resulted in some installation difficulties and required that an idler pulley be utilized for fan belt adjustment.

These investigators had several other interesting findings:

- Improved sealing between the shroud and the radiator reduced the air leakage behind the radiator and improved the heat transfer performance of the radiator, permitting a 3-4 dBA noise level reduction.

- Modification of the fan-to-radiator and fan-to-engine distances resulted in a noise level reduction of 4 dBA -- a benefit which is greater than that predicted in laboratory evaluations, again demonstrating the effect of the engine compartment environment.

- Even with the improved airflow environment, including the usage of tight tip clearance and a venturi shroud, the "conventional truck fan is still as good or better as any known design available."

- Optimization of the radiator core geometry by alterations of number of tube rows and fin configuration resulted in approximately a 5 dBA reduction.

The purpose of this report is to review the current state of the art for quieting of truck cooling system fans. Recent improvements in analytical capabilities and increased understanding of fan noise are reviewed. Practical considerations drawn from these insights are noted. In light of the experience described above, considerable emphasis has been devoted to the investigation of the cooling system airflow requirement and an evaluation of alternative cooling system airflow configurations which would yield significant cooling system sound level reductions. Recommendations for actions which would foster the design, development, and installation of quieter fans are made.

II. FAN AIRFLOW REQUIREMENT

For internal combustion-engined vehicles, only a fraction of the chemical energy of the fuel is transformed into useful work. The remainder of the fuel energy is dissipated by various mechanisms. The relative proportions of these fuel energy paths are defined in Table 2.1 for vehicles operating at maximum load. The fraction of chemical energy rejected to the cooling system is roughly constant regardless of the engine operating condition (although due to throttling losses, gasoline engines reject relatively more waste heat at low load conditions). Therefore, the greatest heat rejection requirement placed on the cooling system will be at the high load, high fuel consumption conditions. These conditions are at the maximum rated or governed speed of the engine.

TABLE 2.1

ENERGY BALANCE IN MOTOR VEHICLES[7]
(Percent of Fuel Energy at Maximum Load)

	Gasoline-Engined Vehicles	Diesel-Engined Vehicles	
		Trucks	Automobiles
Coolant	20	30	27
Exhaust Gases	35	30	37
Radiation*	20	10	13
Useful Work	25	30	23

*Includes radiation from exhaust pipes, coolant jacket and pipes, and engine walls without coolant circulation, e.g., crankcase.

Vehicle operation at low speed in the lowest gear presents the greatest demand on the cooling system, especially the fan, since waste heat production is high but little ram air benefit is present. Consequently, low vehicle speed operation at maximum engine speed (typically at vehicle speeds of 5 to 15 mph) is the cooling system design condition.

Cooling system performance is generally evaluated by means of a parameter known as air-to-boil (ATB), theoretically the inlet air temperature at which the coolant boils. ATB is tested particularly during low speed, high load vehicle operation. For automobiles several design conditions are generally checked to assure the cooling system will perform adequately under all circumstances. These conditions are maximum speed, towing on a grade, and idle. For medium and heavy trucks, low speed in the minimum gear is the single condition that is generally sufficient to test the adequacy of the cooling system.

ATB can be determined at any ambient temperature during a cooling system test, as

$$ATB = T_B - T_{TT} + T_A$$

where, T_B = coolant boiling point
T_{TT} = top tank temperature, measured in test
T_A = ambient temperature, measured in test.

Most diesel engine manufacturers specify maximum top tank temperature, minimum air-to-boil, and boiling point temperature. Thus a design ambient temperature is implied. Detroit Diesel and Caterpillar limit top tank temperature to 210°F while Cummins limits top tank temperature to 203°F.[8,9,10] The design conditions for gasoline engines are generally considered proprietary, since the engine and vehicle manufacturers are the same. Design ambient temperatures are typically about 110°F.

COOLING SYSTEM DESIGN PROCESS

The cooling system design process consists of essentially four steps: the determination of total heat rejection requirement, the selection of a radiator configuration, the calculation of airflow requirement, and the selection of a suitable fan. Typically these steps are performed in the order as stated here:

- The heat rejection requirement is a sum of those heat loads resulting from the engine, automatic transmission, air conditioning condenser, and engine retarder--as applicable to the specific vehicle.

- The radiator configuration is selected based upon the physical installation constraints and cost, and the magnitude of the heat rejection requirement.

- The volumetric airflow requirement is determined from the radiator heat rejection capability and the heat rejection rate dictated by the installation and the cooling system design requirements. The required pressure rise is determined by the pressure drop incurred by the radiator and the pressure drops incurred through the rest of the vehicle.

- The fan is selected based upon the airflow operating point, physical installation and cost constraints, and noise performance.

A slightly different cooling system design procedure is used by Ford.[11] In this procedure the fan configuration is initially selected based upon an a priori noise performance goal. The radiator configuration is then selected. The airflow, for which the aerodynamic balance is achieved, is calculated. Finally, the heat rejection capability of the system is evaluated with respect to cooling system criteria. This procedure is illustrated, in simplified form, in Figure 2.1. This approach appears to be attractive from the noise control standpoint since it gives first priority to noise performance. However, it can only be truly effective if the chosen fan noise design level challenges the cooling system designer.

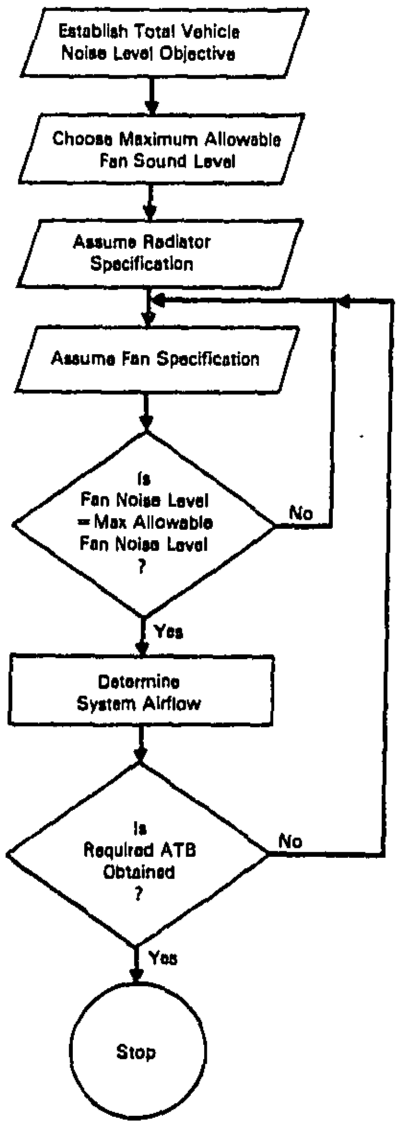

FIGURE 2.1. COOLING SYSTEM DESIGN PROCEDURE

Specifications for the cooling system are defined from various sources:

- The engine manufacturer specifies the heat rejection rate of the engine, the required cooling performance (ATB), the maximum allowable top tank temperature, the coolant flow rate provided by the engine, and --possibly-- the coolant type.

- The radiator manufacturer provides heat transfer performance curves and airflow resistance curves for the radiator cores.

- The fan manufacturer provides aerodynamic performance curves for the fan.

- The vehicle manufacturer defines the vehicle configuration (cab design, transmission type, and so forth), the installation and cost constraints, the vehicle airflow resistance, and the design ambient temperature desired.

RADIATOR DESIGN AND COOLING SYSTEM PERFORMANCE

The radiators of current highway vehicles have generally been optimized on the basis of minimum weight and cost. Face area has increased to better utilize ram air for cooling and to gain fuel economy by reducing fan use in normal operations. It is possible, however, to optimize a radiator on the basis of the minimum required air power.* The parameters which affect the cooling system air power requirement, which are discussed in the following paragraphs, are:

- Thickness
- Face area
- Fin density

*Air power is the ideal horsepower required to move a certain weight of air through a pressure rise. The air power dictates the fan performance requirement which in turn relates directly to noise generation. See Appendix A, Nomenclature, for definition of air power.

- Fin design
- Coolant flow rate
- Radiator loading
- System resistance
- Maximum ambient temperature
- Maximum top tank temperature
- Coolant type
- Heat exchanger type.

As part of a cooling system study for a military combat/tactical vehicle, a parametric study was conducted of a typical radiator used in an automotive cooling system.[12] The vehicle analyzed differed from highway truck design practice in that the vehicle pressure drop (less radiator core) was quite large, 5.79 in. H_2O with 15,000 ft^3/min. airflow, and the radiator air inlet temperature was relatively high, 120°F. (Complete description of the cooling system of this vehicle is provided in Table 2.2.) These differences make practical thicker radiator cores than normally suitable in highway vehicles. However, the <u>trends</u> defined in this study are applicable to highway vehicles.

TABLE 2.2

DESCRIPTION OF COOLING SYSTEM FOR MILITARY VEHICLE[12]

Heat Transfer Rate: 9,000 BTU/min
Core Face Area: 6.19 ft^2
Coolant Type: 50/50 aqueous ethylene glycol mixture
Coolant Inlet Temperature to Radiator: 200°F
Coolant Temperature Drop Through Radiator: 10°F
Air Inlet Temperature to Radiator: 120°F
Fin Density: 11 fins/in.
Core Type: tube and fin, with wavy fins
Vehicles (less radiator) Pressure Drop: 5.79 in. H_2O
with 15,000 ft^3/min. airflow

This study will be cited liberally in the following paragraphs. Except where otherwise noted, figures reproduced from this study pertain to the configuration specified by Table 2.2.

The military vehicles as stated above have highly restricted cooling airflow paths. The author describes a representative vehicle in which the pressure drops through the system are as in Table 2.3.

TABLE 2.3
COOLING AIRFLOW PRESSURE DROPS THROUGH A MILITARY VEHICLE[12]
(with 18,000 ft^3/min. airflow)

Flow Element	Intake Grill	Radiator	Engine Compartment	Exhaust Grill	Total
Pressure Drop, Δp (in. H_2O)	2.1	2.0	3.7	2.2	10.0

In this analysis of airflow resistance effects on cooling performance, the author adopts the practice

$$\Delta p_T = \Delta p_I + \Delta p_R + r\Delta p_C + \Delta p_E$$

where the subscripts denote,

T = Total
I = intake grill
R = radiator
C = engine compartment
E = exhaust grill

and the coefficient, r, is defined as the engine compartment air resistance index. The engine compartment air resistance index is taken as 1 in the following figures from this work except as specifically noted otherwise.

Thickness. Modern highway vehicle systems can have an overall vehicle pressure drop at the design flow one to four times the radiator core pressure drop. Military vehicles have an overall vehicle pressure drop approximately five times the radiator core pressure drop. Figure 2.2 illustrates the interaction of system pressure drop, core pressure drop, air velocity, and fan power with core thickness for the previously described military vehicle for a constant rate of heat transfer.[12] Compared to highway truck practice, this vehicle has very restricted air flow, slightly high air inlet temperature, and a relatively small radiator face area. However, the radiator heat transfer for this vehicle is about 1500 BTU/min/ft^2, not particularly different from most highway vehicles. Note that for constant heat transfer rate airflows, the minimum values for fan (air) power, air velocity system (total) air pressure drop, and radiator core air pressure drop do not occur until core thickness of about 11 in., 9 in., and 6.5 in., respectively.

Face Area. The required fan air power for given heat transfer rate will always be less for a larger face area at a given core thickness. This is illustrated in Figure 2.3. A radiator weight reduction (i.e., smaller core volume) at constant power can sometimes be achieved by using a smaller face area and greater thickness. This is illustrated by Table 2.4. The smaller core volume in turn generally translates to lower total cost.

TABLE 2.4

EFFECT OF CORE FACE AREA ON REQUIRED RADIATOR VOLUME

Face Area (Ft^2)	Thickness (in.)	Core Volume (ft^3)	Air Power HP
8.28	3.2	2.21	20
6.19	4.0	2.06	20

Fin Density. Increases in fin density increase the heat transfer area for a given core weight but also increase the pressure drop. A core having a higher fin density with fins of the same design would reach its

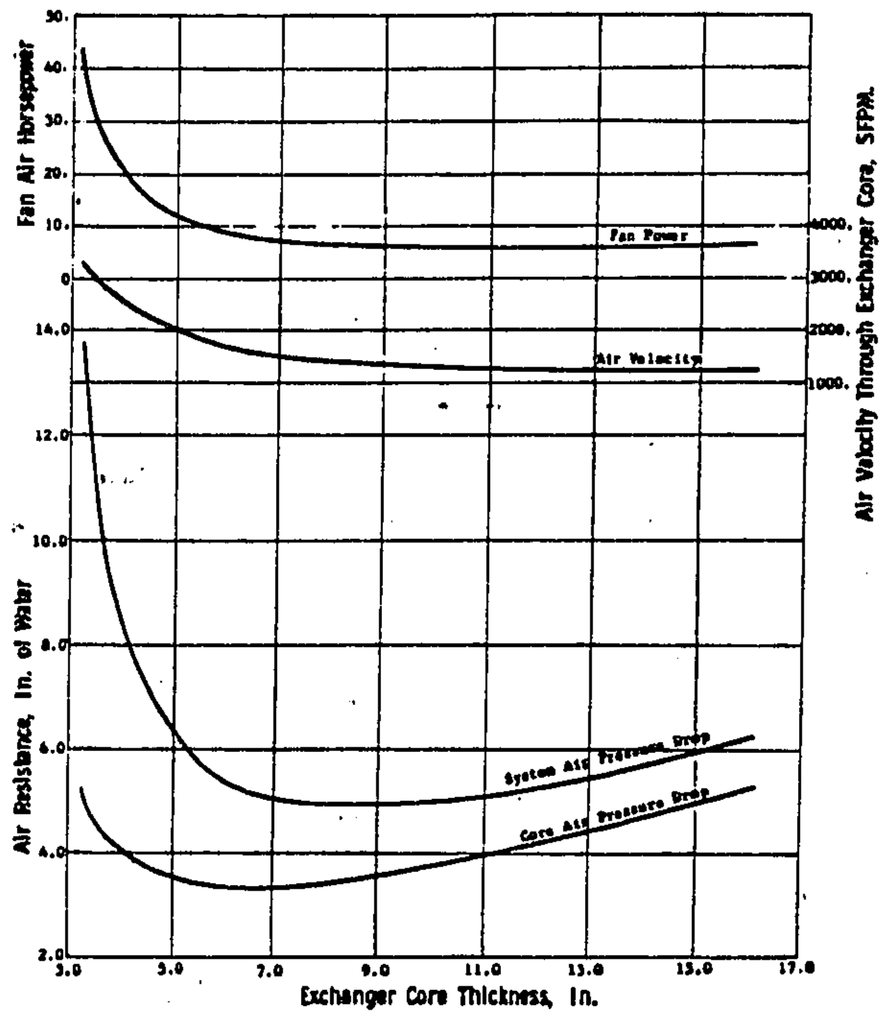

FIGURE 2.2. EFFECT OF RADIATOR CORE THICKNESS[12]

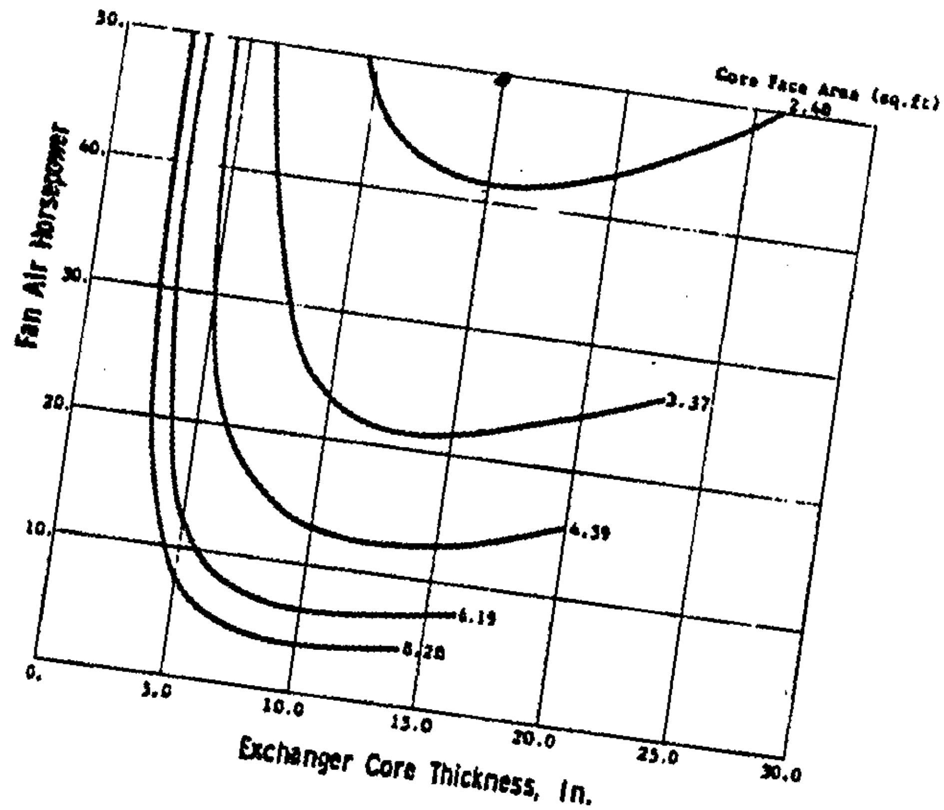

FIGURE 2.3. EFFECT OF RADIATOR FACE AREA[12]

optimum thickness at a lower thickness than a core with a lower fin density. Increasing fin density increases the overall performance of a radiator because the air side heat transfer effectiveness is normally the limiting factor on radiator performance. Thus, increasing fin density matches the air-side heat transfer capacity more closely with that of the water-side. To reduce the cost of radiators and increase performance for a given weight, greater fin densities for radiators have been used. Automobile radiators are now being produced with densities as high as 24 fins/in. The optimum fin density will depend on the fin thickness required for durability in the installation, and the relative changes in heat transfer effectiveness and pressure drop with thickness. For each core geometry under consideration, a core thickness for minimum air power in the available face area can be determined. A complete parametric study that would include not only variation in core thickness but variation in fin density as well would be required to determine the combined optimum of both parameters.

Fin Design. Convective heat transfer from the radiator core metal to the air is impaired by the formation of the boundary layer in the airflow. Thus, texturing of the fins, which generates turbulence in the airflow, increases the heat transfer capacity of the radiator. These fin texturing devices include dimples, louvers, and waves. In a wind tunnel investigation of the effects of installation parameters on truck cooling system performance, a radiator with 11 fins/in. fin density and plate (flat) fins had the same heat transfer capabilities at equal flow and temperatures as a second radiator with only 9 fins/in. and louvered fins. Thus, a louvered fin radiator with the same fin density has a higher heat transfer capability. However, for the same heat transfer capability, the plate fin type radiator has the least pressure drop. When a radiator is optimized on the basis of minimal weight, a louvered fin radiator will be the obvious choice, while the most efficient radiator on the basis of minimum air power will be a plate fin radiator. Generally, louvered fin radiators have been chosen for highway vehicles where plugging is not a problem while plate-fin radiators are used for off-highway applications.

Coolant Flow Rate. As discussed above, automotive radiator performance is generally limited by the air-side heat transfer capability of the radiator. Consequently, increasing coolant flow rate will provide significant benefits only for those systems which are coolant-side limited or have been altered already to improve air-side heat transfer. Figure 2.4 is a typical radiator manufacturer's heat transfer correction curve. In this figure, the design point is indicated as 2 gal/min/in. core width and all of the radiator core performance curves are constructed using this coolant flow. Note that doubling of flow rate results in a 5% heat transfer increase, while a halving of flow results in a 6.5% decrease. Further flow reductions will result in significantly greater heat transfer decreases.

