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Development of Experimental Techniques to Determine Convective Heat Transfer Coefficients in Heat Exchanger Devices

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Abstract

Air-water heat exchanger devices are essential components in engine cooling as well as in energy transformation for power generation systems. According to the principle of heat exchanger design, the heat transfer performance is directly related to heat transfer area of the heat exchangers. Heat transfer resistance on the air side of the heat exchanger dominates the overall heat transfer resistance. Hence, the heat transfer area with high permeability and high thermal conductivity surface are preferred. However, the increase in surface area also causes large pressure drop which a higher fluid pumping power machine is required. The optimum design to enhance the heat transfer coefficient and minimize the pressure drop should be crucially considered simultaneously. This study aims at developing of experimental techniques to determine convection coefficients in heat exchanger devices. A wind tunnel with the size of flow area at 35 cm x 35 cm is constructed to suit a selected conventional aluminum finned-flat tube car radiator. Non-linear surface fitting method is performed in order to develop correlations to predict the convective heat transfer coefficient. It is expected that the experimental techniques and proposed correlations can be used by engineers to assess quantitatively a preliminary design of air-water heat exchangers.

Keywords: air-water heat exchanger, heat transfer coefficient, thermal resistance

1. Introduction

Heat exchangers play a key role in many devices which heat transfer process is involved in their operation. The selection of heat exchangers including type and size is a crucial step in order to ensure that the transfer of heat meets the requirement. Thus, the estimation of required heat transfer rate needs to be achieved prior to the heat exchanger selection. To obtain the heat transfer quantity, convective heat transfer coefficient of the heat exchange devices is necessary. Correlations for determining the convective heat transfer coefficient or Nusselt number (Nu) have been established and widely published in various open sources. These correlations are applicable for a certain exchanger geometry and a specified range of Reynolds number (Re) and Prandtl number (Pr). The estimated heat transfer rate obtained from the correlation for different exchanger geometry or out of the range of suggested Reynolds number and Prandtl number may be unreliable.

For simple situation where fluid flows through a smooth circular tube, there are well-known correlations ready to be used for the estimation of convective heat transfer or Nusselt number. Sieder and Tate's [1], Dittus and Boelter's [2] and Gnielinski's [3] correlations are such correlations that have been widely used. In the case where heat transfer on complicated heat exchanger is analyzed, the aforementioned equations may not be applicable. Nilpueng and Wongwises [4] and Kaew-On and Wongwises [5] are two examples focusing on the heat transfer performance which are selected to be presented here. The former journal paper [4] revealed experimental study of single-phase heat transfer and pressure drop inside a plate heat exchanger with a rough surface. The correlations to estimate for Nusselt number and friction factor of water were proposed. The heat transfer analysis was carried out based on the overall heat transfer coefficient (U) between the hot and cold water stream. The Nusselt number correlation of the hot water side was developed by employing the modified Wilson plot technique [6]. The later publication [5] determined the heat transfer coefficient correlations for R134a flowing at low mass flux in a multiport mini channel. The study also used the overall heat transfer coefficient concept. Some relevant dimensionless parameters were included in order to developed new correlations for predicting the frictional pressure drop and the heat transfer coefficient. Mao et al. [7] and Kamsanam et al. [8-11] presented the experimental study of heat transfer in oscillatory flow which is the heat transfer phenomenon in thermoacoustic devices. For that, heat transfer coefficient in steady flow on selected heat exchangers needs to be obtained in advance. The heat exchangers used in the studies were finned-tube heat exchangers with frontal area diameter and fin spacing of 50 mm and 0.7, 1.4, 2.1 mm, respectively. The overall heat transfer coefficient (U) or LMTD approach was adopted to determine the heat transfer coefficient on both hot water and cold water stream simultaneously.

Since heat exchanger is an essential component in many applications. Heat load produced from working process needs to be transferred according to the system requirement. As previously mentioned, equations commonly used to predict the heat transfer coefficient or Nusselt number for fluid flowing through smooth pipe may not be valid for complicated heat exchanger. In addition, correlations developed from experimental study on different types and geometry may not be applicable. The current study focuses on developing experimental techniques to determine convective heat transfer coefficient in heat exchanger devices. The heat exchanger used in this study is an aluminum finned-flat tube type which is similar to car radiator. Correlations to predict the heat transfer coefficient are also developed. These correlations may be useful in designing and selecting the heat exchanger to suit the required applications.

