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HEAT TRANSFER CORRELATION FOR AN ENGINE COOLING RADIATOR

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Abstract. *In internal combustion engines, fuel energy is partially converted to work and heat needs to be rejected to the environment. Radiators are compact heat exchangers designed to reject this heat in automotive applications. This work aimed to study the thermal behavior of a small scale radiator in steady state. An experimental bench was assembled with the radiator fit inside a wind tunnel. Air flow in the tunnel was adjusted by a fan and hot water was used as a heat source. It was electrically heated in a tank, pumped to the radiator and then went back to the tank, in a closed circuit. The experiments were carried out with constant water flow, whereas air velocity was varied with the fan. The heating power was also varied in function of the number of electric heaters turned on inside the water tank. Water inlet and outlet temperatures were measured with thermocouples and air temperature was evaluated by using a grid of thermocouples in the wind tunnel. The overall heat transfer coefficient indicated the important influence of frontal air velocity on heat exchange. A dimensionless correlation for air heat transfer coefficient, based on Nusselt number, was suggested.*

Keywords: *radiator, heat exchanger, heat transfer*

1. INTRODUCTION

The field of automotive engine is highly assisted by heat exchangers. The thermal energy from internal combustion that was not converted to work must be reject to the external environment to avoid engine overheating. Due to this need and to conciliate an efficient heat exchange with vehicle design, reduction of weight and size and lowest possible cost, compact heat exchangers, known as radiators, are used.

Basically, the radiator is a cross flow compact heat exchanger that is composed by flat tubes with louvered fins between them. After being heated by the engine, the fluid is pumped to the interior of the flat tubes, exchanging heat with the air from the environment, that passes through fins, as the vehicle moves. The fins are welded transversely to these tubes to enhance the heat transfer.

It becomes extreme important to know how radiators are affected and what project is necessary to meet the cooling capability required. Studies of radiators performance are found in literature (Ng, 2002). The author evaluated the tests constraints and accuracy of the cooling performance.

Experimental test via wind tunnel is a typical method to study heat exchangers. An open circuit wind tunnel was developed in Wang et al. (1997), to study the heat transfer and friction characteristics of a heat exchanger by varying its geometric parameters. Another experimental investigation is found in Wang et al. (2015) where the air side performance of the heat exchanger was studied.

Steady state behavior of automotive radiators placed in wind tunnel were carried out to evaluate the thermohydraulic performance of the heat exchanger. In Cuevas et al. (2011), the authors experimental results were compared with the classical correlations given in the literature. Karthik et al. (2015) investigated the air side performance of the test radiator and presented the heat transfer and flow characteristics results.

The aim of the present work is to study the steady state behavior of a radiator inside a wind tunnel, when subjected to different air supply flows and heating power to the circulating water. An analysis to the air side is proposed to evaluate the heat transfer coefficients and obtain a final non-dimensional correlation for the present heat exchanger.

2. METHODOLOGY

2.1 Experimental procedure

In this work, steady state experimental tests were taken in a typical radiator used in the automotive industry. It is made of aluminum with fins and flat tubes. It has 35 tubes and 36 rows of fins. The total heat exchange surface area is 1.035 m^2 and it has frontal dimensions of (0.146 x 0.178) m. General parameters of the radiator used in the experimental tests are given in Tab. 1.

Table 1. General parameters of the test radiator

Number of flat tubes	35
Number of fin rows	36
A_f - Frontal area of the radiator (m^2)	0.0260
L - Flow length or heat transfer matrix depth in the air flow direction (m)	0.0210
A - Total heat exchange surface area for the air side (m^2)	1.0350
A_i - Total heat exchange surface area for the water side (m^2)	0.2158
A_{min} - Minimum free flow area for the air (m^2)	0.0255
Radiator material	Aluminum

For the analysis, an experimental bench was assembled in wind tunnel with the radiator inside as shown in Fig. 1. Basically, it is composed by 2 circuits, the primary and the secondary. Primary circuit is closed, and it is represented by the water which is pumped with constant flow from a tank to the radiator, passes through its tubes and returns to the water tank. The water inside the tank is heated by up to 5 electric heaters of 1 kW each, allowing to control the water temperature at radiator inlet. Secondary circuit is open and occurs in the wind tunnel, where the atmospheric air flows, passing through the radiator fins. The air velocity is controlled by wind tunnel fan frequency variation.

Figure 1. Experimental apparatus

The experimental bench was outfitted with several measurements instruments for the data acquisition, such as temperature, pressure and mass flow rate.

The water flow pumped was constant, equal to 4.18 L min^{-1}, measured by the time to fill a certain volume. The air velocity in the wind tunnel varied from 1.27 to 5.93 m s^{-1}. It was measure by Pitot tube and an inclined liquid manometer (Dwyer, 3 % accuracy). Therefore, the obtained air mass flow rate was 0.036 to 0.174 kg s^{-1}.

