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Procedia Engineering 105 (2015) 46 – 55

Procedia Engineering

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6th BSME International Conference on Thermal Engineering (ICTE 2014)

Experimental Study on Heat Transfer and Friction Factor in Laminar Forced Convection over Flat Tube in Channel Flow

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Abstract

The heat transfer and fluid flow over a flat tube in the channel with laminar forced convection is experimentally investigated. The experiments were conducted at a flat tube in the flow direction, the five air velocity between 0.2 and1.0 m/s, and Reynolds number based on the hydraulic diameter (Re_{Dh}) was considered from 124.5 to 622.5. The uniform heat flux supplies are at the surface of the tube are 354.9, 1016.3 and 1935.8 W/m^2 respectively. The experimental results indicate that the average Nusselt numbers ($\overline{Nu}_{Dh}$) for the flat tube increased with increase of Re_{Dh} at fixed of the heat flux supply. The $\overline{Nu}_{Dh}$ increased with an increase of heat flux supply at fixed Re_{Dh}. On the other hand, the friction factor decreased with increases of the front free stream velocity. The $\overline{Nu}_{Dh}$ relationship with Re_{Dh} in the power law, the$\overline{Nu}_{Dh}$–Re_{Dh} correlation was found to be $\overline{Nu}_{Dh} = C \times (Re_{Dh})^m$. The correlation achieved good predictions of the measured data with the minimum root mean square value is 99.70%.

Peer-review under responsibility of organizing committee of the 6th BSME International Conference on Thermal Engineering (ICTE 2014)

Keywords: Laminar forced convection; heat transfer; fluid flow; Reynold's number; heat flux; Nusselt number.

1. Introduction

The cylinders with different shapes of the cross section (e.g., circular, elliptic, flat) are the basic construction blocks of several engineering applications as an example heat exchangers, automotive radiators and nuclear reactors.

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1877-7058
Peer-review under responsibility of organizing committee of the 6th BSME International Conference on Thermal Engineering (ICTE 2014)
doi:10.1016/j.proeng.2015.05.005

The heat exchangers played a pivotal role in energy applications for a long time ago. The issue of heat transfer by forced convection from flat tube shape is significantly due to its essential nature, in addition to that numerous direct, applications energy conservation, heat exchanger and numerous others [1–4]. Flat tubes the fact that the playing a vital role in many engineering applications (e.g. automotive radiators, modern heat exchangers). It is a design has recently been presented using air conditioning for automotive condensers and evaporators. The recent developments in automobile industry aluminium manufacturing technology made the cost of the flat tube heat exchanger building more convenient [2]. In the other hand, the flat tube heat exchangers are expecting to be the better air-side heat transfer coefficients and less than air-side of pressure drop compared with a circular shape heat exchangers, the pressure drop in a flat tube is required to be less of circular tubes, due to a smaller wake area. For the same reason, the expected noise and vibration is less than in flat tube shape compared to the circular shape of tube heat exchangers.

The many studies of heat transfer by forced convection from circular tube on the cross-flow. For the examples, the comprehensive convective heat transfer from circular tube shape was studied [1, 5]. Ota et al. [6] have experimental study heat transfer by forced convection of air flow over the elliptic tube. It was shown that the local Nusselt number was completely various for those that circular tube shape. Baughn et al. [7] presented investigate experimentally heat transfer by forced convection from the single circular cylinder, the tube in tandem with tube in the inlet region of the tube bundle under the uniform heat flux condition. It has been observed that, for together staggered and in-line tube configuration, the local Nusselt number distribution depends on row location of the tubes. Stanescu et al. [8] Studies experimentally the optimum space of circular tube shape in cross-flow of heat transfer by forced convection, the study followed given by Bejan et al. [9] on the optimum of configuration of circular tubes in free convection. Together the two studies deem only equilateral triangle at the staggered arrangement and not investigated of the geometry of the tube. Khan et al. [10] Investigated experimentally of the heat transfer by forced convection of air cross flow over one in-line elliptical tube configuration with the minor-to-major axis ratio of 0.33 and at the horizontal plane of attack. The results indicated that the increased heat transfer rate with an increase of both air and water flow. The heat transfer by forced convection from a circular tube on cross-flow to liquids and air has been studied experimentally may be found in Sanitjai and Goldstein [11]. The measurement of heat transfer shows three zone flows around the tube: recurrent vortex flow zone, reattachment of the shear layer zone and the boundary layer zone of the laminar flow. Chang and Mills [12] have experimental studies of the impact of aspect ratio on heat transfer by forced convection from a circular tube in the air cross-flow. The result of the study shows that the mean heat transfer coefficient raises with reduction the aspect ratio. The heat transfer and pressure investigation experimental both in-line and staggered flat tube configuration by Tahseen et al. [13], Ishak et al. [14]. From the studies shows the effect of heat flux supply, the front free-stream velocity of air flow on the heat transfer coefficient. In the other hand, show the effect of Reynolds number on the pressure drop of cross flow. The results indicate the Nusselt number increase always with an increase of Reynolds number. Three dimension experimental and numerical forced convection study for optimum staggered configuration of finned circular and elliptic cross section of tubes by Matos et al. [15]. The results of the study showed the presence of a relative increase heat transfer up to 19% in the optimal configuration of elliptical tube compared with the optimal circular tube.

