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# Airflow Distribution through the Radiator of a Typical Australian Passenger Car

**H. Jama, S. Watkins, C. Dixon and E. Ng**

School of Aerospace, Mechanical & Manufacturing Engineering
RMIT University, VIC, 3081 AUSTRALIA

## Abstract

The airflow distribution and non-uniformity across the radiator of a full-size, Australian made Ford falcon was tested at the RMIT Industrial Wind Tunnel. The cooling air intakes of the vehicle were shielded by a quarter, one-half and three-quarters and fully blocked. Four different possibilities of shielding methods were investigated with the aim of determining the best method of shielding to be employed. Results from these tests have shown the optimum method for shielding the front-end of the vehicle in terms of airflow uniformity to b the horizontal method followed by the vertical method. These shielding methods also produced the higher average airflow velocity across the radiator which is analogous to better cooling.

## Introduction

The aerodynamic drag coefficient of most passenger vehicles is now around 0.3. The use of body shape and external detail optimisation has led to this low drag coefficient. The remaining areas of exploration and optimisation are the underbody and cooling system. The cooling system of a typical passenger vehicle contributes between 6 and 10 percent to the overall drag of the vehicle [5]. Furthermore engine cooling systems are designed to meet two rare and extreme conditions. Firstly, driving at maximum speed and secondly driving up a specified gradient at full throttle while towing a trailer of maximum permitted mass. At all times, in fact the majority of the time, the cooling system operates below maximum capacity while incurring a drag penalty [4].

A system that matches the required cooling airflow through the radiator to the operating condition of the vehicle has been recently proposed. It involves the varying of the cooling air intakes by the use of servomotors and controllers such that in the majority of the cases described above, the cooling air intake areas are less than the baseline condition.

It is generally known that the velocity of the airflow through the radiator is a function of the vehicle speed and the "heat transferred by a radiator is a function of the airflow rate across the radiator" [6]. However, the non-uniformity is another factor to determine in engine cooling. Others like Chiou [2] have suggested that radiator heat transfer effectiveness "deteriorates due to two-dimensional flow non-uniformity on both the air and coolant sides". Therefore an experimental program was designed that investigated methods of reducing the airflow through the radiator and engine compartment by shielding the front-end of a passenger vehicle. The velocity distributions as well as the non-uniformity of the cooling airflow across the radiator were also measured.

## Test Vehicle and Experimental Set up

The vehicle used in this investigation was an Australian made Ford Falcon AU. This vehicle is a middle range family vehicle that weights approximately 1550kg. It comes with a four-speed automatic transmission as standard equipment. The air conditioning and engine-cooling components that are pertinent to this investigation consisted of a condenser, fitted in front of the radiator, and a mechanically driven centrifugal water pump, dual electric fans with shroud combination and an airdam. The airdam aids in engine cooling by creating a favourable pressure gradient for the cooling airflow. The front-end cooling air intakes consist of a decorative grille and a lower intake area.

To study the variable front-end geometry, four front-end shielding methods were employed. In each of the methods the front-end cooling air intakes were shielded by an area of ¼, ½, ¾ and totally shut. These shielding methods employed were vertical, horizontal, side-to-side and side-to-centre as illustrated in Figure 1 to Figure 4. In the vertical shielding method evenly distributed vertical strips were used. The underlying principle for this type of shielding is that many vehicles already have vertical strips as part of their decorative grille and lower cooling intakes. To implement this type of shielding one could envisage plates sliding behind each other that would change the area of the cooling air intakes.

However, other vehicles exhibit the opposite by having decorative grilles and lower cooling air intake openings that are covered by horizontally placed strips. The analogous method of shielding that is envisaged is that of having horizontal plates sliding behind each other. The other configuration investigated was closing the intake opening from one side to the other. This configuration was chosen as it can be applied to small vehicles that have very small radiators and even smaller condensers placed in front of these radiators. Instead of this normal arrangement, it is envisaged that the condenser could be placed besides the radiator. Then, in periods of extended non-operation of the condenser like winter, one could entirely block off the condenser side to the cooling airflow. The last option considered is to symmetrically shield both the grille and lower cooling air intake from both sides to the centre. This last method was used to investigate the interaction between external and internal flows.

Figure 1: Vertical method (1/2 cooling air intake area shielded).

Figure 2: Horizontal method (1/2 cooling air intake area shielded).

Figure 3: Side-to-side method (1/2 cooling air intake area shielded).