2. Experimental apparatus

To study the heat transfer from a selected finned-flat tube heat exchanger, an experimental apparatus is set up as illustrated in Fig. 1. A wind tunnel is constructed with the size of flow area at 35 x 35 cm while the length for the upstream and downstream section measured from the heat exchanger is 4.8 m and 2.4 m, respectively. The tunnel is a wooden board structure which is affixed by metal sheet on the internal surface in order to minimize friction loss and flow disturbance. At the inlet of the tunnel, honeycomb like structure made from cardboard is installed to perform as a flow straightener to minimize turbulent scale generated from the entrance effect and the blower. The heat exchanger used in the current study is an aluminum finned-flat tube type which is similar to a car radiator. The heat exchanger geometry is as shown in Table. 1.

The flow of air is generated by an electric blower which the operating speed can be controlled in order to vary the air volume flow rate. A centrifugal pump is also installed in the hot water flow loop to deliver hot water through the heat exchanger. Hot water from the heat exchanger exit conduit will return to a hot water bath for reheating and circulating in the loop during the test. Heating element in the hot water bath is an immersion heater with the heating power of 3,500 watts. The flow of hot water can be read directly from a variable area flowmeter. Regarding the air side, air at room temperature flowing from the blower through the test section is also measured to obtain flow velocity. The flow velocity of air can be achieved by means of a Pitot-static tube which is installed in front of the heat exchanger. There is another Pitot-static tube also attached at the downstream of the air flow in order to reassure the velocity measurement. The installation of both Pitot-static tubes are designed to be traversed along the vertical plane on their position. Six probes of type K thermocouple are used for air temperature measurement. Three probes are located in front of the heat exchangers while the rest are placed at the back.

Table. 1 Geometry of the finned-flat tube heat exchanger

Parameters	Value
Frontal dimension	34 cm x 34 cm
Depth (Length in flow direction)	2 cm
Number of water flow channel	34 channels
Hydraulic diameter : air side	1.78 mm
Hydraulic diameter : water side	3.06 mm
Heat transfer area : air side	4.48 m^2
Heat transfer area : water side	0.75 m^2