<u>Radiator Loading</u>. Radiator loading is the heat rejection requirement placed upon the radiator per unit of radiator core volume. Table 2.5 compares radiator loadings for a military vehicle and comparable commercial vehicles. Several comparisons are of interest in Table 2.5. For the military vehicle, increasing the required heat rejection rate with constant face area resulted in greater radiator loadings and dramatically increased air power requirements-- even when minimum air power radiator cores are used. Note that, although these systems have quite low radiator loadings, their air power requirement is relatively high due to their high system airflow resistance. For the heavy truck, increasing face area resulted in lower radiator loading and significantly lower required air power. In general, radiator loading should be minimized for the minimum air power requirement in a given installation although it is not suitable for comparisons between different installation. A range of required radiator loading for good design practice has been recommended as 1.6 to 2.2 BTU/min/in.[3]

<u>System Resistance</u>. The effect of increases in the airflow path resistance is indicated in Figure 2.5. Recall that the power plant compartment air resistance index, r, is a multiplier placed upon the engine compartment and that inlet, outlet, and radiator resistances are held constant. (However, for the condition of r=0, the resistance of the inlet and outlet is also equated to zero. Thus, this curve is the minimum fan power for the radiator core operating with zero additional pressure drop either in the inlet or outlet paths, an unattainable minimum.) The range of system resistance depicted in Figure 2.5 is extreme in the context of highway

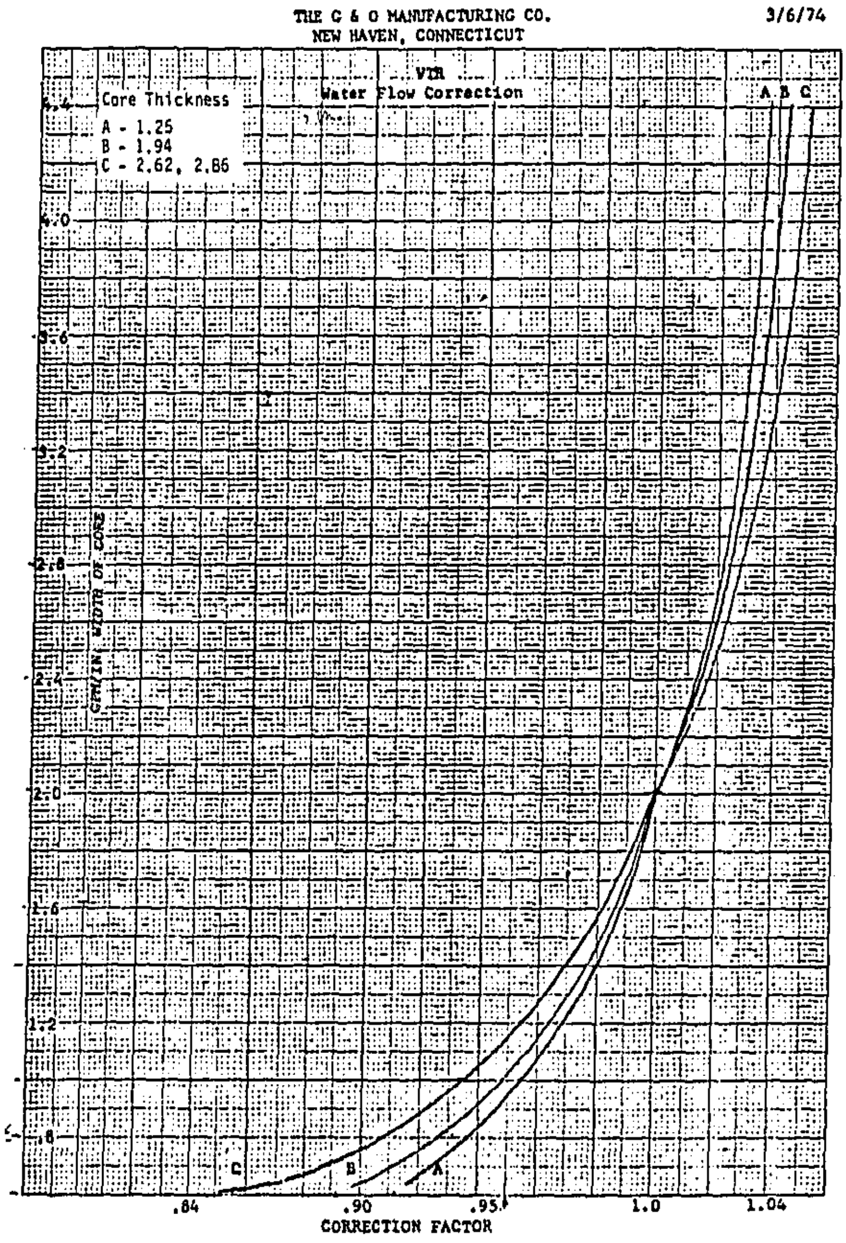

FIGURE 2.4. EFFECT OF COOLANT FLOW RATE
(FOR GM-TYPE, SIDE FLOW LOUVERED-FIN RADIATOR)

TABLE 2.5

EFFECT OF RADIATOR LOADINGS

VEHICLE APPLICATION	REF.	CORE FACE AREA (in.^{2})	CORE THICKNESS (in.)	HEAT REJECTION RATE (BTU/min)	RADIATOR LOADING (BTU/min-in^3)	AIR POWER (HP)
Military Vehicle[a]	12	891	10.3[f]	7,200	0.784	3.0
		891	11.5[f]	9,000	0.878	6.3
		891	12.5[f]	11,250	1.01	13.3
Ford heavy truck[a]	11	1200	3.29[b]	9,600	2.43	4.1
		1700	3.29[b]	9,600	1.72	2.3
GMC medium truck[c]	13	694	2.63	6,160[d]	3.37	3.5
		694	2.63	6,602[e]	3.61	-
Good practice range	14	--	---	--	1.6-2.2	-

NOTES:

[a]diesel-engined

[b]estimated

[c]gasoline-engined

[d]without air conditioning

[e]with air conditioning load added to radiator

[f]for minimum air power, thickness

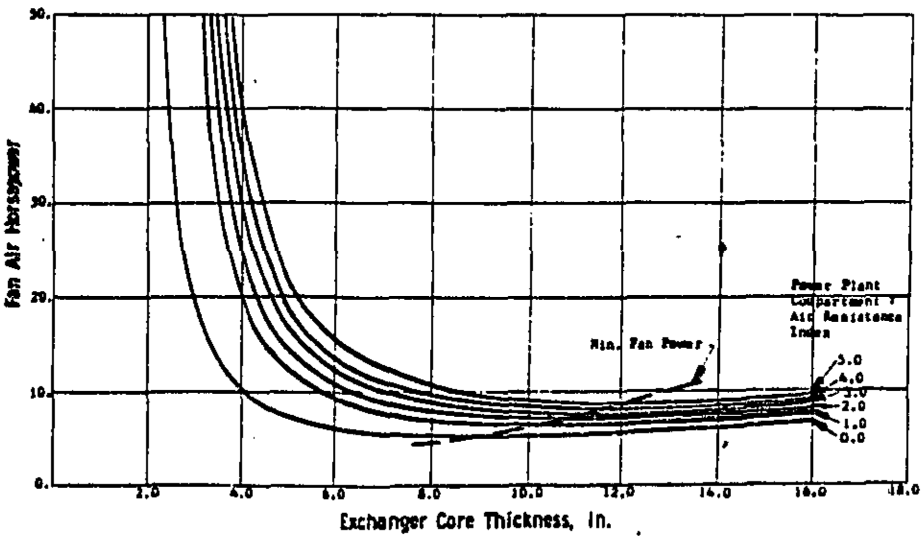

FIGURE 2.5. EFFECT OF VEHICLE AIRFLOW RESISTANCE[12]

vehicles for which the total vehicle resistance represented by r=1 is not likely to be exceeded. However, the trend illustrated is significant -- more restricted systems require thicker radiator cores for constant air power.

Maximum Ambient Temperature. Figure 2.6 illustrates the effect of the design ambient temperature upon the optimum core thickness for minimum fan power. The 120°F ambient used as the design condition is reasonably representative of current highway trucks since a margin is allowed for air conditioning. Note that a 30°F drop in ambient only drops optimum core thickness slightly. A 30°F rise requires more than three times the fan power and a much thicker radiator core.

Maximum Top Tank Temperature. The heat rejection capability of a radiator is a strong function of radiator inlet air temperature, i.e., top tank temperature. Figure 2.7 illustrates the effect of top tank temperature on the core thickness for minimum fan power and thus also on the overall size and weight of the cooling system. The 200°F maximum coolant temperature is conservative by most current truck practices. Although Cummins requires 203°F and Detroit Diesel and Caterpillar, 210°F, Chrysler uses 245°F for automobile gasoline engines.[15] Higher top tank temperature (maximum coolant temperature) would reduce both fan power and optimum thickness substantially. In Figure 2.7 a 20°F rise in the allowable top tank temperature would cause the optimum core thickness to occur at approximately 8 in. instead of 11 in. and a halving of the required fan power.

Coolant Type. The effect of changes in the engine coolant on the size and required fan power for an optimum heat exchanger is illustrated in Figure 2.8. A coolant which has a higher specific heat is able to extract the same heat from the radiator core at a lower flow rate or more heat at the same flow rate. The figure illustrates the conditions for two coolants, water and aqueous ethylene glycol. The water, having the higher specific heat, results in lower fan power. This effect is particularly pronounced for thin radiator cores where the use of glycol results in a significant air power penalty. (Note also that a cooling system performance evaluated with water coolant may significantly overstate cooling performance using glycol.) Nevertheless, the use of water in engine cooling systems is no longer recommended by most engine

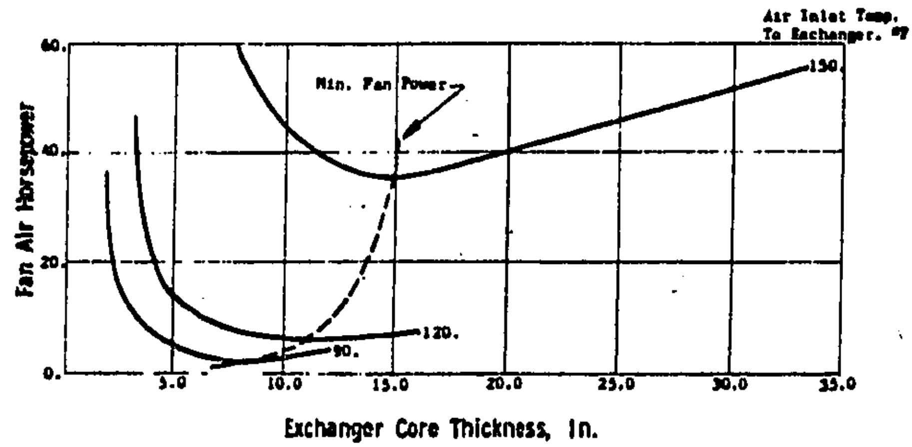

FIGURE 2.6. EFFECT OF AMBIENT TEMPERATURE[12]

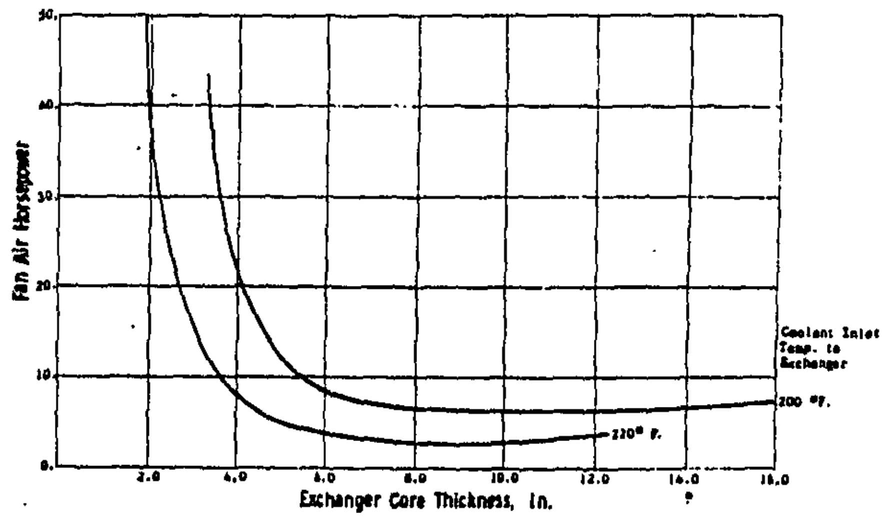

FIGURE 2.7. EFFECT OF TOP TANK TEMPERATURE[12]

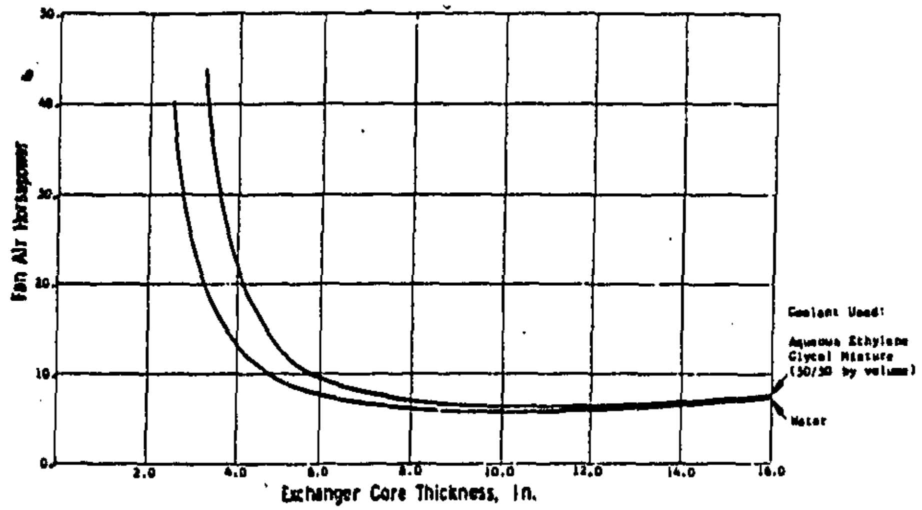

FIGURE 2.8. EFFECT OF COOLANT TYPE[12]

manufacturers because of the need for corrosion inhibitors which are present in most ethylene glycol cooling system conditioners. Using water, the immediate heat transfer capabilities of the cooling system would be improved slightly, but core scaling would be more severe than with a glycol mixture and ultimately the heat transfer capabilities would be less. Furthermore, glycol coolants are capable of operating at higher temperatures, thus further improving the heat transfer as discussed previously.

Heat Exchanger Type. The heat transfer at any specific point in a heat exchanger is directly proportional to the temperature difference between the hot and cold fluids (e.g., for the automotive radiator, the coolant and air). Consequently, the most efficient heat exchanger is one in which this difference is maximized at all points in the exchanger core. This maximum difference would be achieved when the air and coolant flow in parallel but opposite directions such that the entering coolant "sees" the exiting air and vice versa. This heat exchanger configuration is known as counterflow. In automotive radiators the cooling air flows perpendicular to the path of the coolant. This configuration is known as crossflow. Since the temperature differential is not maximized throughout the radiator, its heat transfer is only some fraction of that possible with a counterflow heat exchanger. Counterflow heat exchanger performance can be approached with a crossflow radiator by directing the coolant flow through multiple passes through the exchanger core such that the differential between the average coolant temperature for each pass and the air temperature for that pass is maximized. This configuration is known as a multipass crossflow heat exchanger.

The automotive radiator is a single pass crossflow heat exchanger. Its effectiveness, ϵ, (as defined in Appendix A), is approximately 50%.* Alteration of cooling system operating parameters --notably increasing maximum top tank temperature-- can increase the effectiveness to about 70% (for a top tank temperature of about 250°F).

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*As calculated for the GMC gasoline-engined medium truck described in Appendix C.

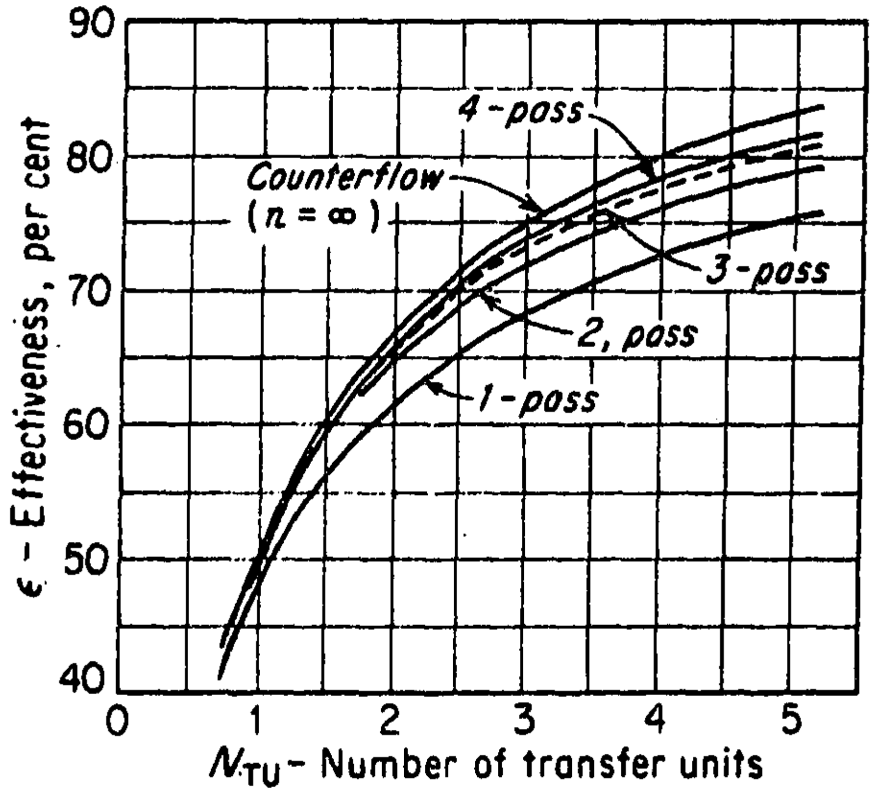

FIGURE 2.9. EFFECT OF HEAT EXCHANGER TYPE[16]

The potential benefit multiple passes in an automotive radiator are illustrated in Figure 2.9.[16] At current cooling system operating conditions, use of multiple coolant passes will result in heat transfer improvements of about 2-3%. However, for radiators operating at higher effectivenesses --such as those with high top tank temperatures-- heat transfer improvements of about 10% can be obtained depending upon the number of coolant passes possible.

FAN SELECTION PROCESS

Fans for current highway trucks are generally selected on the basis of cost, noise, and power consumption (i.e., parasitic fuel consumption). Since thermostatic speed modulation is typically used to minimize noise generation and power consumption for high performance fan installations, cost is the predominant selection consideration. (A result of these fan speed control devices, consequently, has been to retard the development and implementation of inherently quieter, more efficient fan designs.)

Currently no industry-wide practice exists for the aerodynamic performance testing of fans for automotive applications. Fan manufacturers provide fan performance data based upon proprietary test practices which vary and are idealized. (These procedures use fan-to-shroud clearances that are much less than obtained in practice and the engine compartment environment is not simulated. Thus, aerodynamic performance relative to actual installations is overestimated.) A Society of Automotive Engineers (SAE) test procedure for the measurement of power consumption of automotive fans is being drafted (SAE J1339). This procedure does specify test installation parameters but still accepts somewhat optimistic test conditions.

Fan performance data provided by manufacturers normally consists of pressure rise and power input as a function of volumetric flow rate for a given fan size and operating speed; examples of these data are provided in Figures 2.10. The specification of a fan for which the available performance curve (pressure rise vs. airflow) does not intersect the desired operating condition is determined by use of the "Fan Laws." The Fan Laws relate pressure rise and airflow to fan speed and size for geometrically similar fans at the same point of rating on the performance curve.

