EFFECT OF RAM AIR AND FAN AIR ON AUTOMOTIVE HEAT EXCHANGER PERFORMANCE

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Introduction

An experimental investigation with a test radiator from a popular passenger vehicle was undertaken to examine the effect of a range of design variables on its thermal performance. The work reported here is part of a wider study conducted for NATRA Pty. Ltd. and Davies, Craig Pty. Ltd. by the CSIRO Division of Energy Technology on the design of optimum automotive cooling systems.

The experimental work was carried out with a NATRA radiator type 92000114 of standard copper/brass construction with a 2 row/11 f.p.i. core. A closed circuit controlled temperature wind tunnel capable of air flows up to 1000 L s^{-1} was constructed for the project. Controlled temperature hot water was supplied to the test radiator from a 37 kW electric heater. The primary measurements from which the heat transfer rate was derived were the radiator water flow rate (measured to $\pm$ 1% of rate) and radiator inlet and outlet water temperatures (measured to $\pm$ 0.2 k). The closed circuit wind tunnel incorporated an air flow measuring device with an accuracy of $\pm$ 2%.

This paper presents some of the results from experimentally determining the radiator's heat transfer capability as a function of total air flow rates and different velocity distributions across the radiator face. Ram air refers to air flow through the radiator due to the vehicle motion while fan air is the air flow due to the operation of a cooling fan. Ram air flow can occur through the fan air section but no fan induced flow occurs through the ram air section.

Measurement of radiator heat dissipation

Specific heat dissipation

The specific heat dissipation is defined as the heat dissipation rate of the radiator per unit ventilated area per degree mean temperature difference, i.e.

SHD = q/(A x MTD) where q is the heat transfer rate in kW, A is the ventilated area in m^2 and $MTD = [(\frac{T_{wo} + T_{wi}}{2}) - T_{ai}]$.

T_{wo}, T_{wi} and T_{ai} are the water outlet and inlet temperatures and air inlet temperature respectively.

A series of tests were performed in which the air velocity, water flow rate and ventilated area size, position and shape were varied [1]. The conclusions were:-

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(i) For a given radiator core the SHD is primarily determined by the ventilated area velocity.

(ii) The SHD is only slightly dependent on the water flow rate and is insensitive to the ventilated area size, position and shape.

The data from many tests for a typical water flow rate of 1 kg s^{-1} resulted in an equation of the form;

$$SHD = 1.052\ V^{0.57}$$

where V is the average velocity of the air onto the radiator face. The 0.57 exponent is close to the typical value of 0.6 for louvred fin radiator cores [2]. This supports the assumption that the radiator under test was typical of the type normally used in automobiles.

In all tests, the shape, size and position of the ventilated area were varied using blanking pieces which only allowed air to pass through the area under consideration; everywhere else over the radiator face it was essentially zero. In a practical vehicle situation, the presence of a ventilating fan/shroud assembly with an associated thermostatically controlled electric motor will only partially restrict the ram air flow through the fan region. In this case, because of the non-linear dependence of the SHD on the average velocity, it could be expected that the overall heat transfer capability of a radiator will be determined to some extent by the relative aerodynamic resistances and areas of the fan and ram air sections. The results from examining this are given in the following section.

Effect of ram air/fan air veocity ratio

For experiment and analysis, the real situation was simulated by partly blanking different areas of the test radiator with grids whose measured dimensionless pressure drops $K = 0.5\ \rho\ V^2$ were similar to the K values to be expected for typical fan/shroud combinations. The results of some of the tests are shown in Table 1. It can be seen that over the range of K's considered and the resulting velocity ratios, there is no significant effect of velocity ratio on the overall heat transfer. Measurement of the velocity distribution over the radiator was not possible for this project, so the velocity ratios shown were calculated from the following simple analysis.

Referring to Fig. 1 it can be shown that;

$$V_1 = \left(\frac{V^2}{1 + K_1}\right)^{1/2}; \qquad V_2 = \left(\frac{V^2}{1 + K_2}\right)^{1/2}; \qquad V_1/V_2 = \left(\frac{1 + K_2}{1 + K_1}\right)^{1/2} = \left(\frac{R_2}{R_1}\right).$$

With the assumption that the MTD is the same for regions 1 and 2, the total heat transferred, Q_T, is;

$$Q_T = CV^n \left(\frac{A_1}{R_1^{\,n}} + \frac{A_2}{R_2^{\,n}}\right) MTD, \qquad \text{where } SHD = CV^n$$

For a given radiator with fixed R_2, maximum cooling is obtained when $R_1 = R_2$ i.e. when the fan/shroud assembly presents no additional resistance and therefore V_1 and V_2 are a maximum for a given vehicle forward speed V. As R_2 and R_1 increase more air is "spilled" around the radiator and the mass flow rate through and heat transfer from the radiator decreases. For the in-duct case, the situation that applied in the laboratory, it can be shown that the velocity ratio is as above, but because the mass flow rate through the radiator for a constant average upstream velocity is independent of the resistance, the relationship between V_1 and V_2 is different. In this case, it can be shown that:

$$\frac{V_1}{V} = \frac{A\frac{R_2}{R_1}}{(A_1\frac{R_2}{R_1} + A_2)} \quad ; \quad \frac{V_2}{V} = \frac{A}{(A_1\frac{R_2}{R_1} + A_2)} \, .$$

If Q is the total heat dissipation rate from the radiator for the same total air flow with no velocity variation over the radiator face, than it can be shown that:

$$\frac{Q}{Q_T} = \frac{(f + a)}{f}^{1-n} r^n \left[\frac{(\frac{f}{r} + 1)^n}{(1 + \frac{r^n}{f})}\right]$$

where Q_T is the total heat transferred due to V_1 and V_2 through areas A_1 and A_2; $f = \frac{A_1}{A_2}$ and $r = \frac{R_1}{R_2}$.