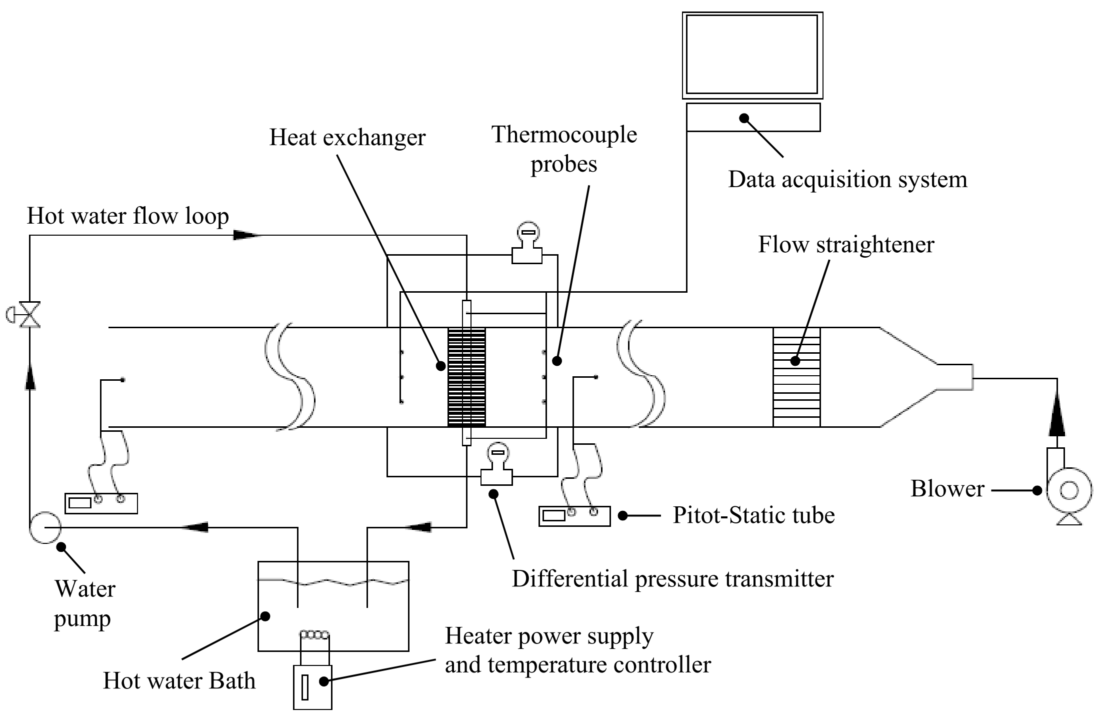

Fig. 1 Schematic diagram of the experimental apparatus.

Two types K thermocouple probes are also installed to collect hot water temperature, one at the inlet and another at the exit of heat exchanger. Temperature data can be collected by a data acquisition system which acquires data at the frequency of 10Hz. Pressure drop across the heat exchanger is obtained by differential pressure transmitter for further investigation.

3. Experimental procedure

As hot water is a heat source of the experiment, water in the hot water bath is heated by an immersion heater. Temperature of water at the heat exchanger inlet is the input signal for temperature controller. In this experiment, temperature at the heat exchanger inlet is monitored not to exceed 90ºC which is the limit of water pump for continuous running condition. While waiting for the temperature of hot water to reach a set point, the blower is turned on and the running speed is adjusted to accomplish the designed flow rate. Once the flow rate of air and hot water reach a steady condition at the preset values, temperature data from all thermocouple probes is collected by the acquisition system. The air flow rate is calculated from air flow velocity which is calculated from the pressure difference of the Pitot-static tube reading. Since the Pitot-static tube can be moved vertically along the cross section, the air flow velocity to be used for volumetric flow rate calculation is achieved from the average of nine reading values at different vertical location. When data collection for the first data point is completed, water flow rate is then adjusted to the next values. According to the hot water pumping power, water flow rate in the experiment is varied in the range of 0.1 – 0.8 GPM. As the test has been carried out until the flow of water reach the maximum rate, the blower running speed is increased to achieve higher air volume flow rate. Then, the acquisition system is triggered. The flow velocity of air in the current study is between 1.0 – 2.7 m/s. The experimental procedure will be repeated until the acquired data is sufficient according to the designed conditions.

4. Data reduction

The aim of the present study is to develop measurement techniques to determine the heat transfer coefficient on both air side and water side of a finned-flat tube heat exchanger. The heat transfer rate between hot water and air is calculated by:

$$\dot{Q}_w = \dot{m}_w c_{p,w}(T_{w,i} - T_{w,o}) \tag{1}$$

or

$$\dot{Q}_a = \dot{m}_a c_{p,a}(T_{a,o} - T_{a,i}) \tag{2}$$

The subscripts w, a, i and o in the above equations denote water, air, and the location at the inlet and exit of the fluid stream, respectively. $c_{p,w}$ and $c_{p,a}$ are specific heat capacity at constant pressure of water and air. The mass flow rate of water ($\dot{m}_w$) is achieved by reading from the variable area flow meter. On the air side, mass flow rate is obtained from:

$$\dot{m}_a = \rho_a A V_{avg} \tag{3}$$

where ρ_a, A and V_{avg} are air density, air flow area and average air flow velocity obtained from nine points along the vertical plane located in front of the heat exchanger. The air flow velocity over the Pitot-static tube is estimated from the equation [12]

$$V = \sqrt{2\Delta P / \rho_a} \tag{4}$$

where ΔP denotes the difference of total pressure and static pressure which is obtained from a micro manometer readings. As the measurement of air flow velocity is performed on 5 points of the same cross section that is located in front of the heat exchanger as illustrated in Fig. 1, the average velocity (V_{avg}) in Eq. (3) is obtained by the average of those 5 values.