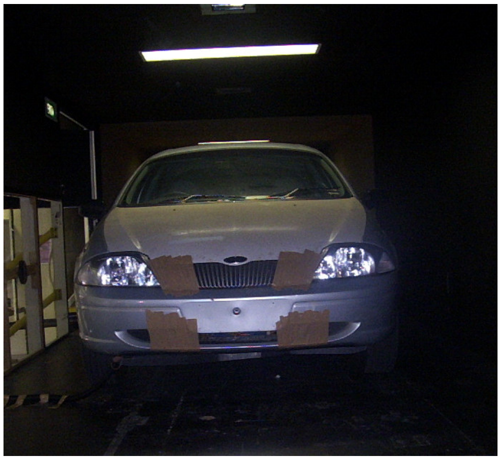

Figure 4: Side-to-centre method (1/2 cooling air intake area shielded).

## Test Facilities, Equipment and Parameters

The vehicle was tested at the RMIT Industrial Wind Tunnel. This tunnel has a 3m wide, 2m high and 9m long working test section and a 2:1 contraction ratio. It has been described in detail in the works of Watkins [10] and Ng et al. [8]. This tunnel has a blockage ratio of 0.35 for a full-size Australian passenger vehicle. Although this blockage is very high for aerodynamic testing, it was shown by Ng et al. [8] that it can be used adequately for evaluating the cooling performance of a passenger vehicle. Others [3, 7] had earlier shown that it could be used to evaluate the cooling performance of the front section of a passenger vehicle.

The choice of equipment available to quantify airflow distribution across the radiator was limited by the cost and complexities involved. A pressure based technique established by Ng et al. [9] to quantify airflow distribution across an automotive radiator was used in this investigation. The equipment consisted of 24 pairs of hypodermic tubes inserted into the radiator and condenser assembly, a pressure measuring unit, a PC computer and the associated software. The technique is relatively low cost, robust and suitable for measuring complex airflow. For a detailed description please refer to Ng et al. [9].

The airflow non-uniformity index (*i) was* used to quantify the non-homogeneity of the airflow. If the radiator is segmented into a finite number *n* of areas elements. The non-uniformity was then defined by Lee and Hong [6] as follows;

$$i = \frac{1}{n}\sum_{k=0}^{n} \frac{|\, \dot{m}_K \dfrac{A_R}{A_K} - \dot{m}_{tot} \,|}{\dot{m}_{tot}} \tag{1}$$

Where;
$A_K$ = size of one area section,
$A_R$ = Area of radiator matrix,
$\dot{m}_K$ = mass flow rate through one section and
$\dot{m}_{tot}$ = Total mass airflow.

When the sectioned areas are of equal size then the non-uniformity simplifies to;

$$i = \frac{1}{n}\sum_{k=0}^{n} \frac{|V_K - V_{av}|}{V_{av}} \tag{2}$$

Where;
$V_K$= the airflow velocity of one section and
$V_{av}$=the average airflow velocity across the radiator.

## Results and Discussion

Once the results were obtained both contour plots of the velocity distributions as well as the average velocity of the airflow and the airflow non-uniformity index were analysed. The velocity distributions were plotted using a commercial software called Tecplot. The data points were extrapolated to the full area of the radiator using the Kriging method of spatial statistical interpolation available in the software. The Kriging method uses a general trend and a specified number of points to weigh from and adds a random noise component to find the value of the point being interpolated [1].

Figure 5 in the following pages shows the airflow distribution at the radiator as being highly non-uniform. This is for the baseline configuration for the simulated road speed of 100km/h. It can be seen that the top and bottom sections show higher airflow velocities. This was due to the fact that these regions are behind the flow inlets and is consistent with the physical location of the decorative grille panel and lower intake area. The middle section exhibits very low velocities due to being located directly behind the bumper bar. It was found that the airflow velocities ranges between 2.6m/s and 7.5m/s compared to a free-stream velocity of 22.8m/s. These velocities are the average for each location recorded over a 30 second period. As the area of the front-end cooling air intakes were shielded the distribution of the airflow across the radiator changed.

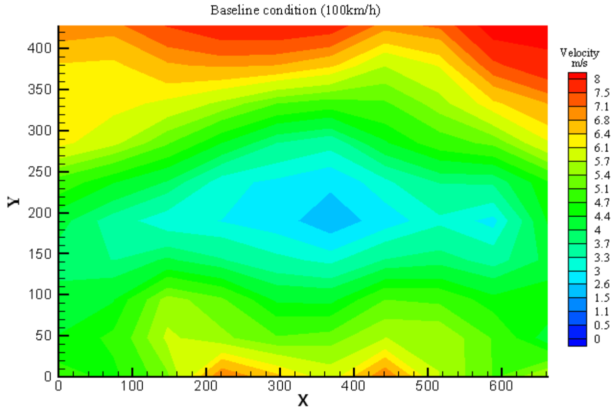

Figure 5: Baseline vehicle airflow distributions (100km/h road speed).