This heat tranfer ratio as a function of r for various values of f is shown in Fig. 2. $\frac{Q}{Q_T}$ increases for increasing r and f but for values of r less than about 2, there is little effect of velocity distribution on the overall heat dissipation rate for the same total airflow.

For the 92000114 radiator used in this investigation, the measured K_2 (radiator core only) is about 4.2 at 4 ms^{-1} velocity. For a value of r of 2, K_1 would then be about 21. Estimated values of K_1 for the radiator together with a full rectangular-to-round shroud and installed fan (a high K combination) is about 10.6, corresponding to a value of r of only 1.5. It is apparent that in most circumstances the distribution of air flow will have little effect on the heat transferred, a conclusion that applies in either the in-duct or in-car situation. In the latter case, altering the aerodynamic resistance of the radiator assembly will change the total heat transfer rate only if the average velocity changes.

The insensitivity of the system to air flow distribution (for practical values of K) is supported by the tests in which the use of grids of different K at the same total air flow rate had no discernible effect on the heat transfer rate. Unfortunately, there are no results available to test the validity of the conclusions in a realistic situation. This is planned for the future.

One of the interesting aspects of this work is the possibility of utilising in-situ water side measurements of radiator performance in a vehicle as a way of determining the total air flow rate through the radiator, an important consideration in analysing the effect of body style changes on the under bonnet aerodynamics and hence radiator performance. All that is required is a knowledge of the SHD of the case as a function of velocity for a given water flow rate, which can be readily determined in equipment such as the closed circuit wind tunnel facility at the Division of Energy Technology.

References

1. CHAN, C.Y.L. and COOPER, P.I. - Automotive engine cooling system design with particular reference to air circuit components. Second Workshop on Wind Engineering and Industrial Aerodynamics. CSIRO Division of Energy Technology, 28-30 August, 1985.

2. STAPLEFORD, W.R. -Factors affecting the selection of fans for automotive engine cooling applications. Int. Conf. on Fan Design and Applications, Guildford, England, September 1982.

TABLE 1. Results of Thermal Tests on Test Radiator with Various Ram/Fan Air Velocity Ratios

Test No.	Air Flow Rate $L.s^{-1}$	Ram/Fan Area Ratio $\frac{A_2}{A_1}$	Estimated Ram Area Resistance K_2	Estimated Fan Area Resistance K_1	Calculated Velocity Ratio $\frac{V_2}{V_1}$	Ave Face Velocity V	Specific Heat Dissipation SHD $kW.m^{-2}.K^{-1}$ MEASURED*	$1.05\ V^{0.57}$
110A	850	1.0	4.6	11.7	1.51	3.54	2.13	2.16
111A	850	1.0	4.6	9.2	1.35	3.54	2.20	2.16
104A	850	0.25	4.6	11.7	1.51	3.54	2.17	2.16
105A	850	0.25	4.6	9.2	1.35	3.54	2.19	2.16
110B	720	1.0	4.9	12.2	1.50	3.02	1.97	1.97
111B	720	1.0	4.9	9.7	1.35	3.02	1.97	1.97
104B	720	0.25	4.9	12.2	1.50	3.02	1.99	1.97
105B	720	0.25	4.9	9.7	1.35	3.02	1.99	1.97

*At test conditions: P_{IN} = 20.5 kW, $\dot{m}_w$ = 1.0 kg/s, T_{AI} = 38.0°C

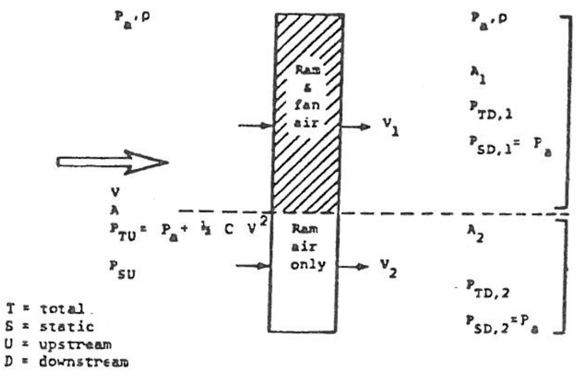

Fig. 1.

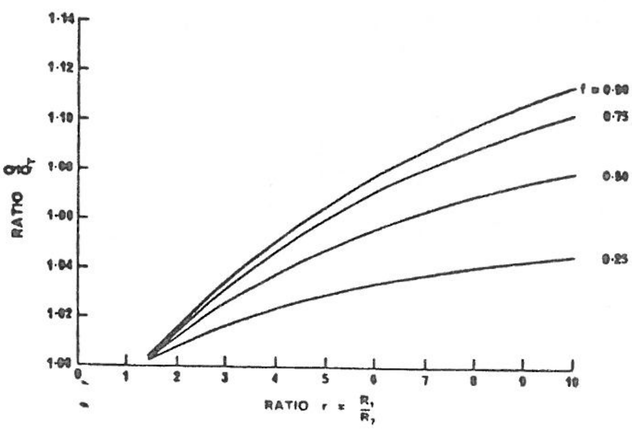

Fig. 2.