AUTOMOTIVE ENGINE COOLING SYSTEM DESIGN WITH PARTICULAR REFERENCE TO AIR CIRCUIT COMPONENTS

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Introduction

Many vehicles with water cooled engines are now fitted with thermostatically-controlled electric cooling fans. The optimum design of an engine cooling system requires a knowledge of the performance characteristics of the radiator core, the cooling fan, the fan shroud and their interaction because the airside capacity is the controlling factor in heat dissipation.

The present paper describes briefly the results of two research projects recently sponsored by Davies Craig Pty Ltd and Natra Pty Ltd on automotive cooling systems. The first project was to design a high efficiency engine cooling fan driven by an electric motor, and to establish fan design techniques for future applications. The second project was to examine the various factors affecting the airside performance of an automotive engine cooling system.

Fan Design Considerations

To minimise power consumption of the fan/motor assembly, it is important to match the fan and fan motor. Figure 1 illustrates this principle. Curve 1 in Fig. 1(a) shows the fan static pressure and air flow rate relationship, where point A is the design point for the fan. For example, a typical design point could be 600 litres/s of air at 100 Pa static pressure. Curve 2 is the fan total efficiency curve, where peak fan efficiency occurs at B. If a fan is properly designed, fan design point A should be near the peak (point B) of the fan efficiency curve. Curve 3 is a plot of fan torque versus air flow rate; point C thus indicates the torque required to drive the fan motor.

Fig. 1(b) shows the method used to optimise fan and motor performance. The volume flow rate versus torque curve (curve 3) is as in Fig. 1(a), except that it has been placed with the origin of its co-ordinates in the top left hand corner. Motor efficiency is shown by curve 4 and the intercept between curves 3 and 4 is the operating point for the particular fan and motor assembly. This point should coincide with the peak efficiencies for both the fan and the motor for proper matching.

Figure 2 shows the typical non-dimensional characteristics K and Λ of a six-bladed fan of arbitrary vortex design, developed during the project for four different blade setting angles. K is defined as the ratio between the total pressure rise through the fan (P_T) and the dynamic pressure in the fan annulus ($0.5\ \rho U_A^2$), while Λ is defined as the ratio between the mean axial velocity in the fan annulus (U_A) and the circumferential tip speed of the rotor (ΩR), i.e.

$$K = \frac{P_T}{0.5\ \rho U_A^2} \quad ; \qquad \Lambda = \frac{U_A}{\Omega R}$$

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The performance of a fan for a given geometry is then given by one curve of K versus Λ and this curve applies for all rotational speeds of the fan. Apart from the advantages of condensing the data presentation for a particular fan, fans of different sizes can then be designed from this data to meet different pressure duties.

Fan Shroud/Radiator Performance under Ram Air Conditions

Ram air is that quantity of air that passes through the radiator due to the forward motion of the car and if sufficient ram air is available for engine cooling, operation of a fan is not required. An unshrouded fan would give essentially unimpeded air flow performance under high vehicle speed conditions, but it may be inadequate at low vehicle speed. Therefore, a fan shroud if used, should give minimum resistance to ram air and maximum air flow under fan-induced conditions.

There is a lack of information on shroud design in the literature, but Ref [1] gives some guidelines on shroud configurations for on-highway vehicles. Several experimental shrouds were constructed: a ring type shroud with bellmouth entry, a ring type with venturi duct, and a box type shroud with straight duct or venturi duct. The aerodynamic resistance of each of the radiator/shroud assemblies was measured in the wind tunnel, and used to estimate the effectiveness of the various shrouds under ram air conditions.

Air resistance in the cooling system is due to the front grille, the radiator and fan/shroud assembly and restrictions in the engine compartment behind the radiator and the amount of air that passes through the radiator-shroud assembly will be less than that calculated directly from the vehicle's motion. Costelli et al [2] indicate that the radiator/shroud assembly contributes about 50% of the total system resistance. Paish and Stapleford [3] have provided a simple relationship for the ratio R_v (V/V_o) between the free stream velocity V_o and radiator face velocity V in terms of the radiator/shroud pressure drop coefficient, K_p, i.e;

$$R_v = \frac{1}{1 + \frac{K_p}{4}}$$

If K_p and V_o are known, then the velocity ratio and the radiator face velocity may be calculated. It is assumed for the purposes of this report that the measured K_p values for various radiator/fan/shroud assemblies represent the total system resistances. It has also been necessary to estimate values of K_p at high velocities. Figure 3 shows the calculated air flow performance of various shrouds, under ram air conditions, with the radiator fan not operating. As expected, at a vehicle speed of 100 km/h, the full "box" shroud (S4) has an estimated air flow rate which is considerably less than the S1 bellmouth "ring" shroud.

Radiator Thermal Performance

A convenient indication of the temperature driving potential of the radiator is the "mean temperature difference (MTD)" [4]. It is defined as the difference between the average inlet and outlet water temperatures (T_{WI} and T_{WO}) and the inlet air temperature (T_{AI}), i.e.:

$$MTD = \frac{T_{WI} + T_{WO}}{2} - T_{AI}$$

Specific heat dissipation (SHD) is used to express the specific heat transfer capability of the heat exchanger. It is defined as the amount of heat dissipated by the radiator q divided by the face area of the radiator A and MTD, such that:

$$SHD = \frac{q}{A.MTD}$$

The specific heat dissipation of the test radiator (2 rows, 11 fpi, 0.24 m^2 face area) was measured with water flow rates from 0.3 to 1.2 kg/s, and radiator face velocities between 1 and 10 m/s. The results are shown in Figure 4, which shows that the specific heat dissipation is a strong function of radiator face velocity, but is relatively insensitive to water flow rate.