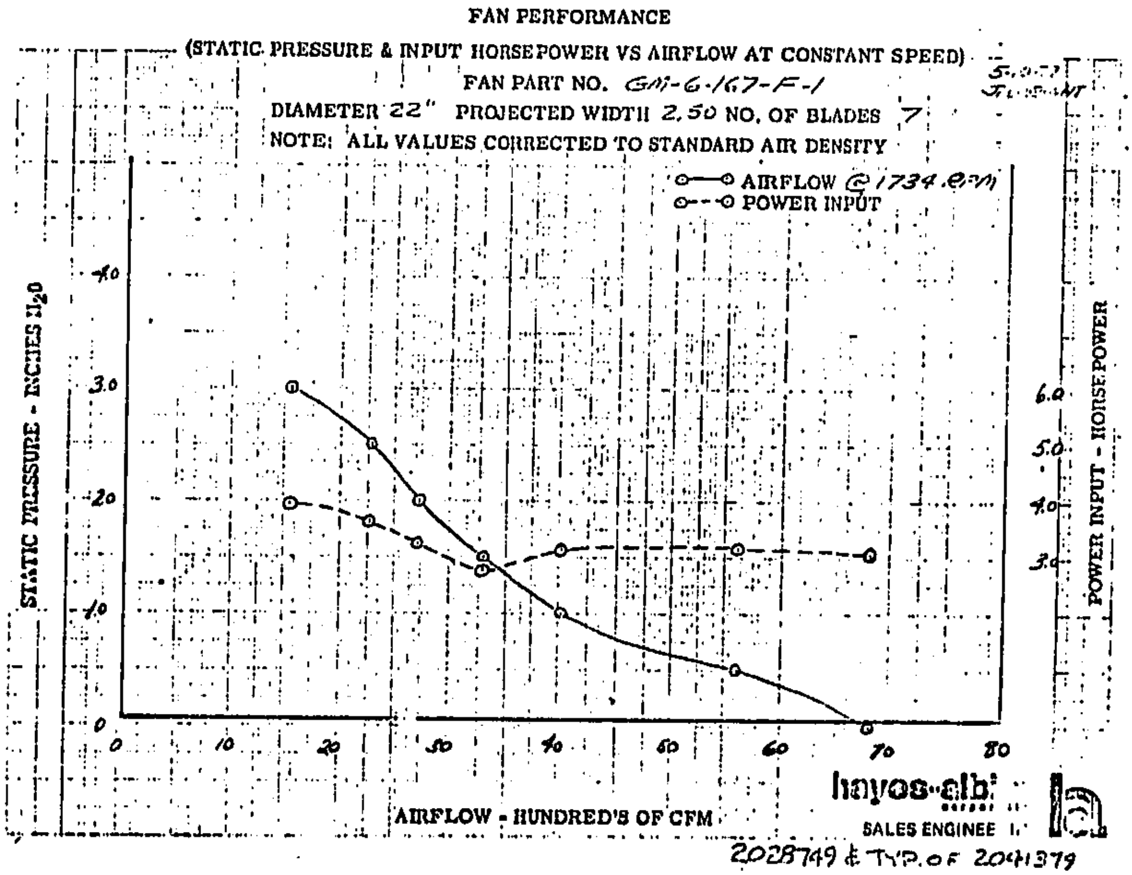

FIGURE 2.10a. TYPICAL MANUFACTURER-SUPPLIED FAN PERFORMANCE CURVES

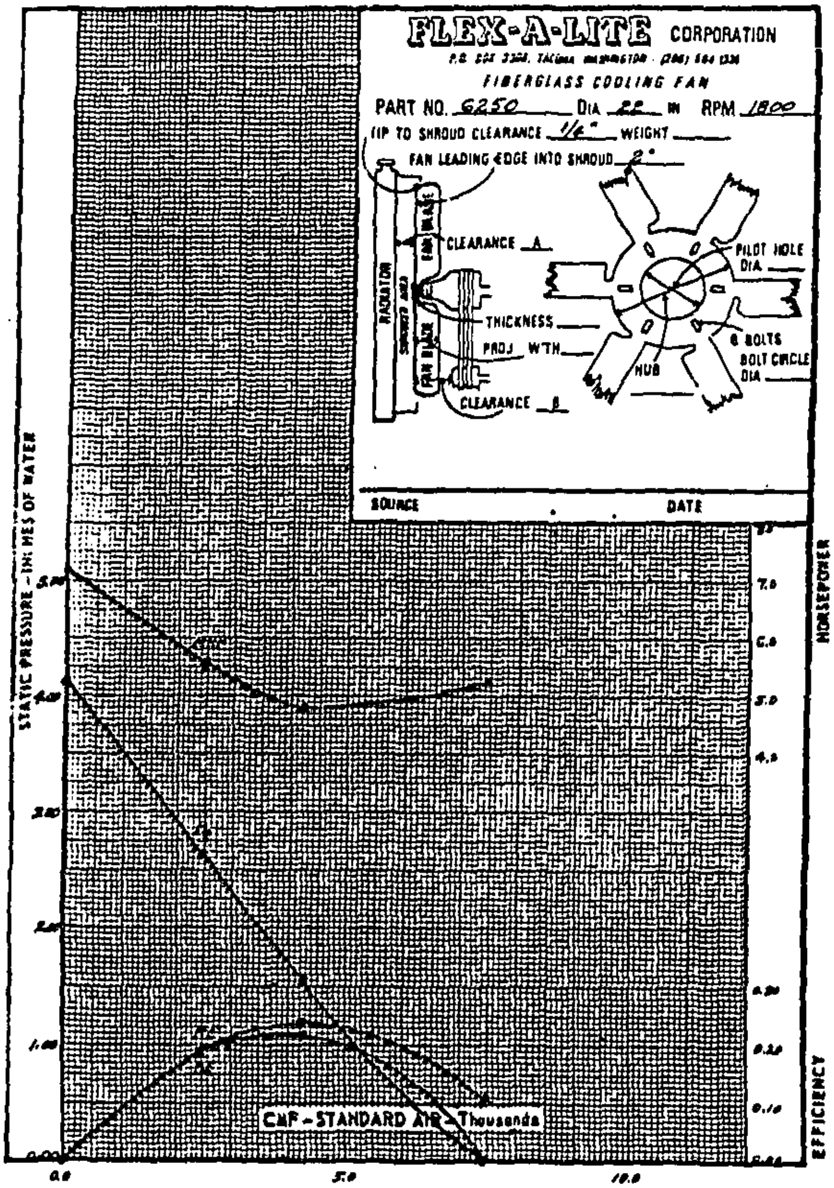

FIGURE 2.10b. TYPICAL MANUFACTURER-SUPPLIED FAN PERFORMANCE CURVES

Non-dimensionalyzing fan performance parameters provides a more convenient means of fan selection. Non-dimensional variables --flow coefficient, ϕ, and presssure coefficient, ψ ,-- specify the dimensionless fan operating point. They are calculated from the parameters of flow, pressure, speed, and diameter. Any fan whose dimensionless performance curve passes through the operating point will meet the application requirements. Figure 2.11 illustrates the matching of a fan to a particular system using dimensionless fan performance characteristics.[17] Three fans are illustrated on the dimensionless pressure coefficient versus flow coefficient plane, identified as A, B, and C. A is a low capacity fan, B a medium capacity fan, and C a high capacity fan. Intersections of system characteristics and fan characteristics represent fan operating points. If diameter is limited, as it is in most trucks, the slowest turning fan will be the one with its characteristics furthest to the right of the graph. For example, at 36 in. diameter, fan C operates at 1370 rpm while fan A operates at 2150 rpm. For noise control, the higher capacity fan appears to be a likely choice since noise is a strong function of fan speed. However, off-design operation into stall, noise performance data, and efficiency would have to be examined before a final choice could be made.

Caution should be exercised in using a non-dimensional flow coefficient since its definition has not became standardized in the literature. Examples of alternative definitions are provided in Table 2.6. In this report the convention of Baranski is used.

TABLE 2.6
DIMENSIONLESS COEFFICIENTS

Reference	Baranski[17]	Mellin[18]	Longhouse[19]
Flow Coefficient,	$\frac{Q}{AU_T}$	$\frac{Q}{d^2U_T}$	$\frac{Q}{A_bU_T}$

A = fan disc area = $\pi d^2/4$

d = fan diameter

A_b = annular area swept by fan blades
= $\pi(d^2-d_r^2)/4$

d_r = fan blade root diameter

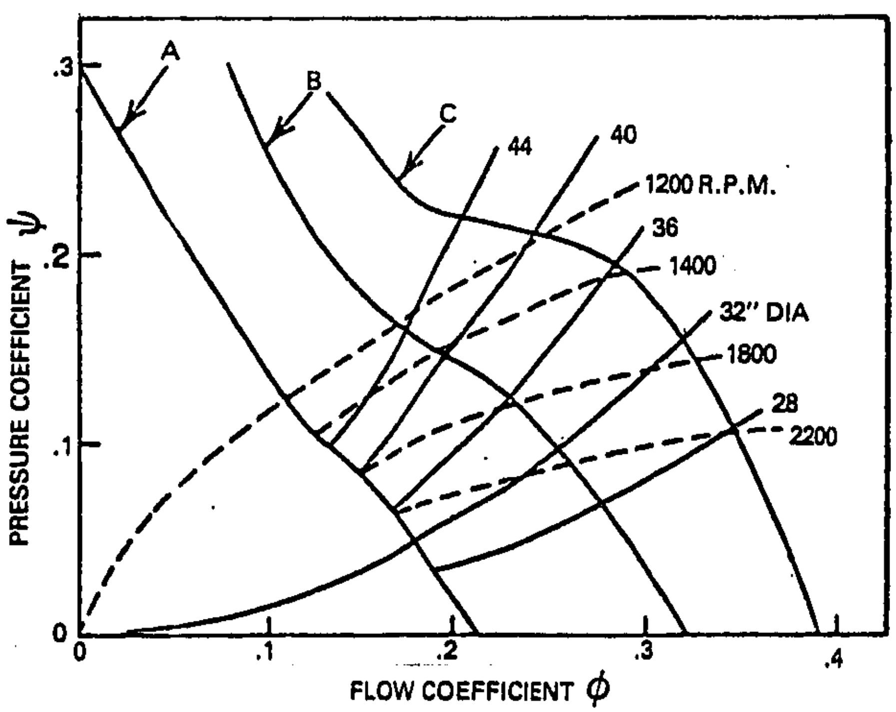

FIGURE 2.11. NON-DIMENSIONAL FAN PERFORMANCE CURVES

A further extension of the use of non-dimensional variables involves specific diameter. The specific diameter of a fan is a function of the flow coefficient for a given fan design. It represents the relative fan size for a given fan performance, and is defined as

$$D_s = \frac{\psi^{1/4}}{\phi^{1/2}}$$

Specific diameter is particularly useful for design of automotive systems since their performance is typically fan size constrained. In automotive applications specific diameter ranges from 1.2 to 2.4 where the lower limit describes unrestricted systems with ram air benefit and no air conditioning condenser, and the upper limit describes very restricted systems or these operating without ram air, such as at idle.[18] A fan selection procedure utilizing specific diameter is described in Figure 2.12.

FAN DESIGN AND COOLING SYSTEM PERFORMANCE

The effects of installation parameters on fan performance are displayed in Figure 2.13. The primary parameters which affect fan performance are:

- Projection of the fan into the shroud
- Fan-to-radiator distance
- Tip clearance
- Shroud type
- Fan-to-engine distance.

These parameters have been listed in the order of the importance of their effect as determined in tests.[20]

Projection of the Fan into the Shroud. Since engine-cooling fans are generally operating in a mixed flow mode rather than an axial flow condition,*

*At low pressure rise conditions the airflow from automotive fans exits axially. As the pressure rise required of the fan increases, an increasingly larger radial component of flow velocity is produced. (The radial flow is essentially a pumping loss of the fan.) The resultant axial-radial flow is known as mixed flow.

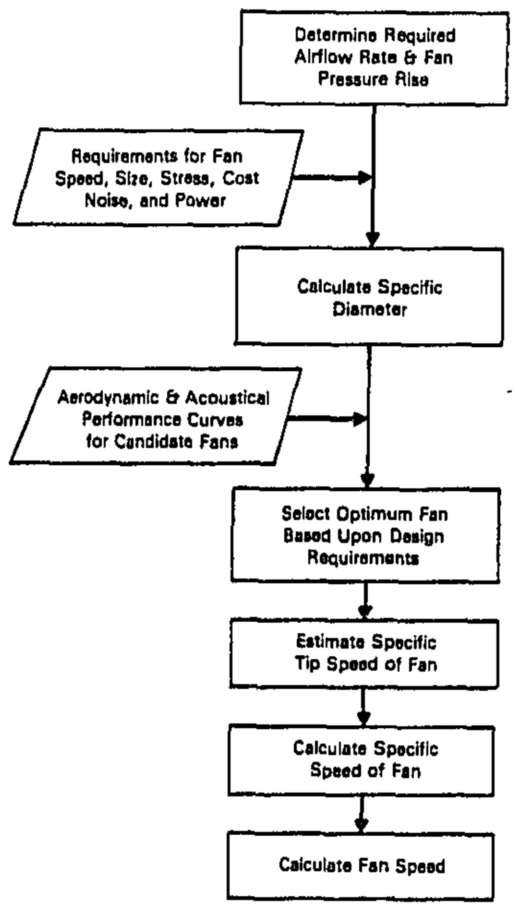

FIGURE 2.12. FAN SELECTION PROCEDURE

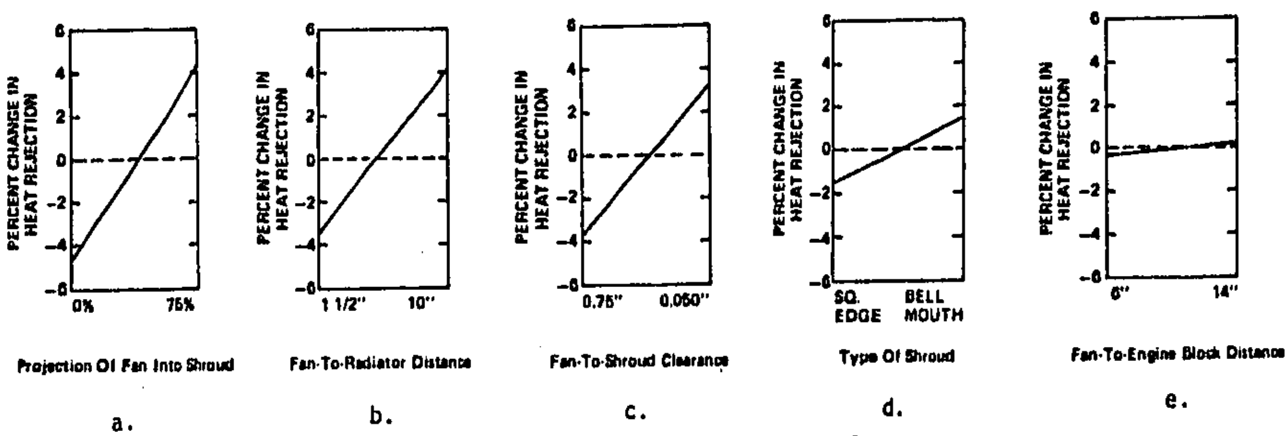

FIGURE 2.13. EFFECT OF FAN INSTALLATION[20]

the portion of the projected width of the fan outside of the shroud affects fan pumping capability. The use of a fan shroud restricts radial airflow. The effectiveness of the shroud in accomplishing this is a function of the extent of the fan insertion into the shroud. As shown in Figure 2.13a, heat rejection improved about 9% for a fan with 75% shroud coverage versus one with 0% coverage. Recommended fan coverage is 50 to 67%.[10] Very high resistance systems work better with 50% coverage while very low resistance systems work better at 67% coverage.[21]

Fan-to-Radiator-Distance. The proximity of the radiator to the fan affects the turbulence at the fan inlet. Cummins recommends that the fan-to-radiator distance be 2 to 4 in. for the best performance. An increase in heat rejection of 8% was observed as shown in Figure 2.13b for increasing fan-to-radiator distance from 1.5 to 10 in. (.05 to .3 diameters). However, in most vehicles the available spacing is such that 10 in. spacing is not possible. If it were available, the additional space might better be utilized for additional core thickness or straightening and fairing of the flow path.

Tip Clearance. Fan shroud effectiveness is also a strong function of fan blade tip-to-shroud clearance. In Figure 2.13c a 0.05 in. (0.3 diameter) tip clearance shroud had about a 7% performance improvement with respect to a 0.75 in. (0.05 diameter) tip clearance shroud. Mellin found peak fan efficiencies increased from 18% to 34% for fan tip clearance reduction from 3.6% to 0.5% of the tip diameter.[18] Further examples of the benefits of reduced tip clearance are found on pages, B-4 and B-6 of Appendix B.[21] The benefits of reduced tip clearance will be greater for aerodynamically sophisticated fans (e.g., twisted, airfoil section blades) vs. simple stamped sheet metal blades.[22] The industry generally feels that 0.75 in. tip clearance is required when the fan is mounted on the engine and the shroud is mounted on the radiator. This allows for movement in the engine supports due to changes in the torque produced by the engine. A 0.05 in. tip clearance would not be possible without fixing the fan and shroud relative to each other. Tip clearance would effectively be zero if a rotating shroud were used although sealing the shroud to the radiator would be required and the clearance between the rotating and fixed portions of the shroud should be minimized.

Shroud Type. The types of fan shrouds are illustrated in Figure 2.14. In Figure 2.13d a bellmouth (or venturi) shroud was found to give about a 3% increase in the heat transfer performance over the square edge (or box type) shroud. An intermediate level of performance is generally found with ring type shrouds. As shown in Figure B.2 of Appendix B the ring shroud also provides a significant performance improvement over the flat plate (or box) shroud. As with shroud clearance improvements, a simple sheet metal fan may not have nearly the performance improvement as one with sophisticated geometry when the shroud design is improved.[22] The venturi shroud provides the greatest performance benefits but is also the costliest and works best with low to moderate airflow resistance systems. The box shroud is least complex, least expensive, but also least efficient. In high airflow resistance systems, it can provide comparable performance to the more sophisticated configurations. The ring shroud is a compromise design and has moderate performance, cost, and complexity.

Fan-to-Engine Block Clearance. The effect of fan-to-engine-block clearance is illustrated in Figure 2.13e. This figure shows that the fan-to-engine-block distance over the range of 6 to 14 in. (.2 to .5 fan diameters) gave negligible increase in the heat transfer performance of the system.

In addition to the above effects, the following variables affected fan performance as indicated:[18]

- Increasing blade pitch angle decreased peak efficiency significantly.

- Increasing blade camber (curvature) had little effect on effeciency up to some critical value of camber beyond which efficiency dropped rapidly.

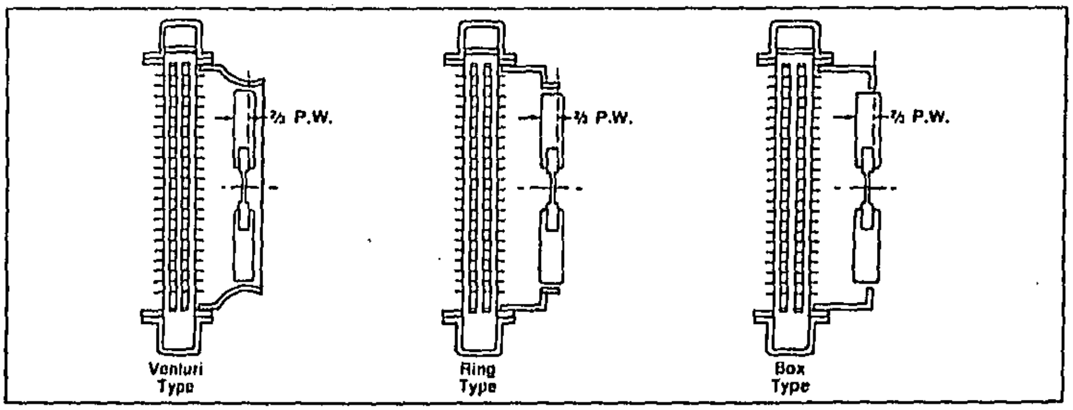

FIGURE 2.14. TYPES OF FAN SHROUD[10]

- Increasing the number of blades above some minimum desirable solidity (ratio of blade area to fan disc area) slightly decreased efficiency.

- Increasing tip solidity with constant number of blades had insignificant effects on efficiency.

- Doubling blade chord with constant solidity increased efficiency slightly due to smaller effective tip clearance.

