Enhancing the Thermal Performance of Vehicle Cooling Modules Using Diffusers

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Abstract. *Uniform airflow distribution is desired over the area of vehicle cooling modules. For this purpose, this paper investigates a novel design where a diffuser is positioned between the vehicle's front-end opening and the first exchanger in the cooling module. The reasoning behind this is that with the diffuser, the magnitude of the airflow velocity is decreased but becomes more uniform as it will be spread over a larger part of the exchanger area, thus enhancing the uniformity of distribution which leads to performance enhancement. Performance evaluation is provided by an in-house computational code, through which a parametric analysis is conducted based on system parameters, such as air inlet opening area to exchanger area ratio, distance of the first exchanger from the inlet opening, diffuser angle and the mass flow rates of water and air. It was shown that, depending on system configuration, an enhancement of up to 67% can be obtained due to the diffuser integration.*

1. Introduction

Heat exchangers thermal performance is affected by several operational and design parameters, the properties of the exchanging flows are some of the most important factors, especially the flow distribution over the area of exchange. In liquid-air heat exchangers, several studies investigate the effects of airflow velocity nonuniformity on the thermal performance [1–6]. Khaled et al. [7-8] considered a tube-and-fin heat exchanger which is often utilised in automotive applications, wherein the heat exchanger is almost always subjected to a nonuniform airflow velocity distribution, it was shown that for a severely maldistributed airflow velocity i.e., standard deviation and a mean value of the distribution being equal, a performance reduction of up to 35% is induced. Consequently, a nearly uniform airflow velocity distribution is desired in order to enhance the thermal performance. This work aims to present a method by which a uniform airflow velocity distribution is obtained in order to optimize the air-liquid heat exchanger thermal performance. The method consists of introducing a diffuser between the vehicle's front air opening and the heat exchanger immediately at its downstream (Figure 1).

Upon introducing the diffuser, the velocity magnitude decreases while the area covered by the flow is increased i.e., as the flow rate is surely unchanged (same mean velocity), the standard deviation of the velocity distribution is significantly reduced, in other words, the nonuniformity is decreased. This decrease in airflow velocity nonuniformity upstream the heat exchanger will result in an enhanced thermal performance.

The performance enhancement is examined thoroughly by investigating different parameters that relate to the diffuser design and the system operation. Such as air and water flowrates, ratio of the air inlet opening are to the heat exchanger area, diffuser length (distance between the front end opening and the first exchanger), and the diffuser angle. The calculations are facilitated by an in-house computational code. The code is briefly explained in section 2, section 3 provides the parametric analysis, conclusions are drawn in section 4.

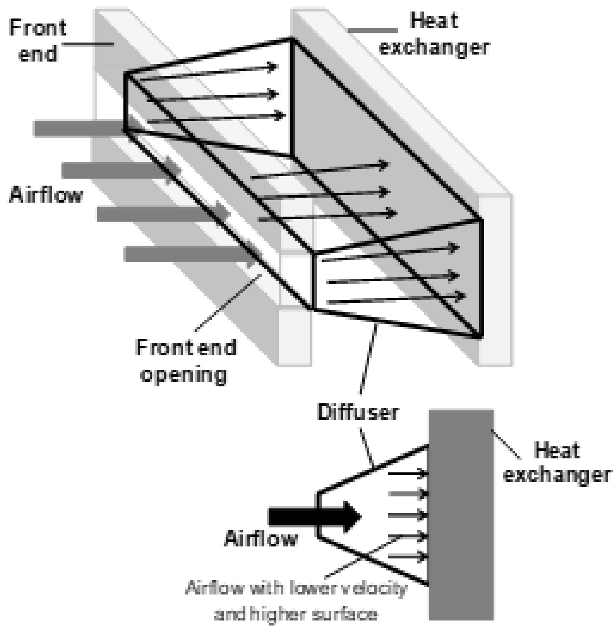

Fig. 1. New suggested design

2. IN-HOUSE COMPUTATIONAL CODE

The code's concept and principles are detailed in [7–9]. A brief description is provided here.