Presented in Figure 6 to Figure 9 are the airflow distributions across the radiator when one-half of the cooling intake areas are shielded. These velocity contour plots are for the simulated road speed of 100km/h. Due to the blockage of the tunnel a lower air speed of 82km/h was used which was confirmed by previous on-road tests to be equal to 100km/h road speed [8]. Figure 6 and 7 show a comparatively uniform airflow than Figure 8 an9. The vertical and horizontal shielding methods show a higher degree of airflow uniformity and are comparable to the baseline configuration but exhibit some dead zones. These dead zones are mainly the signature of the engine block.

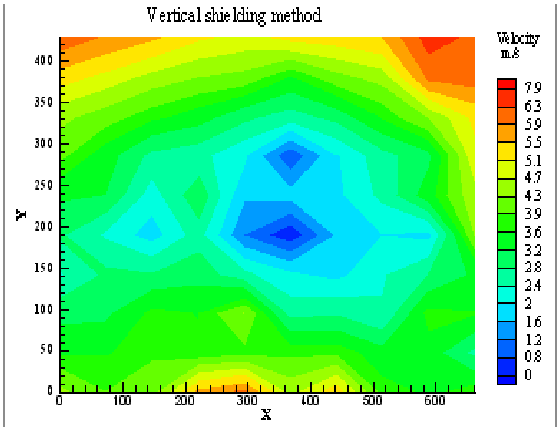

Figure 6: Airflow velocity distribution ½ intake area shielded (Vertical).

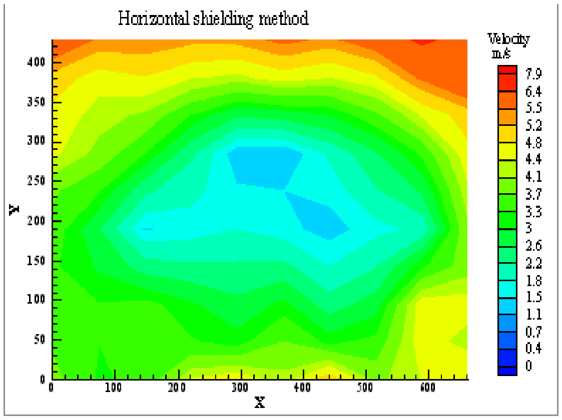

Figure 7: Airflow velocity distribution ½ intake area shielded (Horizontal).

However, the side-to-side and side-to-centre shielding methods show a high level of non-uniformity. Figure 8 shows the region directly behind the shield to be a dead zone with little or no flow. Similarly, Figure 9 shows the side-to-side shielding method produces areas with no or little airflow directly behind the areas where the shields were employed. It should also be noted for each case the average cooling airflow velocity was 2.6m/s and 3.2m/s respectively.

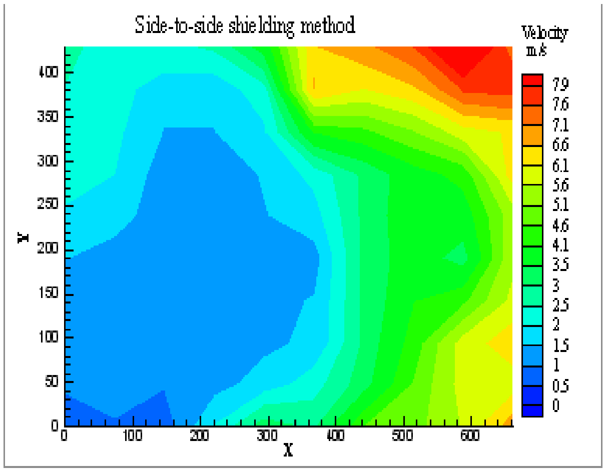

Figure 8: Airflow velocity distribution ½ intake area shielded (Side-to-side).

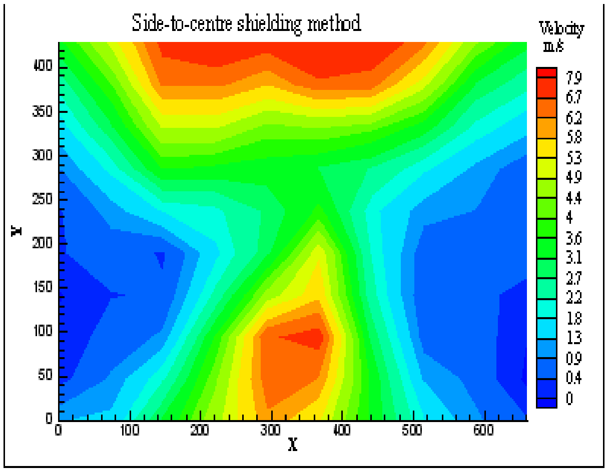

Figure 9: Airflow velocity distribution ½ intake area shielded (Side-to-centre).