III. FAN NOISE GENERATION

In this chapter the understanding of the fan noise generation mechanism, as it exists currently, will be discussed. The effect of fan design and installation on noise generation will be reviewed. Finally, formulas for the prediction of fan noise in vehicles will be discussed.

NOISE GENERATION MECHANISMS

Noise generated directly or indirectly by engine cooling fans can be described in six categories:

- Tip vortex interaction noise
- Vortex shedding noise
- Rotational noise due to blade/inflow interaction
- Blade stall
- Fan blade and shroud structural resonances
- Noise due to fan outflow/engine compartment interactions.

The first three aeroacoustic noise generation mechanisms are fundamental to the fan functioning. The latter three mechanisms are essentially installation design problems.

The vortex shedding and rotational noise mechanisms characteristically occur at various points over a fan's operating range, as illustrated in Figure 3.1. The tip vortex noise, for a given fan tip-to-shroud clearance, increases significantly for high pressure rise requirements. Rotational noise is a strong function of the degree of inflow distortion and is relatively insensitive to fan back pressure except near a stall. The vortex shedding noise dominates only when the fan is lightly loaded, i.e., highest flow coefficients, and is only of significance when the fan is operated with the low level of rotational noise and small tip clearance.[19]

Tip Vortex Noise

Tip vortex noise results when a heavily loaded fan blade becomes stalled and sheds an unsteady wake in which the following blade becomes immersed. This large tip vortex interacts with the trailing edge of the leading blade and the leading edge of the following blade to become a source of noise. Tip vortex noise is the noise mechanism which is quieted when blade tip-to-shroud clearances are reduced. For large tip clearances this noise is relatively consistent and increases fan noise level 10 to 15 dB over the full fan operating range. For small clearances, however, (approximately 3 to 4% of blade cord), tip vortex noise becomes dominant only when the fan is heavily loaded.* Tip vortex noise is eliminated when tip clearance approaches zero (approximately 0.09% blade cord). The tip clearance can best be reduced -- considering practical constraints -- by the use of an integral, i.e., rotating, shroud fan for which significant (6 to 10 dB) noise reductions have been reported.[19,23]

Rotational Noise

Rotational noise is a result of the airflow distortion and turbulence entering the fan disk. It results in the characteristic tonal spectra at blade passage frequencies and overtones. This turbulence is, in part, created

*Longhouse found tip vortex noise to be a function of blade chord and not blade diameter as is normally discussed.[19]

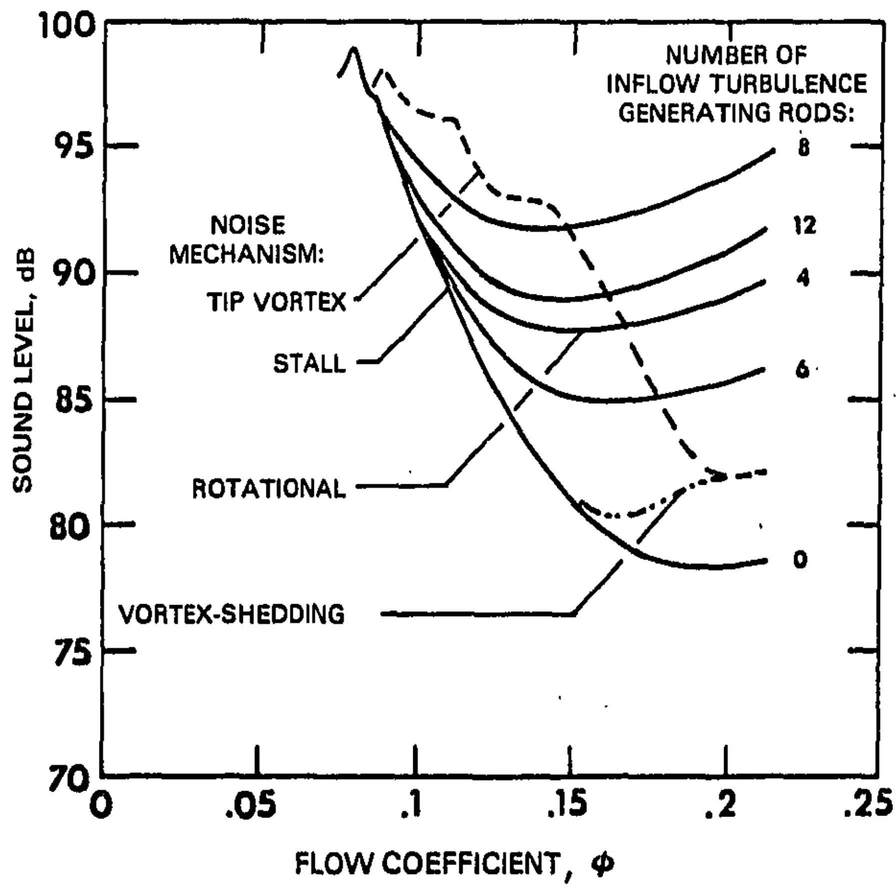

FIGURE 3.1. CHARACTERISTIC CURVES OF THE FAN NOISE MECHANISMS

by the radiator; however, the radiator also tends to act as a flow straightener[24] minimizing the effects of grill and other large amplitude inlet flow pertubations. This mechanism can be quieted by operating the fan more heavily loaded and with reduced tip speed, such as by increasing blade tip angle. The use of uneven blade spacing changes the location of the blade passage frequency and its overtones but does not significantly affect the overall sound level generated by this mechanism.[25]

Vortex Shedding Noise

Vortex shedding noise apparently results when vortices shed from a laminar boundary layer on one side of the blade interact with turbulence created from other side of the blade. This interaction creates acoustic waves which then travel up stream to the point of origin of the boundary layer instability and, with proper phasing, produce a resonance which generates strong narrow-band noise. While this mechanism is quite narrow in bandwidth, it appears in the fan spectrum broadband in nature. This broadband characteristic is the result of the summation of the vortex shedding noise sources along the span. Since this mechanism requires a laminar boundary layer on one side of the blade, it is eliminated when the flow around the blade is fully turbulent. This can be accomplished by such turbulence generating mechanisms as leading edge serrations. Operating the blades in naturally turbulent conditions, by increasing the local Reynolds number, will also eliminate vortex shedding noise. This can be accomplished by operating at higher blade speed, or increasing blade loading or blade chord.[19]

Installation Noise Mechanisms

Blade stall is a result of operating a fan beyond its pumping capabilities. This results when shutoff conditions occur over all or part of the fan inlet area, such as when upstream flow obstructions exist. These obstructions may be: closed radiator shutters and zipped grill covers, a piece of cardboard in front of the radiator, or the plugging of the radiator due to dirt and insect entrapment. This noise mechanism is expected to increase fan sound levels 2-5 dB.

Fan blades and shrouds are the structures which are primarily susceptible to structural resonance. The fan blade is essentially a beam cantilevered from the fan hub and will have a family of resonances -- which for most fan geometries are probably best determined experimentally. (Note that in use the fan will appear stiffer due to the effect of centrifugal force.) These resonances will be excited primarily by fan inflow distortion and result in periodic loading and unloading of the blade as a function of fan speed. In this context, use of plastic fans are attractive since their higher internal damping will make them less responsive at their natural modes. The shroud is essentially a thin-walled shell. It is most likely excited at the throat of the shroud by the passing tip vortices. This loading would be periodic at the blade passage frequency. Both the shroud and fan blade resonances would be excited only at certain fan speeds and would not scale with blade speed.

Noise resulting from aeroacoustic interactions between the fan outflow and objects and structures in the engine compartment has had little or no discussion in the literature but may be particularly significant as other noise sources are quieted. Airflow exiting the fan may locally be at speeds of greater than 120 ft/s.[26] Obstructions downstream of the fan (fan drive belt pulleys, mounting brackets, and miscellaneous sharp edges) can interact with this airflow and result in aerodynamically generated noise. The amplitude of this noise will be related to airflow velocity and, thus, fan speed.

FAN SOUND LEVELS AND SPECTRA

In the past many investigators have discussed the fan sound levels in terms of A-weighted overall sound levels. However, unweighted sound power level spectra, taken at various speeds but at the same flow coefficient for a given fan, normalize with harmonic number.[18] This normalization occurs because both the broadband vortex shedding spectra and the discrete rotational noise spectra vary with fan speed.

The vortex shedding noise mean frequency (at a given flow coefficient for a given fan design) normalizes with Strouhal number, defined as

$$S = \frac{fl}{V}$$

where f = mean frequency

l = characteristic length

V = the average flow velocity through the fan $\propto$ Nd.

The Strouhal number can be shown to be proportional to 60 f/N, harmonic number.

In the following discussion the fan sound level will be discussed in terms of unweighted overall sound power level. Fan noise due to structural resonances, however, will not be a function of fan speed, but will not be considered in the following discussion since it is not inherently part of the fan noise generation characteristics.

The aerodynamic performance of a fan can be described in terms of its pressure coefficient and efficiency curves as a function of flow coefficient, such as is shown in Figures 3.2a and b. The characteristic sound power performance of a fan is that sound power level at a given airflow and pressure rise rather than at constant speed and diameter. This can also be presented as a function of flow coefficient as shown in Figure 3.2c.[18,27] This characteristic sound power level is also called "sound power level coefficient" by Mellin[6,7] and "specific noise level" by Baranski and Pisarski.[28*] Note that the flow coefficient corresponding to peak efficiency is not the same flow coefficient corresponding to minimum characteristic sound power level.[18,28] Mellin found that at a given flow coefficient the sound power generated by the fan varied as

$$W \propto Q \Delta P^{(x-1)/2}$$

*Baranski and Pisarski's specific noise level parameter is similar to the sound power level coefficient parameter of Mellin, except that it is described in terms of A-weighted sound pressure levels.

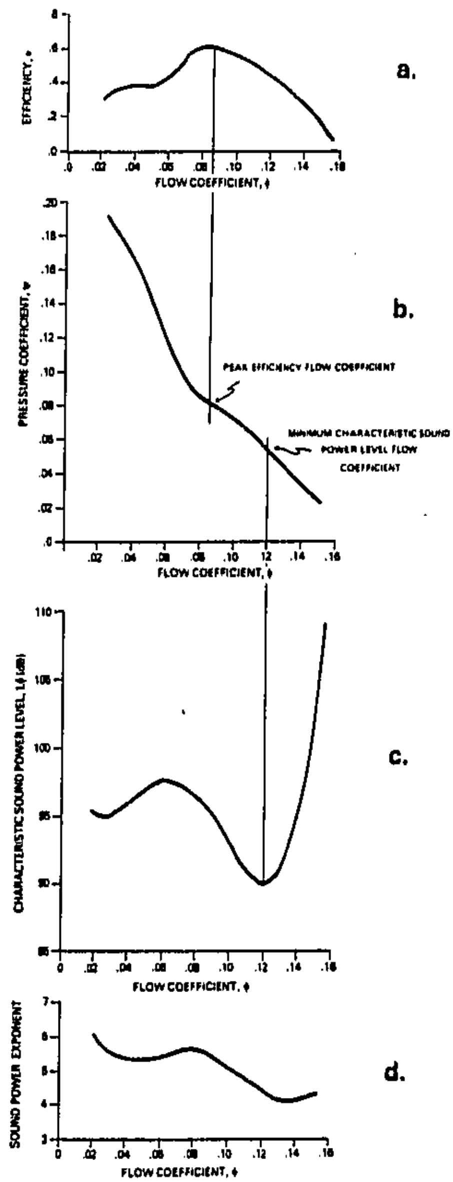

FIGURE 3.2. TYPICAL FAN AERODYNAMIC AND ACOUSTICAL PERFORMANCE CURVES

where x is the sound power exponent. This sound power exponent varies with fan design and flow coefficient of operation. Its magnitude appears to be related to the particular combination of sound generation mechanisms associated with the given fan design and operating point. Its value must be found experimentally from

$$x = (L_{W_2} - L_{W_1})/10 \log (N_2/N_1), \ \phi = \text{constant}$$

This result, as a function of flow coefficient, is illustrated in Figure 3.2d.

Historically, investigators had assumed that x was independent of flow coefficient and assumed a value in the range of 5 to 6 -- an assumption not born out by close experimentation. Sound power exponent also relates the variation of sound power with fan diameter and speed.

$$W \propto N^x \Delta p^{x+2}$$

Note that, if $x = 5$, this relationship reduces to the classically assumed $N^5 d^7$ relationship.

Taken together the work of Longhouse and Mellin have considerable significance. At any given flow coefficient, the total sound power level of the fan can be considered the sum of sound power contributions of the three aeroacoustic noise mechanisms

$$L_{W_T} = 10 \log \left(10^{L_{W_{TV}}/10} + 10^{L_{W_R}/10} + 10^{L_{W_{VS}}/10}\right), \ \phi = \text{constant}$$

where the subscripts denote: TV, tip vortex noise; R, rotational noise; and VS, vortex shedding noise. These mechanisms will have relative contributions which will then dictate the characteristic sound power level and sound power exponent at that flow coefficient. Presumably, each of these mechanisms has its own sound power exponent and the observed sound power exponent of the total sound power is essentially the weighted average at the specific flow coefficient. As a result, a model of fan noise generation can be developed which should accurately predict fan sound power levels and allow the optimization of fan configurations for given performance.

These recent findings also have significant implications for fan selection. The non-coincidence of the peak efficiency and minimum characteristic sound power level flow coefficients means that value judgments and trade-offs must be made in fan selections. This situation appears to be recognized by at least one fan manufacturer.[28] For rational fan selection, a family of fan aerodynamic and acoustical performance curves -- such as illustrated in Figure 3.2 -- will be necessary. As discussed in Chapter II, these curves should be derived using procedures which are representative of vehicle engine compartment environments.

FAN DESIGN AND INSTALLATION EFFECTS

A number of factors affect noise generation characteristics of a given fan and its installation. A review of these effects appears in the following paragraphs.

Blade Tip Clearance. Reduced tip clearances provide improved aerodynamic performance and allow lower fan speeds for a given performance. In addition, reduced tip clearance fans are inherently quieter. Compared to large tip clearance fans (<3% chord), rotating shroud fans are significantly quieter (as much as 10 dB).[19] For fans with fixed shrouds, reductions of 2 to 6 dB have been observed.[18] A 7 dBA sound level reduction was observed for a fan with rotating shroud immersed completely in a venturi type fixed shroud.[23]

Blade Shape. Various shapes have been proposed for fan blades to achieve quieter operation. These include serrations[19] and guide vanes.[29] The use of leading edge serrations reduced vortex shedding noise 10 to 20 d but does not have any significant effect on rotational noise. A guide vane fan has been designed which includes two guide vanes attached to the suction surface of the fan at approximately 1/2 and 3/4 blade span. This fan provided 35% increased pumping capability, approximately 8% improved efficiency, and was 4 dB quieter for a given airflow compared to the same fan geometry without guide vanes. The guide vanes are intended to direct the flow radially from the fan, intentionally inducing a mixed flow characteristic and compensate for large tip clearances required for fixed shrouds. (Unfortunately the comparison data provided above did not specify shroud tip clearance.) These guide vanes presumably reduced both tip vortex noise and rotational noise.

A skewed or swept blade fan has been evaluated which provided approximately a 10 dBA reduction in broadband noise and about a 4 dBA reduction in discrete frequency noise when compared to an unswept fan of comparable solidity and number of blades. While this study did not include aerodynamic performance, evaluation of the fan in a heavy truck indicated the skewed blades had an adverse effect on truck cooling performance.

Blade Chord. Increasing blade chord tends to reduce both tip vortex and vortex shedding noise mechanisms. A reduction of almost 6 dB at some operating points was observed for a fan with a tripled tip chord. Sound level reductions of as much as 8 dB were observed for higher solidity fans using increased blade chord. However, for constant solidity, sound level generation was unchanged with increasing blade chord (with proportionally decreasing blade number).[18]

Number of Blades. Increasing the number of fan blades from 4 to 8 resulted in a 9 dB increase in fan sound levels for lightly loaded blades. However for more heavily loaded conditions, the number of blades was insignificant.[6] As mentioned before, the spacing of the blades does not significantly affect the overall sound power although the tonal content of the fan is significantly altered.[25]

Blade Pitch Angle. When noise level is adjusted to a given performance, a 7 dB decrease minimum noise level is obtained by increasing pitch angle from 16° to 26°. However, sound level increased about 2 dB as pitch angle increased further to 39°. (Note that with increasing pitch angle, peak efficiency decreases in magnitude and occurs at lower loadings.)

Blade Camber. Characteristic sound power level decreased by as much as 10 dB as curvature was decreased (for increasing radius of curvature from 82% to 150% of blade chord) for fans with low loadings. At higher loadings difference between curvatures was insignificant.[18]

VEHICLE FAN NOISE PREDICTIONS

Over the years a number of investigators have attempted to provide simplified fan noise prediction formulas for vehicle fans. These formulas are summarized in Table 3.1 (They are presented in a form such that they predict the sound pressure level from the vehicle fan at 50 ft. In some cases liberties have been taken with the authors' original formulation to allow a comparison to the other equations.) These equations are reviewed to caution users of their limitations.

From the previous discussion, inspection of these equations should make it immediately obvious that some of them are incorrect. In all of the equations except equation 10, simplified sound power exponents have been assumed, generally, in the range of 5 to 6. Equations 3, 4, 5, 6, 8, and 9 attempt to predict A-weighted sound levels, thus ignoring the frequency dependence upon fan speed. Equations 5 and 6 are singled out for particular comment. Both are somewhat more ambitious in relating noise to fan parameters. Equation 5 includes an effect of solidity (nB) which conflicts with recent findings.[18] Equation 6, which appears quite rigorous, was in fact based on test of a single vehicle (1964 Plymouth Fury V-8) with a 4-bladed fan and consequently does not have statistical validity. Finally, Equation 9 can be derived from Equation 10 if the sound power exponent, x, is equal to 5.

TABLE 3.1

VEHICLE FAN NOISE PREDICTION FORMULAS

(Sound Pressure Level at 50 ft)

Equation Number	Expression	Reference
1	$L = K + (55\text{-}60) \log N$	Priede/NCHRP 173
2	$L = K + 60 \log N$	Wyle/NCHRP 173
3	$L_A = 76.7 + 70 \log \frac{d}{30} + 50 \log \frac{N}{2000}$	ASHRAE/NCHRP 173
4	$L_A = 76.1 + 72 \log \frac{d}{30} + 52 \log \frac{N}{2000}$	Daly/NCHRP 173
5	$L_A = K + 50 \log N + 50 \log d + 10 \log nB$	Damkevala, et.al.
6	$L_A = 75.8 + 60 \log \frac{N}{2000} + 60 \log \frac{d}{30} + 20 \log \frac{b}{3} + 10 \log \frac{n}{6} + 20 \log \frac{\overline{w}/Um}{0.07}$	Serendipity (BBN)/NCHRP 173
7	$L = K + 9 \log P + 28 \log \frac{N}{1000} + 30 \log \frac{d}{12}$	from Hawes
8	$L_A = -103.1 + 9 \log \rho \left[\frac{P}{\rho d^5 \left(\frac{30N}{\pi}\right)^3} \right]_\phi + 60 \log N + 70 \log d$	from Rising (from Hawes)
9a	$L_A = L_{\phi_A} + 10 \log Q + 10 \log \Delta p^2$	from Baranski and Pisarski
b	$L_A = L_{Nd_A} + 50 \log N + 70 \log d$	
10a	$L = L_\phi + 10 \log \frac{Q}{3534} + 10 \log \left(\frac{\Delta p}{1.34}\right)^{\frac{x-1}{2}}$	from Mellin
b	$L = L_{Nd} + 10 \log \left(\frac{N}{1910}\right)^x + 10 \log \left(\frac{d}{19.68}\right)^{x+2}$	

K = sound level constant, dB

N = fan speed, RPM

d = fan diameter, in.

n = number of fan blades

B = area per blade, in.^{2}

b = fan blade width, in.