Thermocouples (type T, Omega, limit of error of 0.7 °C verified in the test range with a standard reference) were placed at the test bench to measure the inlet and outlet of water and outlet air of the radiator. Due to the non-uniformity of the air after leaving the radiator, a net of 15 thermocouples was used (based on the works of Wang et al. (2015) and Glazar et al. (2015). The inlet air temperature was a constant of 20 ºC, which was obtained by measuring the ambient temperature. The acquisition of the thermocouples was performed by a National Instrument CompactDaq with 6 modules NI 9211.

A total of 15 tests were considered to characterize the radiator. These were taken at constant water flow and five different air velocities for each one of the three heating powers: 3 kW, 4 kW and 5 kW. All the tests were carried out after reaching the steady state condition, which took about 30 minutes among each test, for a certain heating power.

2.2 Mathematical model

To analyze the thermal radiator behavior in steady state condition, it was set a control volume given in Fig. 2. Energy balances from the first law of thermodynamic Eq. (1) were applied to the radiator control volume, considering the fluids kinetic and potential energy negligible. The energy balances allow to obtain the thermal loads of air and water. Their thermodynamics properties were assumed at the mean temperature between the radiator inlet and outlet.

$$\dot{q}_{rad} = \dot{m}_w c_{p,w}(T_{w,i} - T_{w,o}) = \dot{m}_a c_{p,a}(T_{a,o} - T_{a,i}) \tag{1}$$

where: $\dot{q}_{rad}$ is the heat transfer rate of the radiator [W]; $\dot{m}_w$ is the water mass flow [kg s^{-1}]; $\dot{m}_a$ is the air mass flow [kg s^{-1}]; $c_{p,w}$ and $c_{p,a}$ are the specific heat at constant pressure of water and air respectively [J kg^{-1} K^{-1}]; $T_{w,i}$ is the water temperature at the radiator inlet [°C]; $T_{w,o}$ is the water temperature at the radiator outlet [°C]; $T_{a,i}$ is the air inlet temperature [°C]; $T_{a,o}$ is the air outlet temperature [°C].

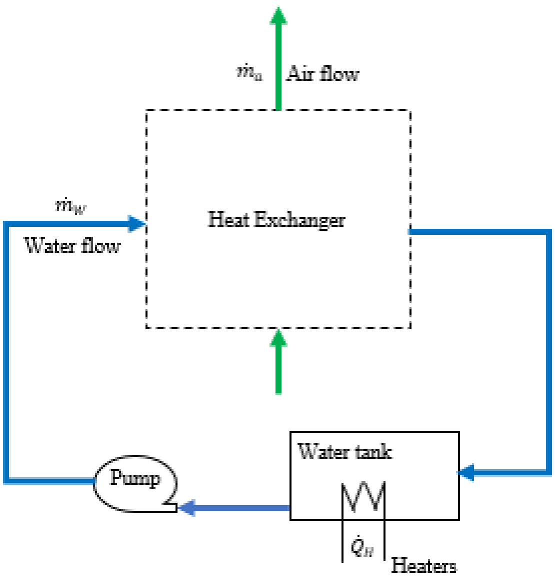


Figure 2. Model control volume

The performance of the heat exchanger may be evaluated by defining the efficiency of the radiator as in Eq. (2). The efficiency might be represented by the ratio between the radiator heat exchange and the imposed heating power by the heaters inside the water tank.

$$e = {}^{\dot{q}_{rad}}\!/_{\dot{Q}_H} \tag{2}$$

where: e is the radiator efficiency; $\dot{Q}_H$ is the heating power [W].

The radiator overall heat transfer coefficient is experimentally calculated by the method of logarithmic mean temperature difference in Eq. (3).

$$\dot{q}_{rad} = U_{exp} A \Delta T_{ML} \tag{3}$$

where: where: U_{exp} is the experimental radiator overall heat transfer coefficient [W m^{-2} K^{-1}]; A is the radiator total heat exchange surface area on the air side [m^2]; ΔT_{ML} is logarithmic mean temperature difference [ºC].

The non-dimensional correlation involving thermal-flow characteristics of the air side was studied as shown in Eq. (4). Then the overall heat transfer can be estimated, assuming no incrustations and neglecting the thermal conduction as in Eq. (5).

$$Nu = a_1 Re^{a2} Pr^{a3} \tag{4}$$

$$\frac{1}{U_{cal} A} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \tag{5}$$

where: *Nu* is Nusselt number; *Re* is Reynolds number, *Pr* is Prandtl number and a_1, a_2, a_3 are coefficients; U_{cal} is the estimated radiator overall heat transfer coefficient [W m^{-2} K^{-1}]; A_i is the heat exchange surface area for the water side [m^2]; A_o is the heat exchange surface area for the air side [m^2]; h_i is the heat transfer coefficient of the water side [W m^{-2} K^{-1}]; h_o is the heat transfer coefficient of the air side [W m^{-2} K^{-1}].