The method is based on discretizing the heat exchanger domain into smaller size subdomains, referred to as Representative Cells (RC), this allows the representation of the nonuniformities experienced by the global domain (Figure 2)

As the domain is divided, the heat transfer problem is also divided such that local energy balance problems are solved at each RC, starting with the cells at the boundary of known inlet temperatures, outlet solutions of each cell are used as inlet conditions of the neighboring one, to eventually obtain the temperature distribution over the global domain, allowing also the determination of the exchanger's outlet temperature by the values found at the suitable boundary RCs. Results obtained by the code were found to be in good agreement with experimental ones [8].

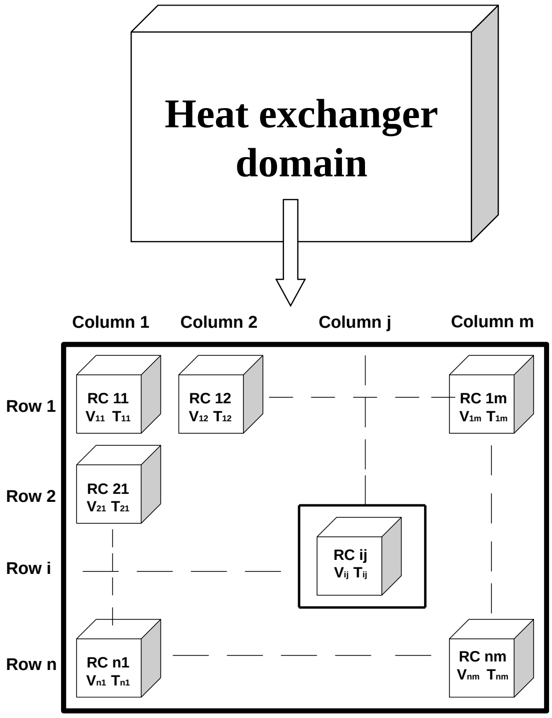

Fig. 2. Division of the heat exchanger domain into sub-domains (representative cells)

3. RESULTS, OBERVATION, AND DISCUSSION

In this section, a parametric analysis is performed, through the developed computational code, on the thermal performance enhancement of a heat exchanger which is downstream of a diffuser. The system setup and the tested parameters are shown in Figure 3.

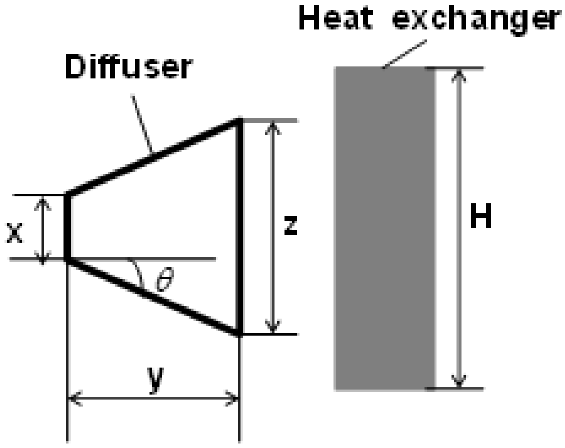

Fig. 3. Schematic including the parameters to be tested

As shown in Figure 3, x represents the diffuser's inlet cross sectional height which is taken equal to the vehicle's front end air opening, y is the diffuser's length which is effectively the distance between the front end opening and the first exchanger in the flow direction, z is the diffuser's outlet cross sectional height, θ is the diffuser angle and H is the height of the exchanger.

The diffuser's inlet and outlet cross sectional heights (x and z) are related by:

$$z = x + 2y \tan\theta \quad (1)$$

The diffuser parameters are normalized by the exchanger's height as follows:

$$X = \frac{x}{H} \quad (2)$$

$$Y = \frac{y}{H} \quad (3)$$

$$Z = \frac{z}{H} \quad (4)$$

Due to the fact that the diffuser's outlet section height must not be larger the exchanger's height, the diffuser's length y will have an upper limit that cannot be exceeded.