$\overline{w}/Um$ = airflow turbulence index (due to radiator and grill)

P = fan power, HP

ρ = air density, slugs/ft^3

ϕ = flow coefficient

L_ϕ = fan sound level at flow coefficient ϕ

L_{Nd} = fan sound level at specified speed and diameter

Q = airflow, ft^3/min (standard conditions)

Δp = fan pressure rise, in. H_2O

x = fan sound power exponent, ϕ = constant $= \left(L_{W_1} - L_{W_2}\right) / 10 \log (N_1/N_2)$

IV. ALTERNATIVE COOLING SYSTEM CONFIGURATIONS

Various design configurations are discussed in this section starting with an illustration of the current practice for a large gasoline-engined medium truck. Three alterations to this baseline design will be successively made until all three alterations are included in the final example. The purpose of this exercise is not to determine the optimum specification for the baseline vehicle, but to illustrate the potential benefits in reduced fan performance requirements for various types of cooling system design changes. An additional section will discuss other possible configurations for the cooling system. The final section will then examine the potential sound level benefits from the baseline of the three detailed alterations considered.

CURRENT PRACTICE

A GMC medium truck, which could be equipped for inner-city delivery work, is selected as a baseline example of current practice. The GVW of the truck, with body, is 25,000 lb., the highest capacity for this classification of medium truck. Table 4.1 shows the specifications of the example truck. The engine selected for this example is a 427 $in.^3$ V-8 gasoline engine, currently the largest gasoline engine now installed in GMC trucks. The design condition for the cooling system at full power and with air conditioner is $110^{o}F$ air-to-boil (ATB) which corresponds to a $120^{o}F$ ATB with the air conditioner not in operation. The fan installed in the baseline medium truck is manufactured by Hayes-Albion, Inc. It is a sheet metal fan formed from cambered uniform thickness blades attached to a metal spider. The spider is

TABLE 4.1

EXAMPLE MEDIUM TRUCK SPECIFICATIONS[13]

Manufacturer:	General Motors Corporation
Model:	GMC C7D042
Radiator:	Tube and center sideflow 4 tube rows 28.8 in. wide x 24.1 in. high x 2.63 in. thick 12.5 fins/in., louvered
Fan:	Hayes-Albion Corporation Model: 2028749 22 in. dia x 2.50 in. proj. width 7 equally spaced blades Part: GM-G-167-F-1
Engine:	427 $in.^3$ V8, gasoline
Fan Drive:	Eaton 280 Viscous Clutch
Peak HP test:	Air-to-Boil: 120^0F (without air conditioning) : 110^0F (with air conditioning) Heat Rejection: 6160 BTU/min (without air conditioning 6602 BTU/min (with air conditioning) Water Pump Flow: 70.0 gal/min Maximum Engine Speed: 4000 RPM Maximum Fan Speed (engaged): 3530 RPM Ram air velocity: 5 MPH

attached to the output of the fan clutch. The use of seven blades combined with 22 in. diameter indicates that this is a high capacity fan. The great majority of current practice medium truck fans are similar. The performance curve for this fan in a wind tunnel test is the first curve found in Chapter II as Figure 2.10a, or in Appendix B, on page B-2.

The radiator installed in this truck is a tube-and-center side-flow radiator* with four tube rows and 12 fins/in. The fins are perforated with louvers. (These fins are manufactured by taking one continuous sheet of metal, perforating it, folding it into a serpentine ribbon, and then soldering it between the two adjoining tube columns. Each serpentine section which results has four tubes on each side.)

Calculations of the airflow required to dissipate the heat generated by the engine are contained in Appendix C. In these calculations a radiator temperature drop of 10° is assumed and the radiator heat transfer performance is assumed to follow the performance of a geometrically similar radiator. Given the air-to-boil requirement (without the air conditioning system in operation) and an assumed maximum top tank temperature of 210°F (based upon GM Detroit Diesel Allison Division diesel engine practice), the design ambient of 118°F was calculated. This design ambient is used for the following alterations to the baseline design.

Based upon the above assumptions and the calculations described in Appendix C, the volumetric airflow required of the fan (at fan inlet temperature) was calculated. Ignoring any installation effects, the pressure rise of the fan was estimated for the calculated airflow and 3530 RPM fan speed. The resultant cooling system operating conditions are summarized as:

- Required airflow, 6890 ft^3/min (at standard conditions)
- Engine compartment temperature, 167°F

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*In common radiator design practice, coolant enters at the top of the radiator, travels downward, and exits at the bottom. General Motors uses a radiator design in which coolant enters at one side of the radiator, travels laterally across the radiator, and exits the other side. This design permits reduced radiator heights.

- Fan airflow, 8310 ft^3/min
- Fan pressure rise, 3.20 in. H_2O
- Air power, 3.47 HP.

INCREASED TOP TANK TEMPERATURE

One of the most efficient ways of increasing the heat transferred for a given size radiator is to increase the differential between the two incoming fluids. Since the operating ambient cannot be changed, the only temperature which would accomplish this is the top tank temperature, the temperature of the coolant leaving the engine. Table 4.2 shows typical maximum top tank temperatures now being used. The top tank temperature limitations specified by manufacturers of diesel engines used in medium and heavy trucks are all based on the premise that it is possible that water might be used as coolant and that the system should be able to operate without the pressure cap in place. The practice with automobiles, however, has been to design the system assuming that the pressure cap will function properly and that the system will be pressurized. Nevertheless, even the automobile manufacturers do not take advantage of the increased boiling point which will be present if the proper solution of ethylene glycol and water is used. The boiling point used as a

TABLE 4.2
TYPICAL MAXIMUM TOP TANK TEMPERATURES

MANUFACTURER	ENGINE TYPE	SPECIFIED TEMPERATURE °F
Cummins Engine Co., Inc.[32]	diesel	203
Caterpillar Tractor Co. Caterpillar Engine Div.[33]	diesel	210
General Motors Corp. Detroit Diesel-Allison Div.[8]	diesel	210
Chrysler Corp.[34]	gasoline	245

design for Chrysler Corporation automobile engines is based on a pressure cap opening at 14 lb/in^2. At 14 lb/in^2, water boils at a temperature of 247°F. Thus, the 245°F shown in Table 4.2 for Chrysler represents the same margin between maximum top tank and boiling temperatures as used by both Caterpillar and GM, assuming atmospheric conditions and water coolant.

As the first alteration on the baseline medium truck cooling system, an increase in the top tank temperature to 245°F is evaluated. The calculations of the required flow with this higher top tank temperature are also contained in Appendix C. For estimating this effect, the coolant temperature drop across the radiator is assumed as in the baseline condition. With the higher temperature differential between the entrance of air and water now available, the required airflow through the radiator to obtain the same heat flow is greatly reduced. However, with the lower airflow rate the temperature in the engine compartment is greatly increased, since a smaller quantity of air must absorb the same heat. The required airflow at the entrance to the radiator is reduced from 6890 ft^3/min to 3840 ft^3/min, nearly a 50% drop in the required airflow. The pressure drop through the system is reduced to approximately 1/3. The increased temperature in the engine compartment increases the volume of air passing through the parts of the flow path down stream of the radiator so the full benefit of reduced pressure, to 1/4 its original value, cannot be realized. The air power required from the fan is a function of the product of the pressure rise across the fan and the airflow. Air power drops from 3.47 HP to 0.677 HP, only 20% of the required fan power in the original baseline design.

In summary the conditions with increased top tank temperature are as follows:

- Required airflow, 3840 ft^3/min. (at standard conditions)
- Engine compartment temperature, 206°F
- Fan airflow, 4910 ft^3/min
- Fan pressure rise, 1.12 in. H_2O
- Air power, 0.677 HP.

MULTIPASS RADIATOR

The efficiency of a heat exchanger is defined as the heat transferred divided by the potential heat transfer. Since both the mass of air and the specific heat of air are less than those of a water and ethylene glycol mixture, the limitation on all automotive radiators is due to the air-side heat transfer. Thus, potential heat transfer can be estimated as that which could be transferred to the quantity of air which moves through the radiator, if it were brought up to the temperature of the entering water. As the efficiency of the radiator increases, the air temperature is brought closer to that of the coolant. When air and coolant temperatures leaving the radiator get closer, benefit can be obtained by increasing the number of passes in the radiator. This keeps the temperature differential between the air and water at a maximum for any position in the core. The baseline vehicle has a radiator containing four tube rows. With suitable baffling in the inlet and outlet tanks of this radiator, these four tube rows can be made to function as separate passes with the highest water temperature occuring in the tube row closest to the engine. Figure 4.1 shows how baffling may be used to create a multipass radiator from the same core geometry. The core illustrated in Figure 4.1 is a five tube row core. Its odd number of passes allows the entrance to be on one side of the radiator and the exit from the radiator to be on the other side as in conventional single pass radiators. If this was a downflow radiator, the top tank would still be located above the radiator core as it now is and the bottom tank would contain the exit from the radiator to the water pump.

The use of a multipass radiator does present some design problems. In order to maintain the same 10°F temperature drop across the radiator with more effective heat transfer, an increased water flow rate will be required. This increased water flow rate may lead to more possibility of vapor lock occurring, since the water velocity in the tubes is now more than four times that of the baseline design due to baffling and the increase in a heat transfer effectiveness.

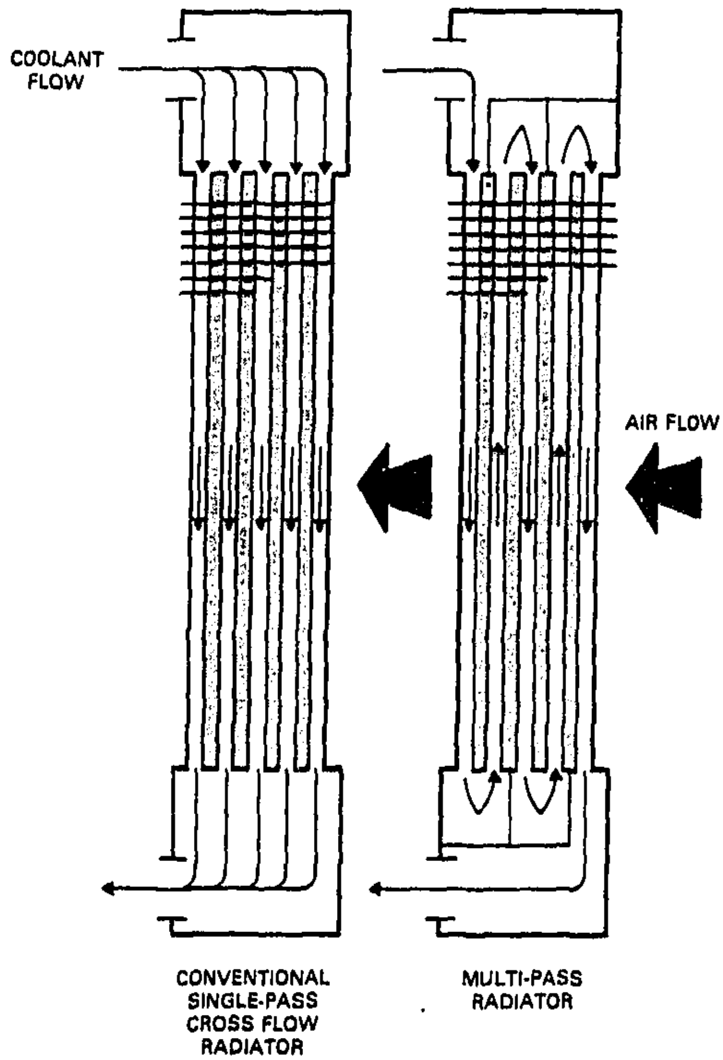

FIGURE 4.1. RADIATOR HEAD TANK BAFFLING FOR MULTIPLE COOLANT PASSES

The effectiveness of the high top tank temperature radiator calculated from the results in the previous section is 0.694. Entering Figure 2.9 in Chapter II with this effectiveness, the number of heat transfer units (N_{TU}) are determined at 3.2. At 3.2 heat transfer units, the effectiveness of a four pass heat exchanger is 0.750. This higher effectiveness can be used to calculate the reduced flow required, the reduced pressure drop through the system and the power needed from the fan. In summary these conditions are as follows:

- Required air flow, 3550 ft^3/min (at standard conditions)
- Engine compartment temperature, 213°F
- Fan air flow, 4590 ft^3/min
- Fan pressure rise, 0.979 in. H_2O
- Air power, 0.548 HP.

THICK RADIATOR CORE

The effect of radiator core thickness on heat transfer effectiveness is also shown in Figure 2.9. The number of heat transfer units increases as the air flow path length increases. Core thickness is thus directly proportional to the number of heat transfer units. As discussed in Chapter II, the thickness of radiator core can be optimized on the basis of minimum required air power. This optimization approach differs from that normally taken where weight and cost are the primary considerations. Calculations of the air power required for each 0.25 in. of thickness, starting with 3.00-in. thick radiators, were made. The minimum required air power was found to occur for the 3.00-in. thick radiator, for which calculations are contained in Appendix C. A 3.00-in. thick radiator with four passes has an effectiveness of 0.773 vs. the .750 effectiveness for the 2.63-in. thick radiator of the previous example. The increased core thickness causes a penalty in radiator pressure drop to be incurred proportional to the thickness. Because of this penalty, core thicknesses of 3.25 in. thick and greater were found to have higher required air powers even though the flow requirements were reduced. The resultant cooling system operating conditions are:

- Required air flow, 3450 ft^3/min (at standard conditions)
- Engine compartment temperature, 216^oF
- Fan air flow, 4490 ft^3/min
- Fan pressure rise, 3.00 in. H_2O
- Air power, 0.540 HP.

OTHER ENGINE/COOLING SYSTEM CONFIGURATIONS

Arrangements other than those just discussed could be considered to reduce cooling system noise, but most of these require greater changes from the baseline condition and some suffer from a lack of design data needed to evaluate them. Table 4.3 summarizes the advantages and disadvantages of five additional cooling system configurations. These modifications are listed in the order of their perceived complication with the least complex first.

High Fin Density Radiator Core

The first modification considered is the high fin density radiator core. As discussed in Chapter II, a louvered-fin radiator results in a less effective radiator when the fin density is increased above 12.5 fins/in. because the pressure drop increases more rapidly than increases in heat transfer. However, data exist which indicate that going to unperforated plate fins allows more heat transfer at the same pressure drop. Such a radiator would certainly have a greater cost than the louvered-fin radiator and would also have higher weight due to the greater amount of metal used in the core. The optimum fin density would have to be determined on the basis of the minimum required air power by performance of calculations similar to those developed for the thick core, 12.5 fins/in. radiator, contained in Appendix C. The relative values of the heat transfer effectiveness and pressure drop in the given application will determine which radiator is best. No single optimum fin density will likely emerge for all applications.

Low Water Flow Engine

At least one manufacturer is now working on an engine which has a lower water pump flow which results in a higher water temperature drop through

TABLE 4.3
OTHER ENGINE/COOLING SYSTEM CONFIGURATIONS

Description	Advantages	Disadvantages	Highlights
High Fin Density Radiator	More heat transfer at same pressure drop	Higher weight, cost	Lower "optimum" thickness than baseline
Low Water Flow Engine	Reduced water pump load	Lower air-to-coolant temperature differential	May combine well with higher top tank temperature or counterflow radiators
Reduced Cooling Requirement	Reduced cooling system requirements	Higher engine operating temperature; significant change in current practice	Uses insulated pistons, altered coolant passages; Army experimenting with no coolant
Auxiliary Radiators	Lower primary radiator requirements. Reduced fan-use when ram air is sufficient	Difficult to obtain airflow, auxiliary fans	May be driven by ram air or use auxiliary fans
Air-/Water-Cooled Engine	Reduced cooling system requirement	Grease and dirt blocked fins may reduce effectiveness; added weight, complexity and cost	Separate flow path for radiator and engine and/or additional booster fan may be required

the radiator while rejecting the same quantity of engine heat. Although this reduces the water pump load on the engine and is likely to increase the engine efficiency by reducing parasitic load, this modification will increase the required fan power on the air-side flow path. In order to minimize the increased fan power required, this low water flow engine may combine well with the use of higher top tank temperatures and multipass radiators to regain the effectiveness of higher water flow engines. Since there will be a greater temperature drop across the radiator on the water side, the average coolant temperature in the engine could be maintained equal to a high water flow case and still allow a substantial increase in the top tank temperature. Use of multiple coolant passes in the radiator would be particularly effective in maintaining turbulent conditions since the water velocity in a tube could be maintained (compared to current practice) by going to a four pass radiator with only one-quarter the water flow, for example.

Reduced Cooling Requirement

The coolant passages around the head and cylinders of the engine could be redesigned such that a greater quantity of the waste heat was left in the exhaust and less engine heat was rejected to the coolant. This would require an increase in the operating temperatures within the cylinders. However, any reduction in the heat required to be dissipated by the engine coolant-- and thus the cooling system-- would also reduce the air power required of the fan. The U.S. Army is now experimenting with operating diesel engines, which have no coolant at all. Such engines have used insulated pistons including ceramic piston heads to maintain the waste heat within the exhaust. Modified Cummins engines which compare to basic current design are now operating in test cells without any coolant.[35]

Auxiliary Radiators

The design speed for the truck cooling system is determined by the grade climbing capabilities of the vehicle which are in turn determined by its power-to-weight ratio. The recent trend has been to increasing power-to-weight ratios; consequently, vehicle design speeds which were formerly 5 mph are now commonly 15 mph. The baseline truck is designed to operate with full power at 5 mph. At this low speed, the benefits of ram air are practically negligable. If the vehicle power is such that the minimum full-power design speed could be increased to about 25 mph, then ram air could be utilized through auxiliary radiators to reduce the cooling capacity of the main radiator system. Such auxiliary radiators might be mounted to the side of the main radiator and could be driven either solely by ram air or could use small electric fans. Any heat dissipation which resulted from installation of such auxiliary equipment would reduce the required heat transfer of the main system and thus the requirement for the fan. Auxiliary radiators might fit behind the headlights or safety lights of a truck. Such radiators would probably be of very little benefit at the low speeds resulting from heavy traffic and this might require a limitation on the engine power when such condition existed.

Air-/Water-Cooled Engine

Another possibility is a combination air-and-water-cooled engine. Such an engine would have cooling fins located at the higher temperature parts of the engine block. Air-cooled diesel engines are frequently used in construction equipment. However, the power levels of such engines are generally less than those of water-cooled engines. One of the biggest disadvantages of use of auxiliary air cooling is that there is great possibility for grease and dirt to block the fins and reduce their effectiveness. Such a grease and dirt buildup might ultimately result in an insulating effect of the fins rather than a benefit. Incorporation of cooling fins on the engine block is bound to add to the weight and cost of the vehicle and may make it more difficult to maintain. If higher top tank temperatures are used with their lower airflows, the resulting higher engine compartment temperatures will reduce the effectiveness of such fins. This may require that a separate flow path be created for the radiator and the engine-fin cooling systems.

The portion of heat lost from a vehicle through convective cooling of the engine block is now quite small. Calculations indicate that at the design condition when the flow through the engine compartment is strictly that due to the fan, the convective heat transfer from the engine block can be as little as 2-3%. To obtain air velocities over the engine sufficient to realize significant heat transfer benefits, an additional engine-mounted fan -- similar in installation to those on air-cooled engines -- may be necessary. (The higher velocity airflow may also result in reduced dirt accumulation in the cooling fins.) This engine booster fan would be thermostatically controlled separately from the radiator booster fan.

As the flow of the fan is reduced through utilization of higher top tank temperatures, or more efficient radiators including multipass types, the engine compartment temperatures will increase to the point that the differential between the engine block and the engine compartment air will be very small. Thus, the heat transfer potential will be even more reduced and the application for an air-/water-cooled engine will disappear unless a separate supply of cool air is obtained for the engine.

POTENTIAL SOUND LEVEL REDUCTIONS

An estimate can be made of the potential sound level reduction based solely on the changes in airflow of the three new configurations from the baseline. Prior to discussing these benefits, an overview of the effects of cooling system configuration on fan operating requirements is provided below.