It has been found in this project that when air flow is induced through the radiator core by an attached fan/shroud assembly, considerable design freedom exists in placement of the fan on the core and the shape of the fan ventilated area. From a thermal point of view, there is no preferred position or shape. For a given air volume flow rate, it is also desirable to ventilate the maximum area possible with the further benefit that the lower core velocities will result in a lower core pressure drop and less fan power. A practical consideration is that the largest fan diameter is normally restricted to the minimum dimension of a rectangular radiator and to utilise more core area for fan cooling, a shroud with a rectangular to circular transition is necessary. For ram air cooling alone at high speed, it is preferable to have a fan/shroud assembly which presents the lowest pressure drop so that for a given vehicle speed, the core velocity is maximised.

The experimental work done in this project has shown that the fan design principles established previously result in a rotor design which presents a small aerodynamic resistance to ram air (as well as optimum efficiency when being driven). The resistance of the shroud (which ensures efficient fan operation) is minimised using a simple bellmouth ring shroud. Contrary to published information, venturi style shrouds did not perform as well and are necessarily more complex. It is apparent that there is insufficient knowledge of the fundamentals of shroud design, particularly those with a limited axial dimension and considerable scope exists for putting shroud design on a rational basis.

References

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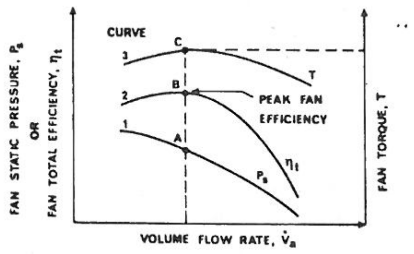
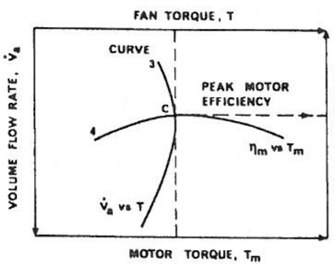


(a) TYPICAL FAN PERFORMANCE CURVES



(b) IDEALISED MATCHING OF FAN AND FAN MOTOR

Fig. 1 Matching of fan and fan motor for optimum performance.

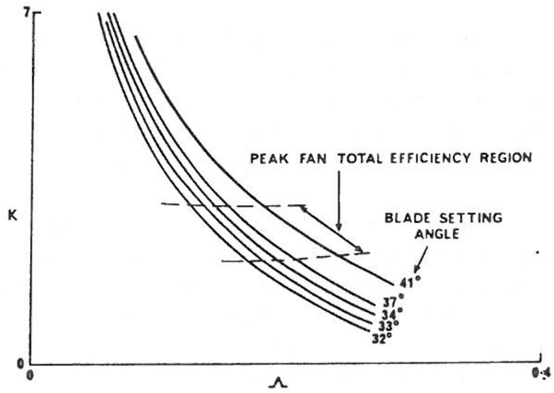


FIG. 2 TYPICAL NON-DIMENSIONAL CHARACTERISTICS OF AN ARBITRARY VORTEX FAN

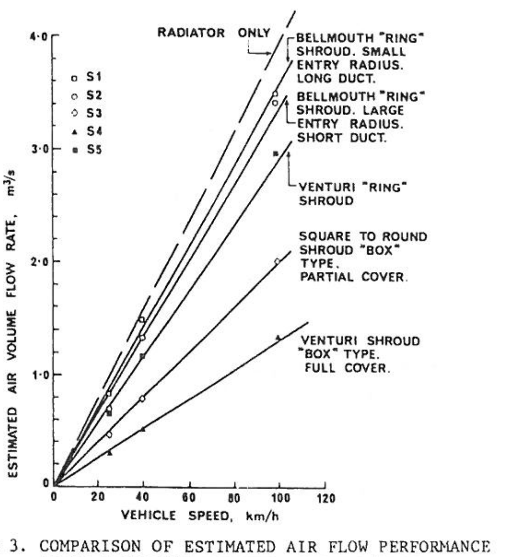


FIG 3. COMPARISON OF ESTIMATED AIR FLOW PERFORMANCE BETWEEN VARIOUS SHROUDS, UNDER RAM AIR CONDITIONS. (FAN NOT OPERATING.)

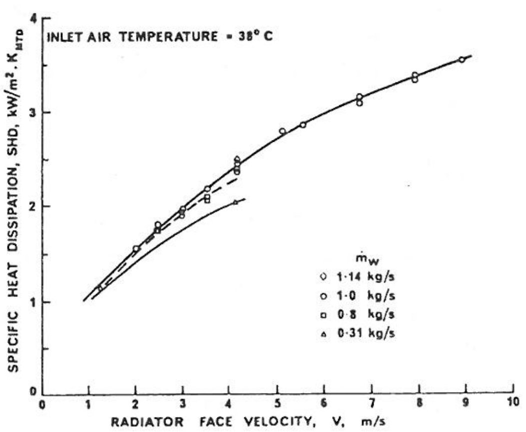


FIG. 4 SPECIFIC HEAT DISSIPATION VERSUS RADIATOR FACE VELOCITY FOR VARIOUS RADIATOR WATER FLOW RATES.