Computations are made with fixed operational parameters being, a water flowrate of 6000 L/h with a 90 °C inlet temperature, and an air velocity of 7 m/s with a 20 °C inlet temperature. The diffuser's angle is set at 30° and four calculation runs are made for an X value of 0.2, 0.3, 0.4, and 0.5, where Y is to be varied from 0 (no diffuser case) to its maximum value which dictated by the aforementioned condition of z and H. Figure 4 shows the evolution of the exchanger's performance and its enhancement percentage (relative to the no diffuser case, Y=0) versus the variation of Y for each X value considered.

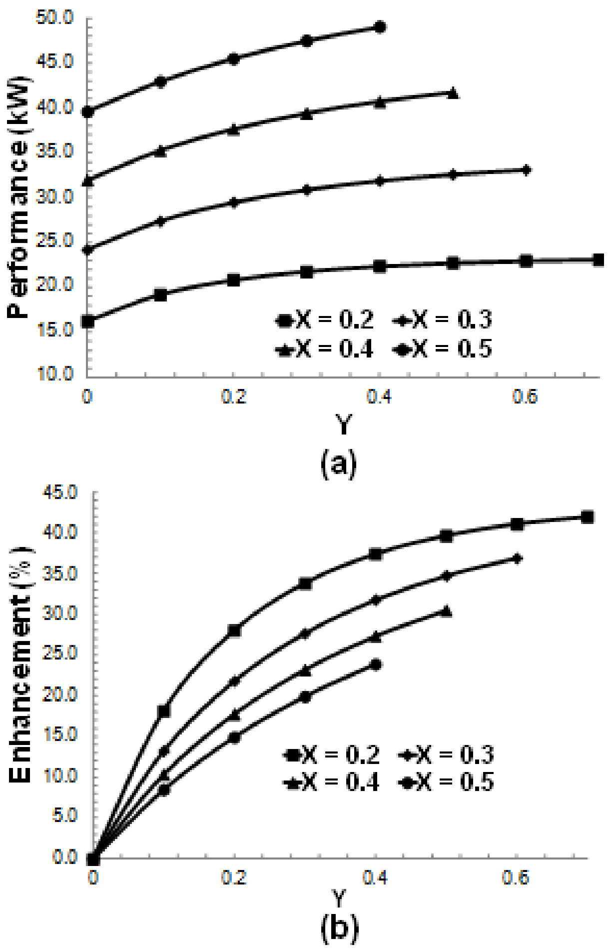

Fig. 4. Variation of the (a) HX thermal performance and (b) enhancement in function of X and Y

Observing Figure 4(a), it is noticed that the exchanger's thermal performance increases with the increase of both X and Y, however, Figure 4(b) shows that the performance enhancement increases with the increase of Y but decreases with the increase of X.

To further elaborate, in Figure 4(a), when X = 0.2 (which corresponds to an air opening area that is 20% of the exchanger area) the thermal performance increases from 16.3 to 23.1 kW as Y is increased from 0 to its maximum possible value of 0.7. While for X = 0.5, and as Y increases from 0 to its maximum which in this case is 0.4, the performance increases from 39.6 to 79 kW.

As for the enhancement, from Figure 4(b), when X = 0.2 and Y is varied from 0 to its maximum of 0.7, the enhancement increases from 0 to 41.9%, while the enhancement moves from 0 to only 23.8% for the case of X = 0.3 and Y increasing from 0 to the maximum 0.4.

4. CLOSURE

This paper introduced the novel concept of adding a diffuser between the front end opening of a vehicle and the cooling module, for the purpose of obtaining a more uniform distribution of airflow velocity over the heat exchanger area in order to obtain an enhanced performance.

Computational results reveal a possible performance enhancement of up to 67% in the presence of a diffuser.

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