A summary of the baseline and three modified cooling system configurations is contained in Table 4.4. The first three columns of data show the modifications made and proceed progressively from the first to the third. The last six columns show the required flow and temperature conditions which exist due to these modifications and their potential noise reduction benefits. Each of the modified cooling systems contains the assumption of the same core geometry, i.e., fin density and type, and relation between air-side and water-side surface areas. No benefits have been estimated for improvements in the flow path geometry through use of improved shrouds or cleaning the flow paths. Thus, the benefits in reduced air power are the result strictly of modifying the arrangements of the cooling system and radiator. The greatest benefit occurs by increasing the top tank temperature 35°F. This reduces the required airpower to 20% of the baseline value. Because of operation at a relatively high effectiveness, the introduction of multiple coolant passes

TABLE 4.4

SUMMARY OF ALTERNATIVE COOLING SYSTEM CONFIGURATIONS

DESCRIPTION	TOP TANK TEMPERATURE (°F)	NUMBER OF PASSES	CORE THICKNESS (in.)	REQUIRED AIRFLOW[2] (ft.^{3}/min)	ENGINE COMPT TEMPERATURE (°F)	FAN AIRFLOW (ft^3/min)	FAN PRESSURE (in H_2O)	AIR POWER[2] (HP)	RELATIVE SOUND LEVEL[3] (dB)
Current Practice	210[1]	1	2.63	6890	167	8310	3.20	3.47	0.0
Increased Top Tank Temperature	245	1	2.63	3840	206	4910	1.12	0.677	-11.4
Multipass Radiator	245	4	2.63	3550	213	4590	0.979	0.548	-12.9
Thick Radiator Core	245	4	3.00	3450	216	4490	0.994	0.540	-12.8

ASSUMPTIONS:

Radiator core face area 24.1 in. high x 28.8 in. wide, 12.5 fins/in.
Design ambient 118°
10°F water temperature drop in radiator

NOTES:

[1] Estimated based on Detroit Diesel Allison Div. practice
[2] At standard temperature and pressure
[3] For $L \propto$ 10log (fan pressure)^2(fan airflow), dB re (Current Practice)

becomes more worthwhile than it would be for the baseline condition. Calculations to determine whether multiple coolant passes or increased core thickness were more effective at this point show the multipass radiator to be the more beneficial. Even so, the required air power drops to 80% of its value with the increased top tank temperature for a benefit of 0.13 HP.
With both increased top tank temperature and a multipass radiator already considered, the remaining benefit gained by increasing radiator core thickness is very slight. Only 0.008 HP can be gained by increasing the core thickness from 2.63 in. to 3.00 in.

Figure 4.2 shows the relationship between the fan operating points for each of the four systems. The baseline truck has a total vehicle pressure drop of 3.2 in. H_2O. In this exercise the system pressure drop has been estimated to follow the square of the airflow at the fan. Since the radiator thickness is unchanged and the geometry is identical for both high top tank temperature and the counterflow radiator configuration, the operating points for each of these conditions are on the same system curve but occur at a lower pressure drop and flow condition. When core thickness is increased, the system resistance curve changes, as is indicated by the fourth design not falling on the same curve with the baseline second, and third designs.

The fan or fans which would be selected for the modified cooling systems are likely to be significantly different from the currently installed fan -- even using the same selection criteria (cost, etc.). Thus, an exact calculation of the effect of the changed fan operating points is not possible here. However, for estimation purposes, the effect of reduced aerodynamic performance requirements can be described by.

$$L = L_r + 10 \log \frac{Q}{Q_r} + 20 \log \frac{\Delta p}{\Delta p_r}$$

from Equation 10a of Table 3.1, and assuming 5th order sound level variation with fan speed and all blade designs have comparable noise generation characteristics. Thus, the sound level can now be estimated directly from the

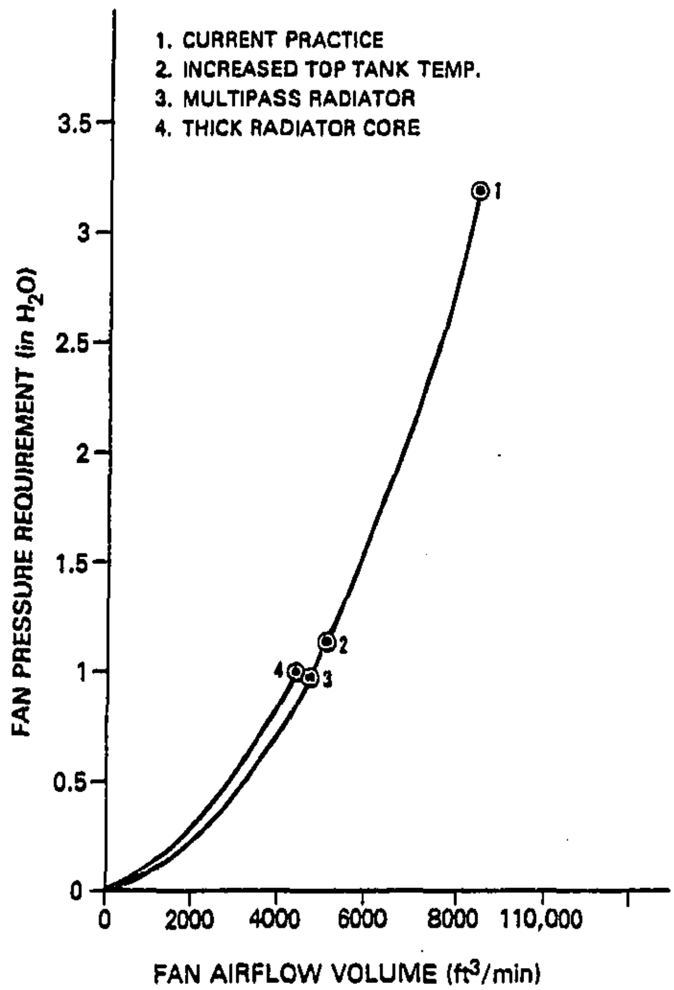

FIGURE 4.2. EFFECT OF COOLING SYSTEM DESIGN ON FAN OPERATING POINTS

required operating conditions. Using this approach, the sound levels for each of the cooling system configurations were calculated (referenced to current practice) and presented in Table 4.4. As seen in Table 4.4, a substantial benefit, 11.4 dB reduction, is obtained from increased top tank temperature. The incremental benefit of the multipass radiator, 1.5 dB, while much less, is not insignificant. The thick core radiator shows a sound level increase as a result of its slightly higher pressure rise requirement; however, with other system configurations -- such as a conventional single pass radiator -- a thicker core radiator may also exhibit sound level benefits.

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APPENDIX A
NOMENCLATURE

A	area, $ft.^2$
B	blade area, $in.^2$
C	heat capacity, $BTU/min/^{o}F$, $= C_p m$
K	sound level constant, dB
L	sound level, dB
N	rotational speed, RPM
P	fan mechanical power, HP
Q	volumetric airflow, ft^3/min
S	Strouhal number = fl/V
T	temperature, ^{o}F
U	velocity, ft/s; overall coefficient of heat transfer, $BTU/min/ft^2/^{o}F$
V	velocity, ft/min
W	sound power, W
b	fan blade width, in.
d	fan diameter, in.
f	frequency, Hz
l	characteristic length
m	mass flow rate, lbm/min
n	number of fan blades
p	pressure, in. H_2O
q	heat transfer rate, BTU/min
r	engine compartment air resistance index
x	sound power exponent
Δ	change
ϕ	flow coefficient, $\frac{Q}{AU_T}$
ψ	pressure coefficient, $\frac{\Delta p}{\frac{1}{2}\rho V^2}$
η	fan efficiency

ϵ	heat exchanger efficiency = $\frac{q}{C_{min}\Delta T}$
π	3.14159
ρ	air density, lbm/ft^3

Subscripts

A	ambient
B	boiling
F	fan
M	mean
R	radiator
S	system (vehicle less radiator)
T	total; tip
TT	top tank

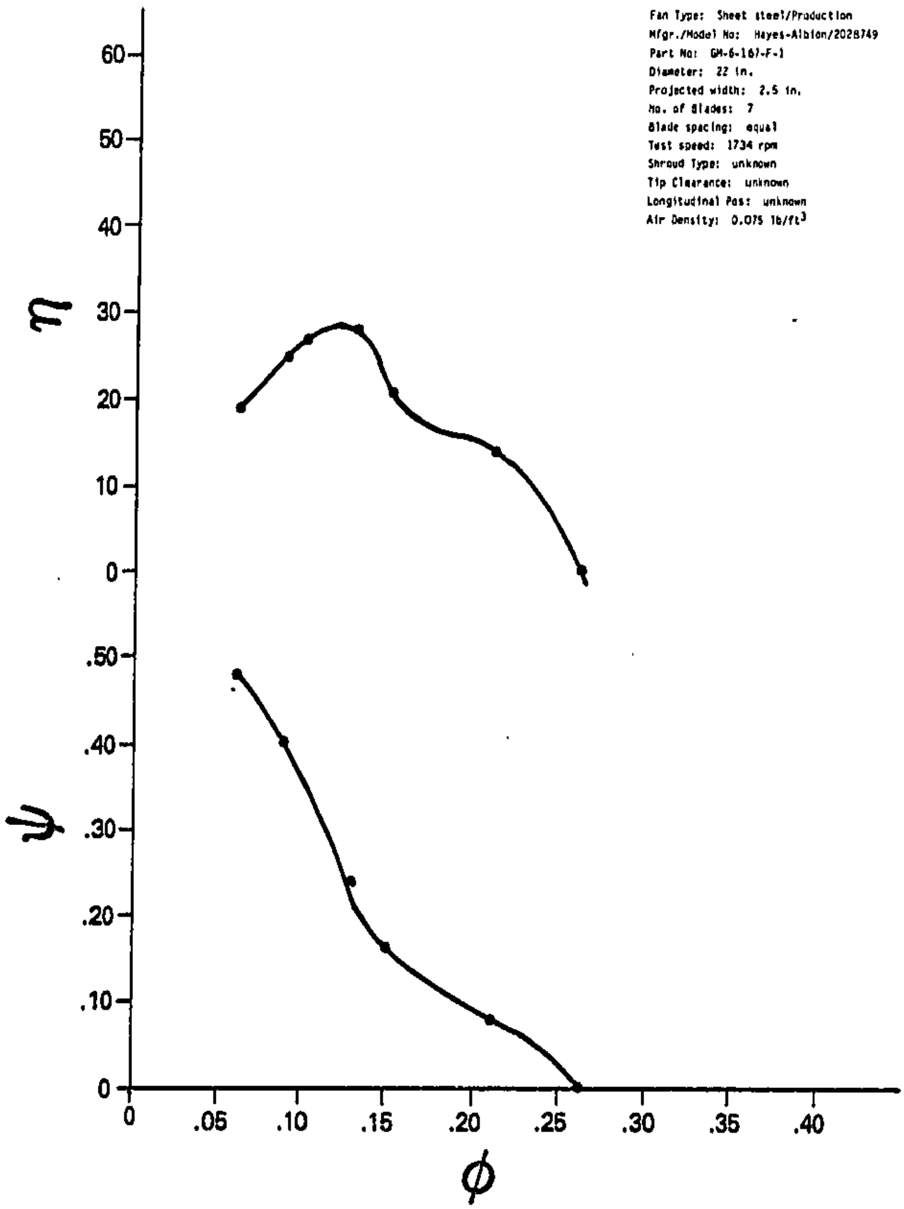
Other

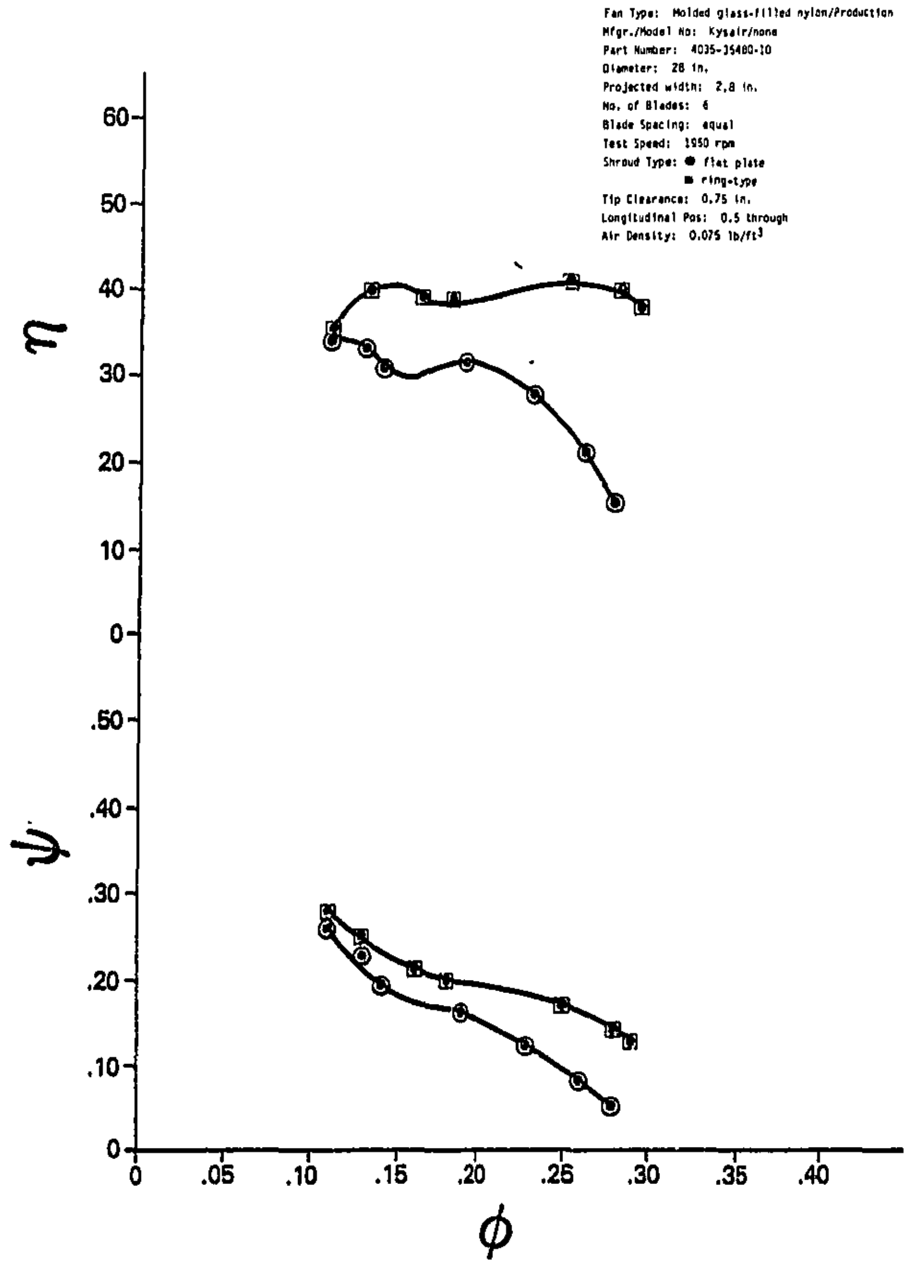
L_A	A-weighted sound pressure level, dBA re 20 μPa
L_W	sound power level, dB re 1 pW
L_ϕ	sound (pressure or power) level at specified flow coefficient, dB
L_{Nd}	sound (pressure or power) level at specified speed and diameter, dB
$\overline{w}/U_m$	airflow turbulence index
C_p	constant pressure specific heat, BTU/lbm/°F
C_{min}	minimum heat capacity of air or coolant in radiator
N_{TU}	number of heat transfer units = $\frac{UA}{C_{min}}$
D_s	specific diameter = $\frac{\psi^{1/4}}{\phi^{1/2}}$
P_A	fan (air) power, HP = $1.58\times10^{-4}\ \Delta pQ$

APPENDIX B

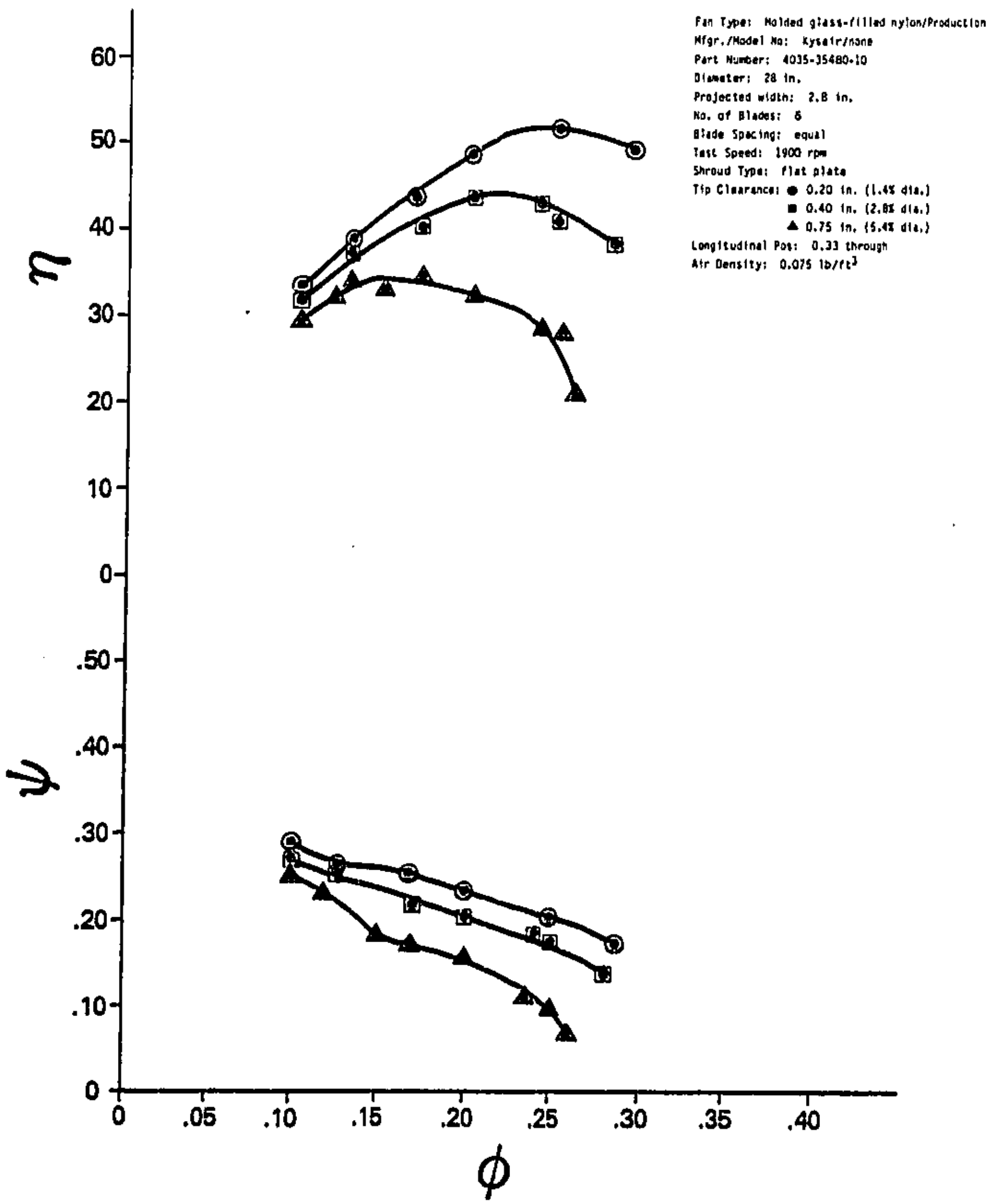
FAN AERODYNAMIC PERFORMANCE CURVES

(Pressure Coefficient, ψ, and Efficiency (in percent), η, versus Flow Coefficient, ϕ)