For the air side, the value of Nusselt number is based on hydraulic diameter and calculated from Eq. (6) whereas the Reynolds number based on hydraulic diameter as well is obtained from Eq. (7) as the way used in Karthik et al. (2015).

$$N_u = \frac{h_o D_h}{k} \tag{6}$$

where: D_h is the hydraulic diameter [m]; k is the thermal conductivity of the air [W m^{-2} K^{-1}].

$$R_e = \frac{G D_h}{\mu} \tag{7}$$

where: G represents the total air mass flux [kg m^{-2} s^{-1}]; μ is the dynamic viscosity [N s m^{-2}].

The total air mass flux from Reynolds number is determined by the Eq. (8) whereas the hydraulic diameter is represented by Eq. (9):

$$G = \frac{\rho v A_f}{A_{min}} \tag{8}$$

where: ρ is the air specific mass [kg m^{-3}]; v is the air velocity inside the wind tunnel [m s^{-1}]; A_f is the frontal area of the heat exchanger [m^2]; A_{min} is the minimum free flow area for the air [m^2].

$$D_h = \frac{4 L A_{min}}{A} \tag{9}$$

where: L represents flow length in the air direction [m].

In order to fit estimated overall heat transfer coefficients U_{cal} to experimental values U_{exp}, water heat transfer coefficient h_i and the parameters a_1 and a_2 were evaluated by the least square's method, in Excel Solver. According to literature information, a_3 was considered equal to 1/3 (Gut, 2003) and the initial value of the water side heat transfer coefficient was adopted 1500 W m^{-2} K^{-1} (Kreith et al., 2014).

3. RESULTS AND DISCUSSION

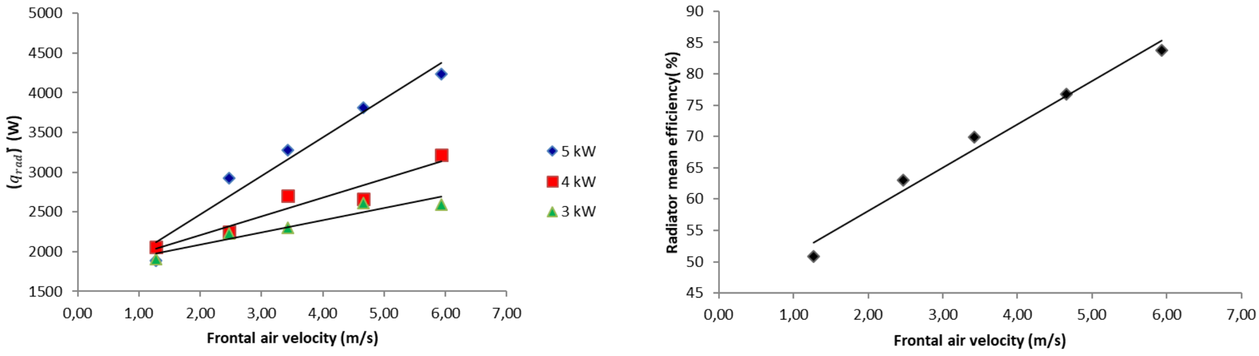
After reaching steady state, it was set a time of 3 minutes of data acquisition for each test. Figure 2(a) represents the radiator thermal rejection obtained by Eq. (1), in function the air velocity. For each heating power, it may be noted a linear behavior of the thermal rejection as the air velocity increases. This occurs due to the increase of air mass flow and the generation of air turbulences among the radiator fins as well. For the higher heating power, the thermal rejection of the radiator is higher as well.

The performance of the radiator was analyzed by Eq. (2). For each imposed heating power, the radiator efficiency was obtained, and then the heat exchange mean efficiency was calculated by the average of each one and given in function of air velocity. The results are illustrated as shown in Fig. 2(b). It may be noted that with the increase of air mass flow rate the heat exchange is enhanced linearly, approaching 85 % of mean efficiency. The heat not exchanged by the radiator was probably lost to environment through the water hoses and the water tank which has large surface area.

To find the heat transfer coefficients of the heat exchanger and evaluate them, the overall heat transfer coefficient was obtained by Eq. (3), from experimental data, for each different 15 tests. Thereby the numerical analysis of non-dimensional correlation to the air side from Eq. (4) was able to be carried out.