The heat transfer in the heat exchanger used in the current experiment involves the thermal resistance consisting of three parts: the resistance for heat transfer from water to the wall of aluminum tube, thermal resistance of aluminum tube and the resistance of heat transfer from aluminum tube to air. It is useful to apply an overall heat transfer coefficient concept as [7, 13]

$$\dot{Q} = UA(LMTD) \tag{5}$$

where

$$UA = \frac{1}{\dfrac{1}{\eta_a(aRe_a^b)A_a} + R_{tube} + \dfrac{1}{(cRe_w^d)A_w}} \tag{6}$$

In Eq.(5), air side Reynolds number (Re_a) is based on the hydraulic diameter of fin spacing and water side Reynolds number (Re_w) is based on the hydraulic diameter of aluminum tube. The logarithmic mean temperature difference ($LMTD$) is given by:

$$LMTD = F\,\frac{(T_{w,i} - T_{a,o}) - (T_{w,o} - T_{a,i})}{\ln\left(\dfrac{T_{w,i} - T_{a,o}}{T_{w,o} - T_{a,i}}\right)} \tag{7}$$

The correction factor (F) is introduced so that the $LMTD$ is suitable for the cross flow arrangement which can be approximated according to [14]. Thermal resistance of aluminum tube is obtained by:

$$R_{tube} = \ln(r_2/r_1)/2\pi L_T k \tag{8}$$

where r_2 and r_1 are the outside and inside tube radii. L_T denotes the effective tube length where the heat transfer to/from the air occurs and k is the thermal conductivity of the aluminum tube. The overall fin efficiency (η_o) is estimated to be around 72% for the short fins with tip convection [7].

In Eq.(5), $\dot{Q}$ and $LMTD_{STD}$ are calculated from Eq.(1) – (2) and Eq.(7) using the measured data. Then, parameters *a*, *b*, *c* and *d* in Eq. (6) are obtained by non-linear surface fitting. The heat transfer coefficient of the air side (h_a) and the water side (h_w) can be defined as:

$$h_a = aRe_a^b \tag{9}$$

$$h_w = cRe_w^d \tag{10}$$

5. Experimental results

The heat transfer coefficient for both water and air sides can be found from Eq.(6). The coefficients *a, b, c* and *d* are obtained by the non-linear surface fitting and are presented in Table.2. Fig. 2 shows the relation of the water side heat transfer coefficient (h_w) with a Reynolds number (Re_W), which is based on an internal aluminum tube diameter. The air side heat transfer coefficients (h_a) are also plotted against the steady flow Reynolds number (Re_a), which is based on the hydraulic diameter of fin spacing.

Table. 2 Heat transfer coefficient correlation determined from data regression process.

Parameters	Present study	Mao's study [7]
Water side		
h_w	$h_w = 0.18Re_w^{2.51}$	$h_w = 0.938Re_w^{0.333}$
Re_w	$12<Re_w<96$	$434<Re_w<2984$
Air side		
h_a	$h_w = 5.05Re_w^{0.47}$	$h_w = 0.308Re_w^{0.697}$
Re_a	$123<Re_a<293$	$115<Re_a<718$

The regression process has been carried out from collected data of 256 points with the R-square value of 0.95. Considering the input data for fitting procedure, the value of $\dot{Q}$ used in Eq.(5) in order to determine the heat transfer coefficient correlations are obtained from the average of the heat transfer estimated on the hot water side ($\dot{Q}_w$) and on the air side ($\dot{Q}_a$). According to the heat transfer rate estimation, the difference of $\dot{Q}_w$ and $\dot{Q}_a$ is within 15%. The difference that is higher than 10% is notified in case of considerably low volume flow rate of water or air or both of them. This may due to the accuracy of the flow rate measurement devices such as the variable area water flow meter and the air flow velocity reading taken from Pitot-static tube.