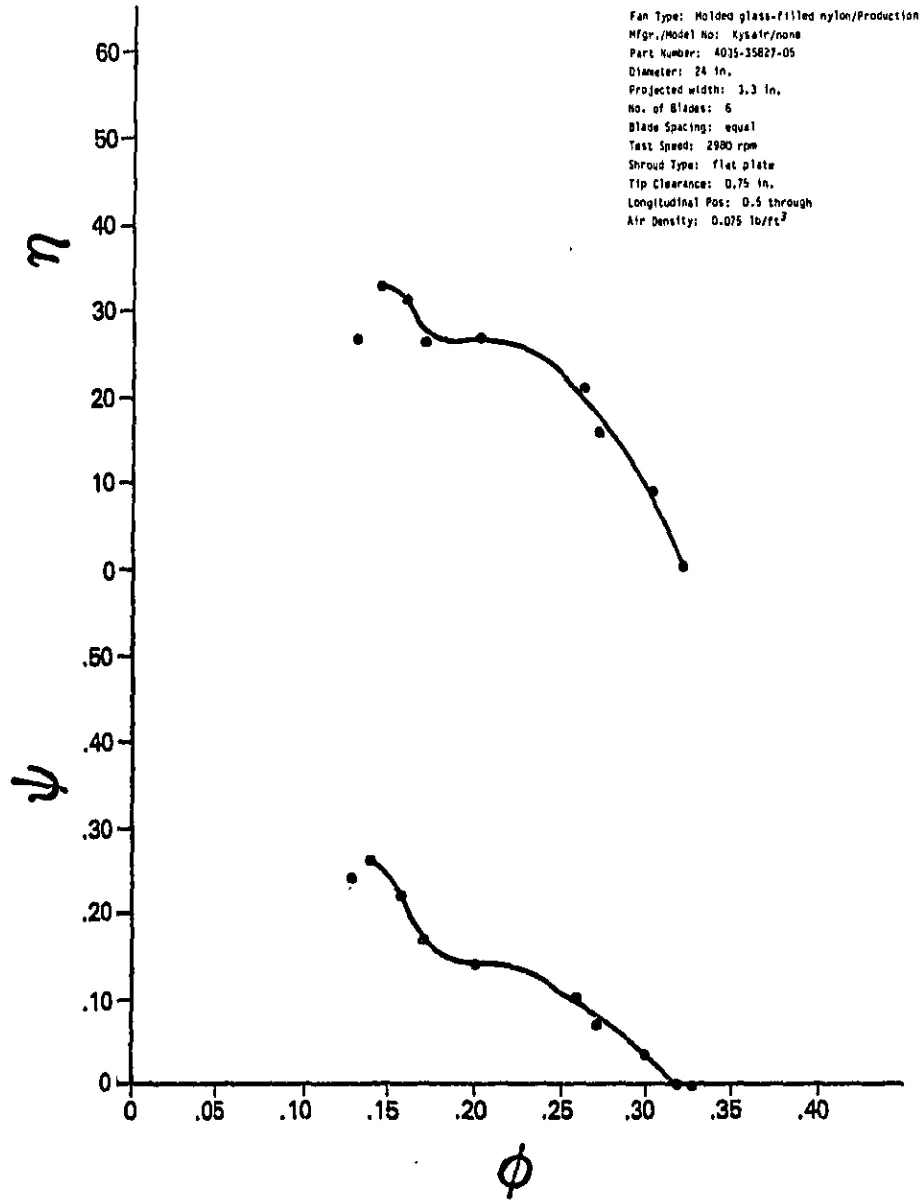

Fan Type: Sheet steel/Production
Mfgr./Model No: Hayes-Albion/2028749
Part No: GM-6-167-F-1
Diameter: 22 in.
Projected width: 2.5 in.
No. of Blades: 7
Blade spacing: equal
Test speed: 1734 rpm
Shroud Type: unknown
Tip Clearance: unknown
Longitudinal Pos: unknown
Air Density: 0.075 lb/ft³
η
60
50
40
30
20
10
0
ψ
.50
.40
.30
.20
.10
0
0
.05
.10
.15
.20
.25
.30
.35
.40
φ

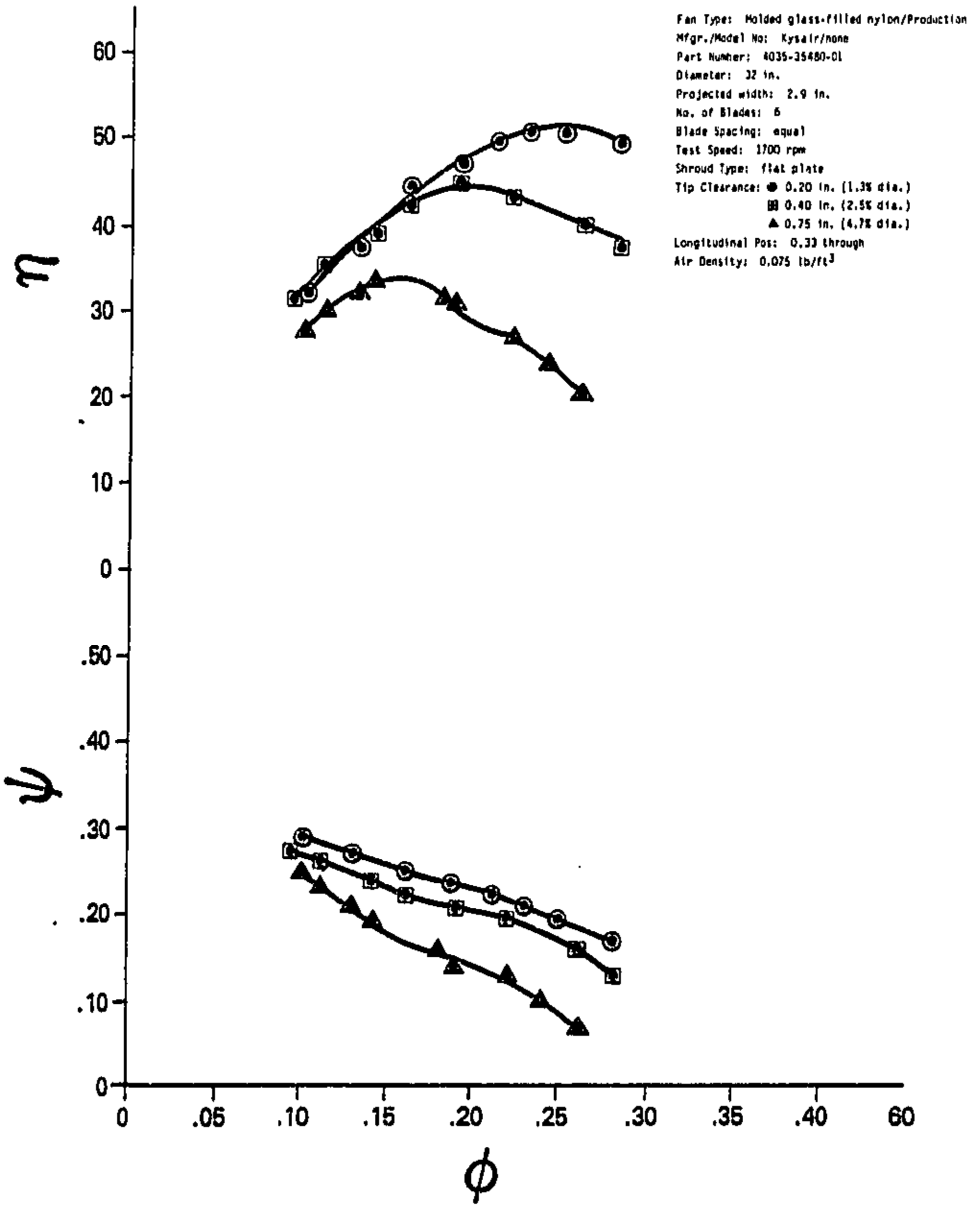


Fan Type: Molded glass-filled nylon/Production
Mfgr./Model No: Kysair/none
Part Number: 4035-35480-10
Diameter: 28 in.
Projected width: 2.8 in.
No. of Blades: 6
Blade Spacing: equal
Test Speed: 1950 rpm
Shroud Type: ● flat plate
■ ring-type
Tip Clearance: 0.75 in.
Longitudinal Pos: 0.5 through
Air Density: 0.075 lb/ft³
60
50
40
30
20
10
0
.50
.40
.30
.20
.10
0
η
ψ
0
.05
.10
.15
.20
.25
.30
.35
.40
φ

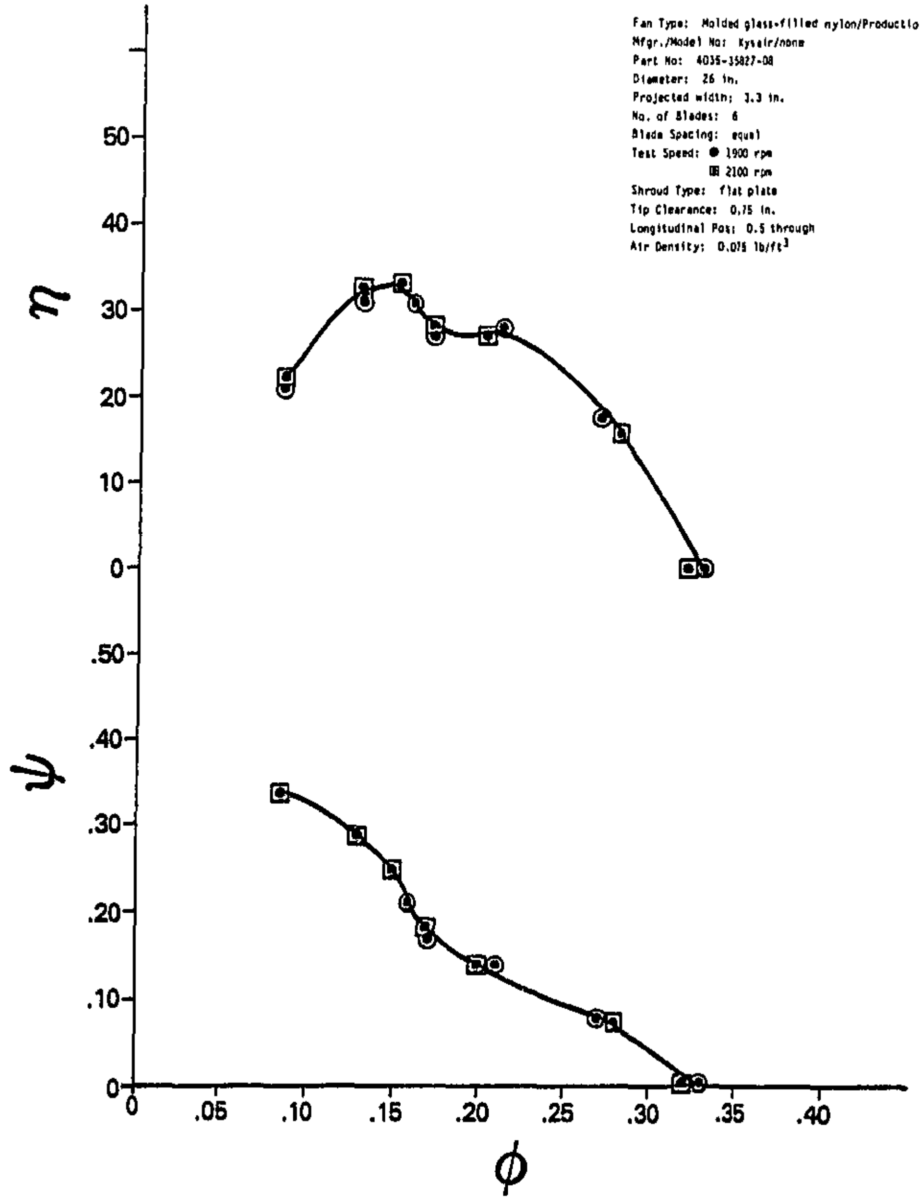


Fan Type: Molded glass-filled nylon/Production
Mfgr./Model No: Kysair/none
Part Number: 4035-35480-10
Diameter: 28 in.
Projected width: 2.8 in.
No. of Blades: 6
Blade Spacing: equal
Test Speed: 1900 rpm
Shroud Type: flat plate
Tip Clearance: ● 0.20 in. (1.4% dia.)
■ 0.40 in. (2.8% dia.)
▲ 0.75 in. (5.4% dia.)
Longitudinal Pos: 0.33 through
Air Density: 0.075 lb/ft³
η
ψ
φ
60
50
40
30
20
10
0
.50
.40
.30
.20
.10
0
.05
.10
.15
.20
.25
.30
.35
.40

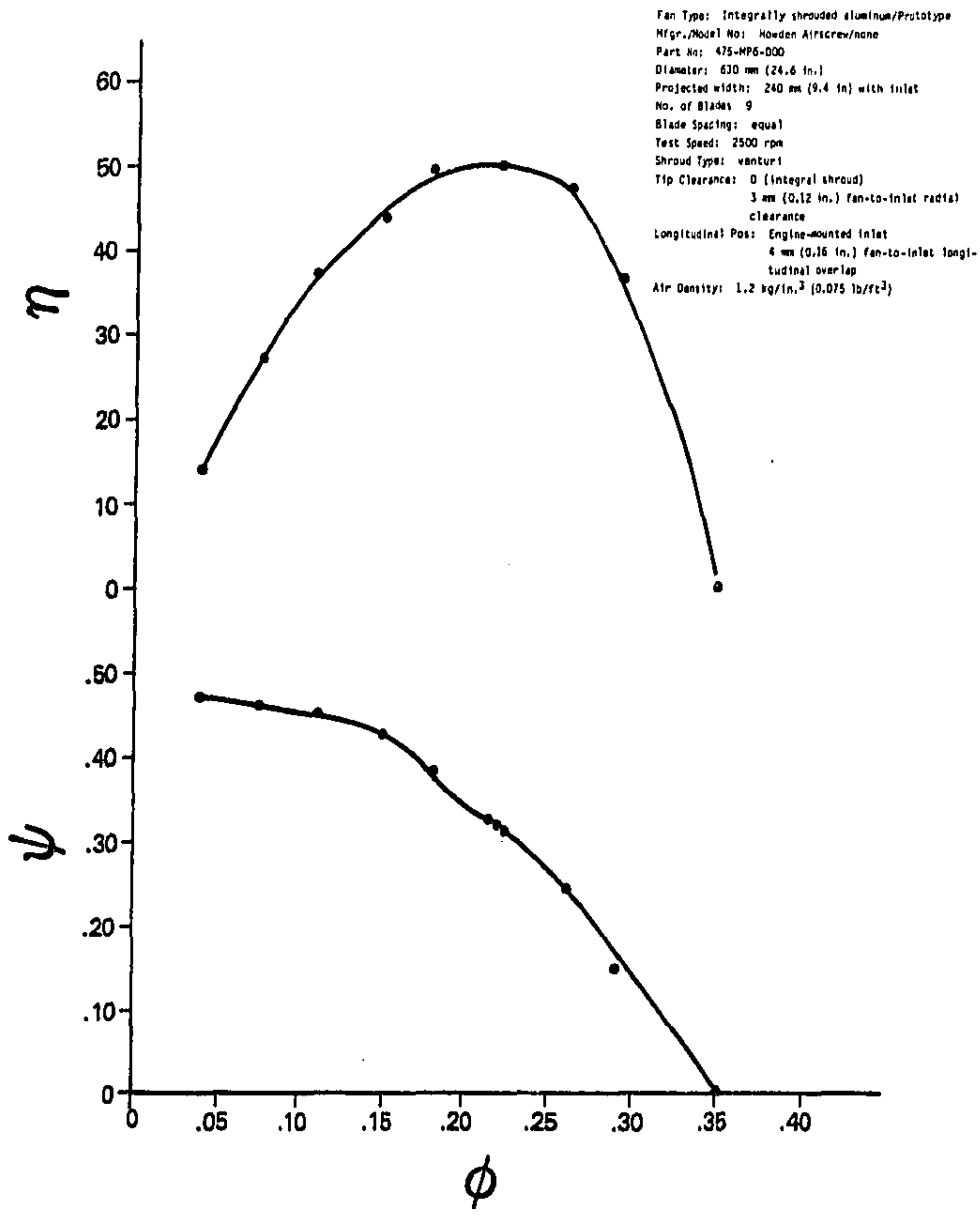

Fan Type: Molded glass-filled nylon/Production
Mfgr./Model No: Kysair/none
Part Number: 4035-35827-05
Diameter: 24 in.
Projected width: 3.3 in.
No. of Blades: 6
Blade Spacing: equal
Test Speed: 2980 rpm
Shroud Type: flat plate
Tip Clearance: 0.75 in.
Longitudinal Pos: 0.5 through
Air Density: 0.075 lb/ft³
η
60
50
40
30
20
10
0
ψ
.50
.40
.30
.20
.10
0
0
.05
.10
.15
.20
.25
.30
.35
.40
φ

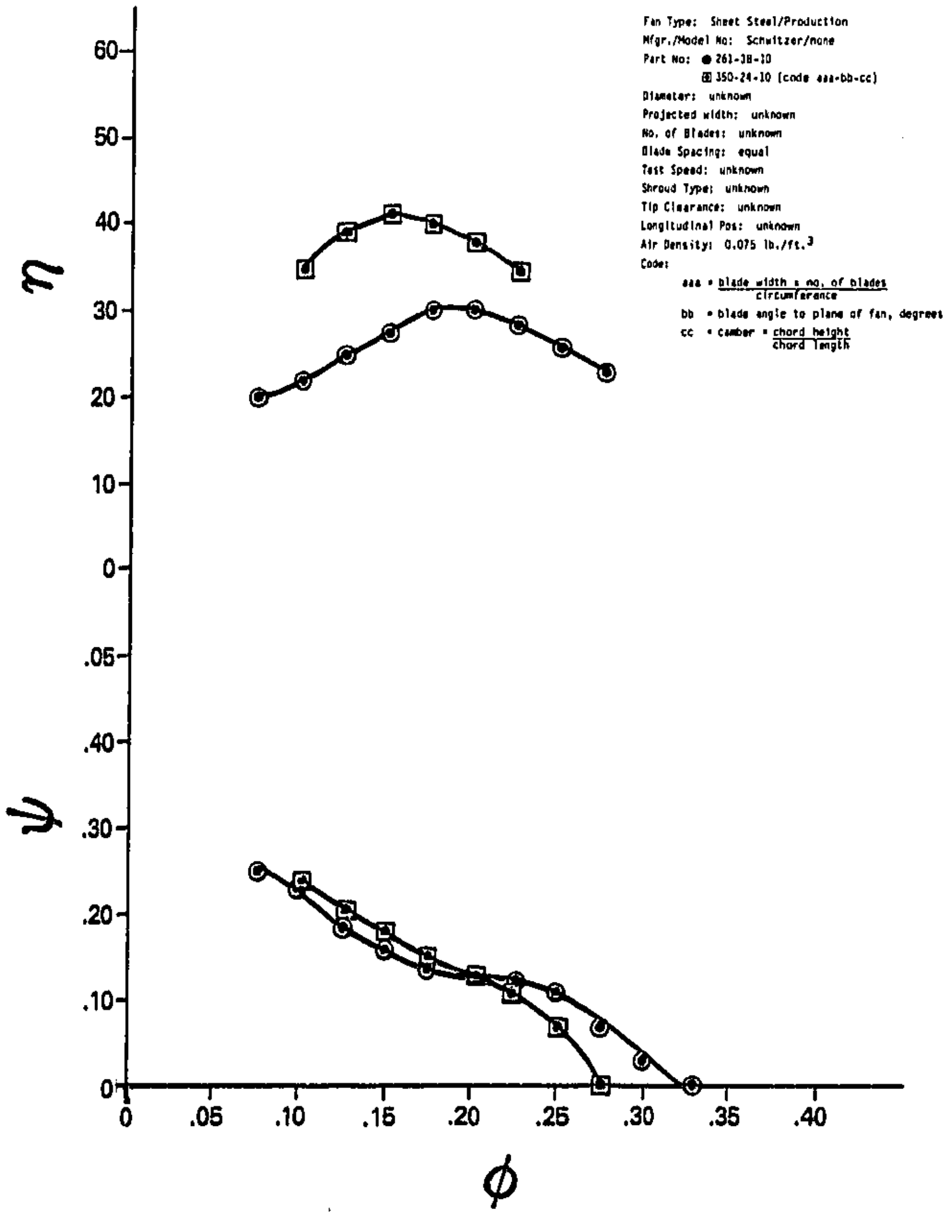


Fan Type: Molded glass-filled nylon/Production
Mfgr./Model No: Kysair/none
Part Number: 4035-35480-01
Diameter: 32 in.
Projected width: 2.9 in.
No. of Blades: 6
Blade Spacing: equal
Test Speed: 1700 rpm
Shroud Type: flat plate
Tip Clearance: ● 0.20 in. (1.3% dia.)
⊞ 0.40 in. (2.5% dia.)
▲ 0.75 in. (4.7% dia.)
Longitudinal Pos: 0.33 through
Air Density: 0.075 lb/ft³
η
60
50
40
30
20
10
0
ψ
.50
.40
.30
.20
.10
0
0
.05
.10
.15
.20
.25
.30
.35
.40
60
ϕ