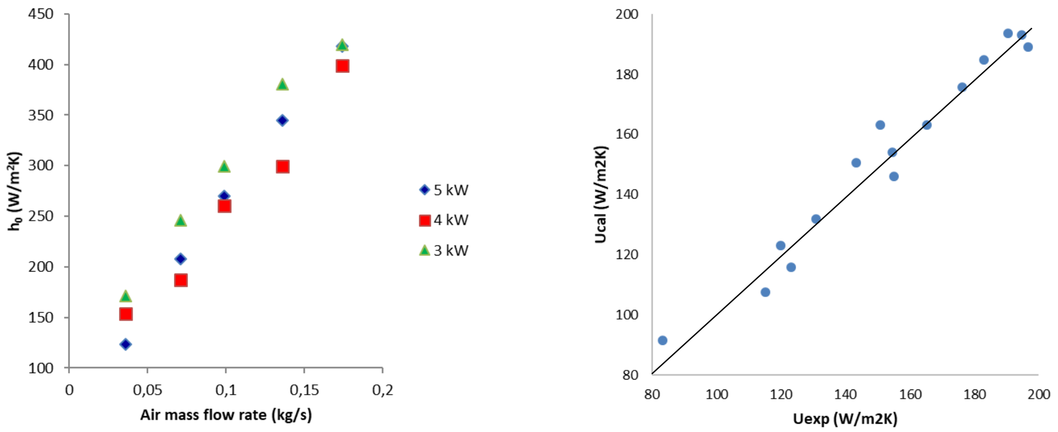
If the water flow is considered uniform, it can be assumed that its heat transfer coefficient h_i is constant. As the constant h_i and coefficients a_1, a_2 were adjusted, the overall heat transfer coefficient was possible to be estimated by Eq. (5) for each test, in order to obtain the minor square error between the experimental and estimated values. The results are shown in Fig. 3.

Figure 3(a) shows the behavior of the heat transfer coefficients of the air side, in function of its mass flow rate. It is interesting to note that, besides the expected linear increase of the coefficients, their values resemble to each other, for each one of the three imposed heating powers. It was obtained a mean correlation of 0.980 among them. It is believed that the small variations are due to the environmental adversity and the difficult to get and control a perfect isolated steady state condition.



(a) (b)

Figure 2. Radiator heat transfer (a) and radiator mean efficiency (b) in function of air velocity



(a) (b)

Figure 3. Heat transfer coefficient for the air side (a) and comparison between experimental and calculated overall heat transfer coefficient (b).

The final values obtained by numerical adjustments were h_i = 1721 W m^{-2} K^{-1}, a_1 = 0.146 and a_2 = 0.780. Figure 3(b) shows a comparison between U_{exp} and U_{cal}. There was a good approach between their values, reaching correlation of 0.982. Thus, the final non-dimensional correlation for the heat exchanger studied is Nu = 0,1459 $Re^{0,7801}Pr^{1/3}$.

The results of Nusselt number is numerically calculated by the obtained non-dimensional correlation. It is given in function of the experimental Reynolds number and illustrated in Fig. 4. The linear relationship between the non-dimensional numbers is noted. This behavior indicates a great influence of the air velocity on the heat transfer coefficients of compact heat exchangers.

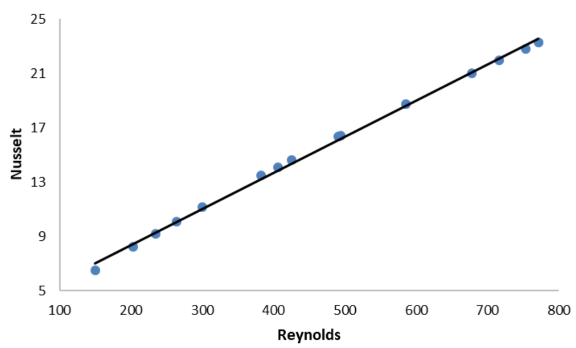


Figure 4. Relationship between Nusselt and Reynolds numbers

4. CONCLUSIONS

The thermal behavior of a small scale radiator was studied in steady state and the effects of operational parameters have been analyzed. An experimental bench was developed in a wind tunnel divided into two circuits. The experimental results showed that there is a linear relation between the heat exchange and the radiator air velocity. So as the radiator thermal rejection, its mean efficiency has the same behavior as well, due to the increase of air mass flow. The 85 % of the radiator thermal rejection shows high importance of the air flow in automotive cooling systems. For the radiator, as a whole, experimental data allows to observe that the overall heat transfer coefficient high values reach nearly to 200 W m^{-2} K^{-1} whereas for only the air side, the heat transfer coefficient reach about 420 W m^{-2} K^{-1}. The linear regression indicated a dimensionless correlation with small deviations from the experimental data.

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