It is important to note that the errors arising from the pressure – based system greatly increased as the velocities being measured reduce. It was found that as more areas of the front-end intakes were shielded, the average airflow velocities across the radiator decreased and the errors involved increased. Using a vane anemometer, it was fond that the pressure-based technique resulted in an error of about 5% when the velocities being measured were higher than 2m/s. This is simply as a result of the measured pressure being reduced significantly as the front-end intake area is reduced. As with any pressure-based measuring systems as the quantity being measured reduces significantly, the accuracy of the system diminishes.

Significant differences between the methods were found when plotting the velocity contours for the different configurations. However, this was not replicated when the average cooling air velocity through the radiator were analysed. It should be noted that the cooling fans were not operating throughout the tests but were in place together with the fan shroud. Figure 9 shows the vertical shielding method yields the lowest velocity while the horizontal shielding method consistently produced the higher velocities. This is explained by the fact that the vertical strips used in shielding the front-end allow the airflow, while passing over the upper and lower intake area, to stay attached as it travels over the bonnet. This diverts the airflow from entering the engine bay and results in less internal flow and increased external flow. In a contrary fashion, it was observed (with the aid of wool tufts) that the horizontal strips allow the airflow to split and be ingested into the cooling system. This explains the higher velocities recorded for the horizontal shielding method. It was surprising to observe the side-to-side shielding method leads to higher average core velocities that the side-to-centre method.

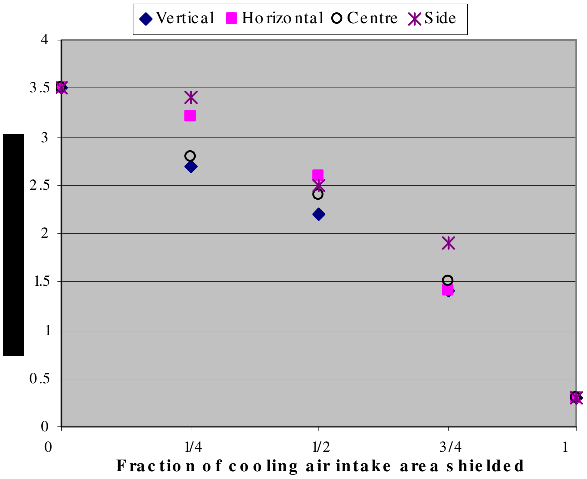

Figure 9: Average airflow velocities behind the radiator for the simulated road speed of 100km/h.

The cooling airflow non-uniformity index was also calculated as the front-end intake areas were shielded. AS defined before, airflow is highly non-uniform when *i* is low and verse versa. The vehicle at the baseline condition had a non-uniformity index of about 0.3. This rises to more than 0.8 when the side-to-centre shielding method is employed and ¾ of the intake area is shielded. It can be seen in Figure 10 that the vertical followed by the horizontal shielding method contribute to a more uniform and hence better airflow distribution across the radiator. In contrast, the side-to-centre shielding method performs poorly in terms of airflow velocity distribution across the radiator.

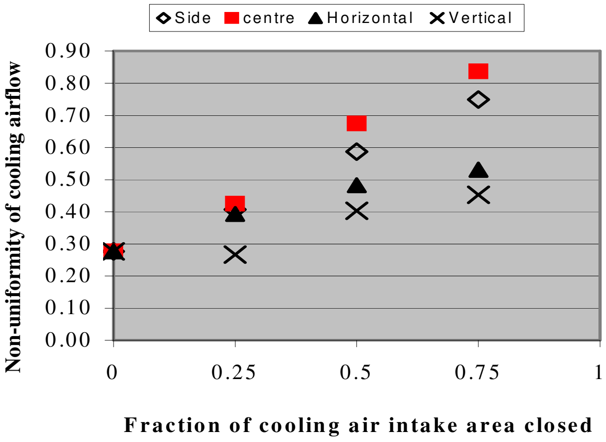

Figure 10: Fraction of cooling intake area shielded vs airflow non-uniformity index.

## Conclusions

The results showed that the best method to shield the front-end of a passenger vehicle would be to employ a horizontal method. This shielding method produced the more uniform cooling airflow distribution compared to the other methods. By extension it should also produce the least reduction in cooling capacity for a given intake area.

It was found and was expected that the non-uniformity index increased significantly as the front-end air intake area was shielded. This increase in the non-uniformity index is expected to correlate with reduced cooling capacity of the vehicle.

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