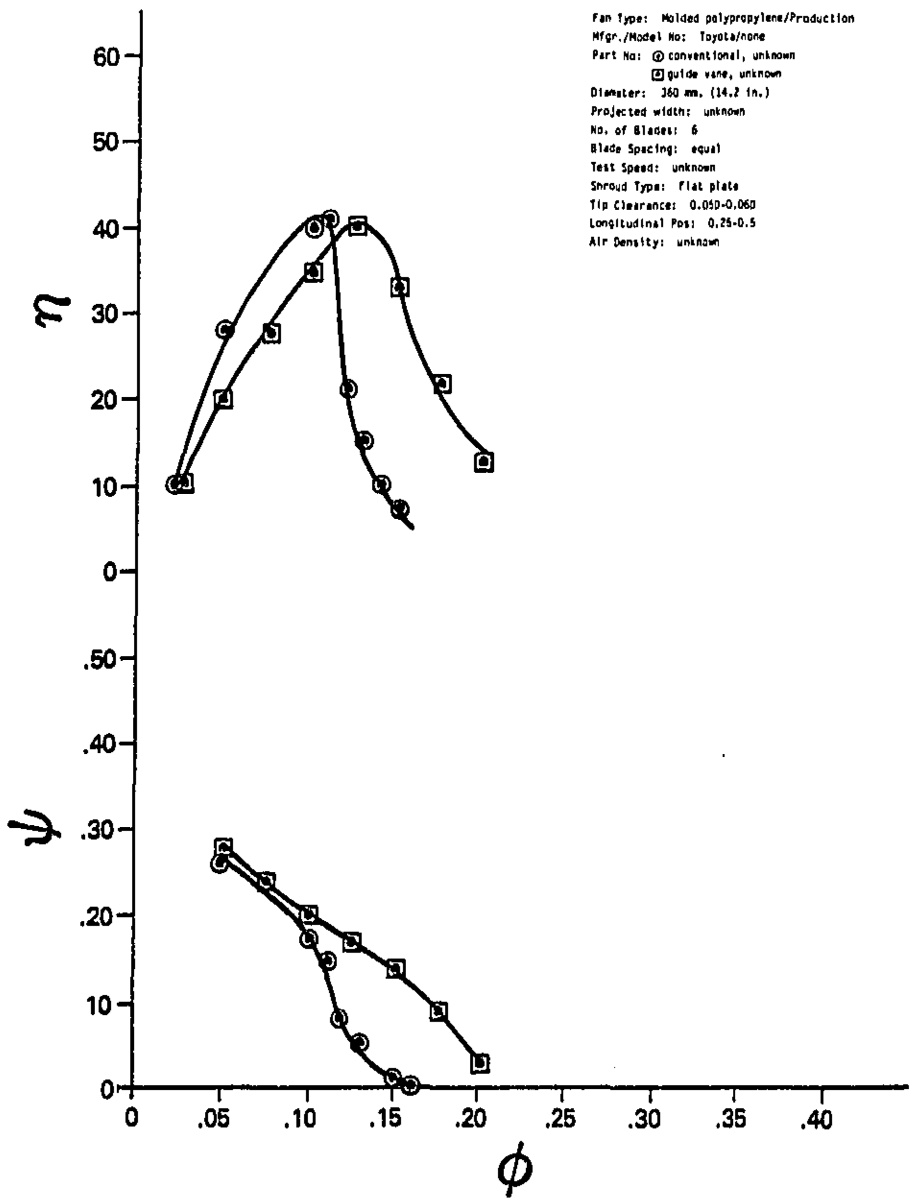


Fan Type: Molded glass-filled nylon/Productio
Mfgr./Model No: Kysair/none
Part No: 4035-35827-08
Diameter: 26 in.
Projected width: 3.3 in.
No. of Blades: 6
Blade Spacing: equal
Test Speed: ● 1900 rpm
▣ 2100 rpm
Shroud Type: flat plate
Tip Clearance: 0.75 in.
Longitudinal Pos: 0.5 through
Air Density: 0.075 lb/ft³
η
50
40
30
20
10
0
ψ
.50
.40
.30
.20
.10
0
0
.05
.10
.15
.20
.25
.30
.35
.40
φ

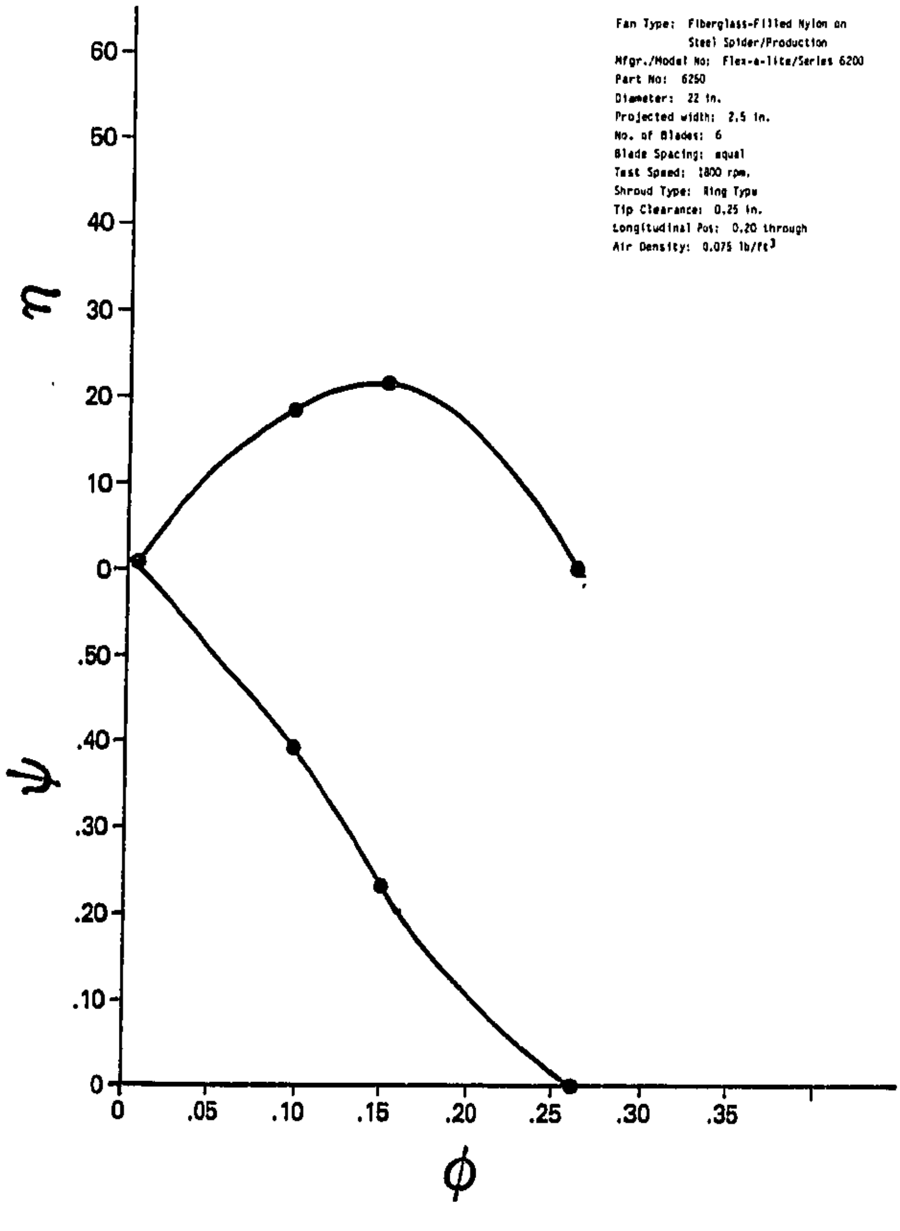

Fan Type: Integrally shrouded aluminum/Prototype
Mfgr./Model No: Howden Airscrew/none
Part No: 475-MP6-000
Diameter: 630 mm (24.6 in.)
Projected width: 240 mm (9.4 in) with inlet
No. of Blades 9
Blade Spacing: equal
Test Speed: 2500 rpm
Shroud Type: venturi
Tip Clearance: 0 (integral shroud)
3 mm (0.12 in.) fan-to-inlet radial clearance
Longitudinal Pos: Engine-mounted inlet
4 mm (0.16 in.) fan-to-inlet longitudinal overlap
Air Density: 1.2 kg/in.^{3} (0.075 lb/ft^3)
60
50
40
30
20
10
0
η
.50
.40
.30
.20
.10
0
ψ
0
.05
.10
.15
.20
.25
.30
.35
.40
ϕ



Fan Type: Sheet Steel/Production
Mfgr./Model No: Schwitzer/none
Part No: ● 261-38-10
⊡ 350-24-10 (code aaa-bb-cc)
Diameter: unknown
Projected width: unknown
No. of Blades: unknown
Blade Spacing: equal
Test Speed: unknown
Shroud Type: unknown
Tip Clearance: unknown
Longitudinal Pos: unknown
Air Density: 0.075 lb./ft.^{3}
Code:
aaa = $\frac{\text{blade width x no. of blades}}{\text{circumference}}$
bb = blade angle to plane of fan, degrees
cc = camber = $\frac{\text{chord height}}{\text{chord length}}$
η
ψ
ϕ
60
50
40
30
20
10
0
.05
.40
.30
.20
.10
0
0
.05
.10
.15
.20
.25
.30
.35
.40



Fan Type: Molded polypropylene/Production
Mfgr./Model No: Toyota/none
Part No: ⊙ conventional, unknown
⊡ guide vane, unknown
Diameter: 360 mm. (14.2 in.)
Projected width: unknown
No. of Blades: 6
Blade Spacing: equal
Test Speed: unknown
Shroud Type: Flat plate
Tip Clearance: 0.050-0.060
Longitudinal Pos: 0.25-0.5
Air Density: unknown
η
60
50
40
30
20
10
0
ψ
.50
.40
.30
.20
10
0
0
.05
.10
.15
.20
.25
.30
.35
.40
φ


Fan Type: Fiberglass-Filled Nylon on
Steel Spider/Production
Mfgr./Model No: Flex-a-lite/Series 6200
Part No: 6250
Diameter: 22 in.
Projected width: 2.5 in.
No. of Blades: 6
Blade Spacing: equal
Test Speed: 1800 rpm.
Shroud Type: Ring Type
Tip Clearance: 0.25 in.
Longitudinal Pos: 0.20 through
Air Density: 0.075 lb/ft³
η
ψ
φ
60
50
40
30
20
10
0
.50
.40
.30
.20
.10
0
0
.05
.10
.15
.20
.25
.30
.35

APPENDIX C

RADIATOR PERFORMANCE AND AIRFLOW DESIGN CALCULATIONS

A calculation of the airflow required to dissipate the heat generated by the engine is described here. In these calculations a radiator temperature drop of 10^{0} is assumed and radiator heat transfer performance is assumed to follow the performance of a geometrically similar radiator. Given the air-to-boil requirement (without the air conditioning system in operation) and an assumed maximum top tank temperature of 210^{0}F (based on GM Detroit Diesel Allison Div diesel engine practice), the design ambient of 118^{0}F is calculated. This design ambient is used for the following alterations to the baseline design. With the design ambient and the mean water temperature in the radiator determined, the air-to-water temperature potential for entering the radiator performance curves is estimated. The heat transfer performance equation relates the velocity of the air through the radiator to the heat rejected by the engine at full power. The airflow rate and heat dissipated also establish the temperature in the engine compartments. By considering the density of the air after it has passed through the radiator, a volume flow rate through the fan can also be determined. Fan performance curves for the fan installed in a baseline truck were provided by General Motors. The geometrical configuration in the truck is assumed similar to that employed during the fan test, as the fan curves can be entered to determine the pressure rise across the fan. Since the design air-to-boil is specified as occurring at 5 mph ram air speed, ram effect is neglected. One of the radiator performance curves generally furnished by manufacturers is the radiator pressure drop as a function of the air velocity through the core. This performance information generally plots as a straight line on log-log paper similar to the heat transfer performance. An equation of this line is used to find the pressure drop across the radiator core. A pressure drop through the remainder of the vehicle can be estimated by subtracting the radiator drop from the pressure rise across the fan. For the multipass radiator and thick core radiator calculations, the radiator effectiveness curves are entered to determine performance improvements. The calculations for each of the four cases analyzed are attached.

CURRENT PRACTICE

$ATB = T_B - T_{TT} + T_A$

$120 = 212 - 210 + T_A$

$T_A = 118\ °F$ design ambient

$T_{WM} = T_{TT} - \frac{\Delta T_W}{2}$

$T_{WM} = 210 - \frac{10}{2} = 205\ °F$ mean water temperature

$\Delta T_M = T_{WM} - T_{A}$

$= 205 - 118 = 87°F$ air-to-water "potential"

$\dot{q} = 22.2\ v^{0.577}$ radiator heat-transfer performance

$\frac{6160 \cdot 144 \cdot 100}{28.8 \cdot 24.1 \cdot 87} = 22.2\ v^{0.577}$

$v = 1430$ ft/min

$Q_S = v A_R = 1430 \cdot \frac{28.8 \cdot 24.1}{144} = 6890$ std. cfm.

$T_F = T_A + \frac{q}{Q_S d_S c_{PA}}$

$= 118 + \frac{6160}{6890 \cdot 0.075 \cdot 0.2428}$

$= 167\ °F$ engine compartment temperature

$Q_F = Q_R \frac{d_F}{d_S}$

$= 6890 \frac{(460+167)}{(460+60)}$

$= 8310$ cfm. volume flow at the fan.

$$\phi = \frac{Q_F}{A_F U_T} = \frac{Q_F}{\frac{\pi^2}{4}\left(\frac{D}{12}\right)^3 N}$$

$$= \frac{8310}{\frac{\pi^2}{4}\left(\frac{22}{12}\right)^3 3530}$$

$$= 0.155$$ flow coefficient

From appendix, Hayes-Albion fan
at $\phi = 0.155$, $\psi = 0.150$ pressure coefficient

$$\psi = \frac{\Delta p}{\frac{\rho U_T^2}{2}} \qquad d_F = d_s \frac{T_S}{T_F}$$

$$= 0.075 \frac{(460+60)}{(460+167)}$$

$$\Delta p_T = \frac{\rho U_T^2}{2}\psi \qquad = 0.0622 \ lb/ft^3$$ air density at fan.

$$= \frac{0.0622}{2 \cdot 32.2}\left[\pi\left(\frac{22}{12}\right)\frac{3530}{60}\right]^2 \frac{12}{62.4}\, 0.15$$

$= 3.20$ in H_2O fan pressure (ram air negligable)

$$d_R = d_S \frac{T_S}{T_R}$$

$$= 0.075 \frac{(460+60)}{(460+118)}$$

$= 0.0675$ lb/ft^3 air density entering the radiator

$\Delta p_R = 4.29 \cdot 10^{-6} v^{1.73}$ radiator pressure drop performance

$$= 4.29 \cdot 10^{-6}\left[\frac{1430 \cdot 0.075}{0.0675}\right]^{1.73}$$

$= 1.48$ in H_2O across the radiator

$\Delta p_S = 3.20 - 1.48$
$= 1.72$ in H_2O across the vehicle (less radiator)

$P = 1.575 \cdot 10^{-4} \Delta p \, Q_S$
$= 1.575 \cdot 10^{-4} \cdot 3.20 \cdot 6890$
$= 3.47$ HP air power at the fan.

INCREASED TOP TANK TEMPERATURE

assume $T_A = 118°F$
assume $\Delta T_W = 10°F$
assume $T_{TT} = 245°F$

$$T_{WM} = 245 - \frac{10}{2}$$
$$= 240$$

$$\Delta T_M = 240 - 118 = 122°F$$

$$\dot{q} = 22.2\, v^{0.577}$$
$$\frac{6180 \cdot 144 \cdot 100}{28.8 \cdot 24.1 \cdot 122} = 22.2\, v^{0.577}$$
$$v = 796 \text{ ft/min.}$$
$$Q = 796 \cdot 4.82$$
$$= 3840 \text{ std cfm.}$$

$$T_F = 118 + \frac{6160}{3840 \cdot 0.075 \cdot 0.2428}$$

$= 206°F$ temperature at the fan

$$Q_F = 3840 \left(\frac{460+206}{460+60}\right)$$

$= 4910$ cfm. flow volume at the fan

$$\Delta p_2 = \Delta p_1 \left(\frac{Q_{F2}}{Q_{F1}}\right)^2$$
$$= 3.20 \left(\frac{4910}{8310}\right)^2$$

$\Delta p_2 = 1.12$ in. H_2O

$$\Delta p_R = 4.29 \cdot 10^{-6} \left[\frac{796 \cdot 0.075}{0.0675} \right]^{1.73}$$

$= 0.537$ in H_2O

$\Delta p_S = 1.12 - 0.537$
$= 0.583$ in H_2O

$P_2 = 1.575 \cdot 10^{-4} \cdot 1.12 \cdot 3840$
$= 0.677$ HP

COUNTERFLOW RADIATOR

Assume $T_A = 118°F$
Assume $\Delta T_W = 10°F$
Assume $T_{TT} = 245°F$

$$\epsilon_2 = \frac{\Delta T_{\delta}}{Q_s d_s c_{pA} \Delta T} \qquad \Delta T = 245 - 118 = 127°F$$

$$= \frac{6160}{3840 \cdot 0.075 \cdot 0.2428 \cdot 127}$$

$= 0.694$

From Figure 3.2, Heat Exchanger Effectiveness
at $\epsilon = 0.694$ single pass
$\epsilon = 0.750$ four passes

$T_F = 118 + 0.75(127)$
$= 213°F$ temperature at the fan.

$$\Delta p = 1.12\left[\frac{.694}{.750}\frac{(460+213)}{(460+206)}\right]^2$$

$= 0.979$ in H_2O

$$P = 1.575 \times 10^{-4} \cdot 0.979 \cdot \frac{2664}{0.75}$$

$= 0.548$ HP

$$Q_s = 3840 \cdot \frac{0.694}{0.750}$$

$= 3550$ cfm

THICK CORE RADIATOR

From Figure 3.2
at $\epsilon = 0.694$, $NTU = 3.2$
For a 3.00 in. thick core

$$NTU_4 = NTU_3 \cdot \frac{T_4}{T_3}$$

$$= 3.2 \cdot \frac{3.00}{2.63}$$

$$= 3.65$$

From Figure 3.2, at $NTU = 3.65$, $\epsilon = 0.773$ for four passes.

$$Q_S = 3840 \left[\frac{.694}{.773}\right]$$

$$= 3450 \text{ cfm}$$

$$\Delta p_R = 0.537 \cdot \frac{3.00}{2.63} \left[\frac{.694}{.773} \cdot \frac{(460 + 118 + (.773 \cdot 127))}{(460 + 206)}\right]^2$$

$$= 0.509 \text{ in. } H_2O$$

$$\Delta p_S = 0.583 \left[\frac{.694\,[460 + 118 + (.773 \cdot 127)]}{.773\,(460 + 206)}\right]^2$$

$$= 0.485 \text{ in. } H_2O.$$

$$P = \frac{0.42}{0.773}(0.509 + 0.485)$$

$$= 0.540 \text{ HP} \quad (\text{optimized} / \pm 0.25 \text{ in})$$

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