Fig. 2 also presents comparisons of the predicted heat transfer coefficient on both water side and air side achieved by the correlation developed from the current study and the estimation from Mao's correlations [7]. It could be seen from the graph that the prediction according to this study is considerably higher than that of Mao's correlations [7]. One major course of discrepancy found in the results comparing with the published literature may be the precision limit on flow measurement device such as the Pitot-static tube. At low air flow velocity, the difference in total pressure and static pressure on the Pitot-static tube which was read from a micro manometer for the air flow velocity calculation according to Eq.(4) is imperfectly sensitive to such the low flow rate. This error may also be a source of the discrepancy. In addition, the experimental range performed on both studies. Regarding Mao's investigation, the test was done on

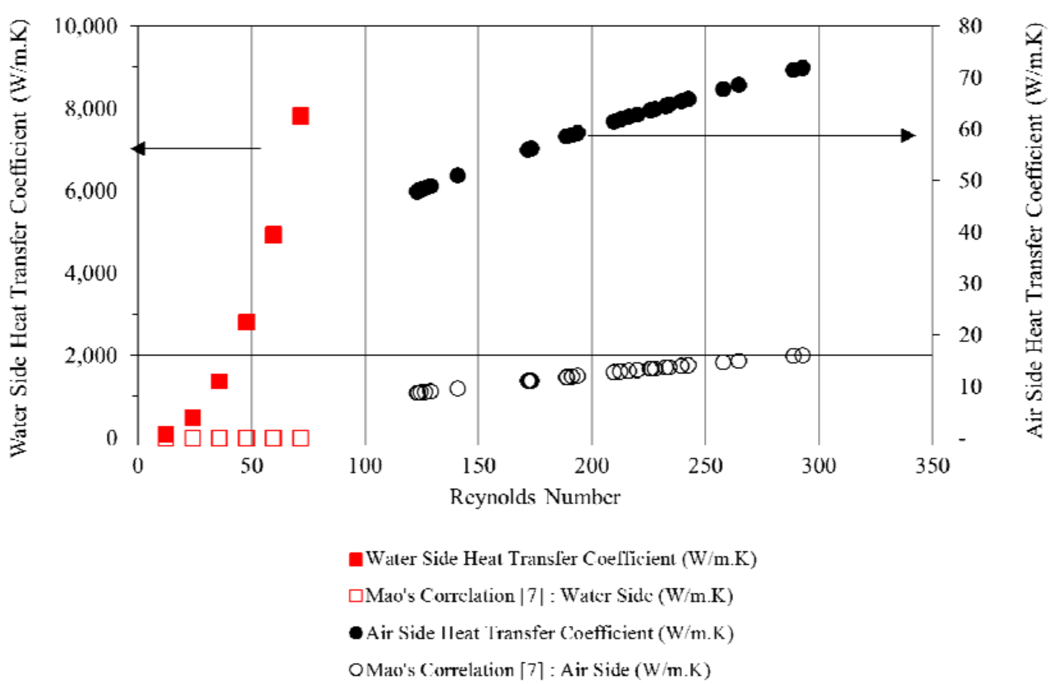

Fig. 2 The dependences of convective heat transfer coefficient on Reynolds number.

the range of Reynolds number that is obviously higher than the ones in this experiment for both water and air stream. This may indicate the substantially dependence of the convective heat transfer coefficient on Reynolds number. Thus, the selection of correlations to predict the convective heat transfer coefficient may be strictly limited to the corresponding range of Reynolds number. Furthermore, the fins on circular tube heat exchanger used in the experiment performed by Mao [7] is not similar to the car radiator type heat exchanger employed in this study. The difference in heat exchanger construction could induce the flow patterns to be dissimilar. Hence, the correlations developed from one heat exchanger may not be applicable to predict the convective heat transfer coefficient for the heat exchanger with different fabrication.

6. Conclusion

An experimental apparatus has been setup in order to develop correlations to predict the convective heat transfer coefficient of both water side and air side on an aluminum finned-flat tube car radiator. The correlations are achieved by means of nonlinear surface fitting from collected data. The experimental results show that heat transfer coefficient is a function of Reynolds number. The developed correlations could be useful for estimating the heat transfer rate from a preliminary design or a selected heat exchanger. Nevertheless, the proposed correlations should be applied for the heat exchanger of similar type with that used in this study. In addition, the expected Reynolds number in the application is suggested to be within the range of this study.

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