Forced convection numerical and experimental of air flow over the elliptic tube bank in cross-flow was reported by Ibrahim and Gomaa [16]. The many design parameters effects of Reynolds number range 5600 to 40000, axis ratios of tube (0.25, 0.33, 0.5 and 1) and the attack angle of the flow range 0 to 150^{o}. The results revealed that, enhance the heat transfer coefficient frequently at the increase of attack angle clockwise into 90^{o}. In addition, the was qualified for the best thermal performance of an elliptical tube heat exchanger the small values of the Reynolds number, attack angle and ratio of an axis. Tahseen et al. [17,18] have 2–D numerical studies incompressible, steady state flow and using the body fitted coordinate (BFC). The first study heat transfer over a two flat tube staggered and second study the heat transfer over series in-line flat tube between parallel plate channel. The two studies show effects of the Reynolds number on the heat transfer coefficient. The results revealed that the heat transfer coefficient increase with an increase of Reynolds number always.

In this study, an experimental investigation on heat transfer and pressure drop over flat tube in air cross flow. The external flow of fluid Reynolds number based on the mean free stream air velocity and outer hydraulic of the tube, changed from 124.5 to 622.5. As well as, the three different heat flux supply at the outer surface of tube (354.9, 1016.3 and 1935.8) W/m^2, respectively, with a view to examine the effects of Reynolds number and heat flux on the air pressure drop and heat transfer rate across the flow of a flat tube.

2. Experimental investigation and techniques

The flat tube arrangement was made from aluminium with the transverse diameter, d_T = 11–mm and longitudinal diameter, d_L = 18.5–mm with the thickness of tube 1 mm. The outer hydraulic diameter D_h = 13.5–mm with the length of 200 mm the photograph and dimensions presented in Fig. 1. The perimeter of the flat tube equal 1.54 times for the perimeter round tube which the diameter equal the small diameter (transverse) for flat tube. The double electric heaters were inserted the inside of the tube to simulate the heat flux originated from a hot fluid. The flat tube was then assembled according to the design in a drawer from Teflon plate type (PTFE), which is the test module. Losses minimized by holding the end of the flat tube between two Teflon walls separated by the distance L = 200 mm.

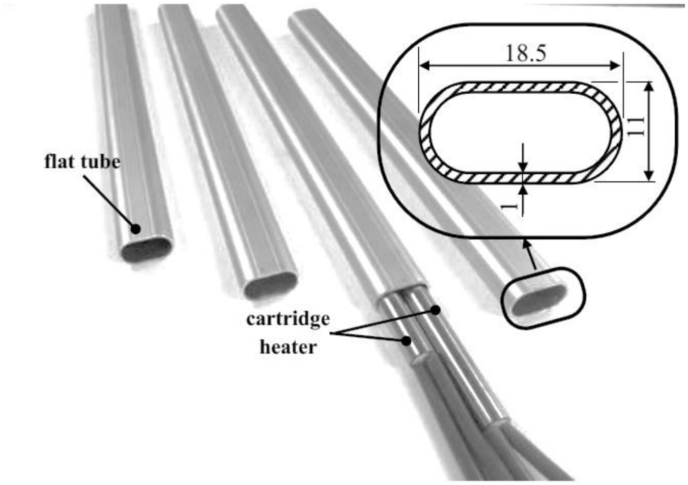

Fig. 1. The photographs of the cross-section flat tube and assembly the heater elements; all dimensions (mm).

Two heating elements, the consisted of cylindrical electric heaters each rated at 50 W up to 850 W with 220 V AC. The outside diameter for the electric heaters is 8 mm and the length of 200 mm the diameter small enough to be inserted in the aluminum tubes see Fig. 1. The heaters were connected in parallel and AC voltage source variables that produced voltages range 0 to 125 V. The maximum power supply 220 V and maximum current of 2.5 A the AC voltage source variable models LOADSTAR 850. The current (I) and voltage (E) measurements were performed with a current clam meter and volt measurement meter with a resolution of 0.05 A and 0.05 V, respectively. Eight thermistors of type EPCOS B57164K0102J NTC (resistance 1000 Ω at 25 ^{o}C) were placed in the test module. The thermistors locations at before, inlet and outlet test rig were placed in the center between the side walls of the wind tunnel and on the midline of the elemental channels. Two thermistors were placed at the arrangement inlet ($T_{in,1}$ – $T_{in,2}$), three at the surfaces of tubes ($T_{s,1}$ –$T_{s,3}$) and two at the outlet ($T_{out,1}$ –$T_{out,2}$) in one elemental channel. An additional thermistor (T_{bef}) was placed on the extended region at 400 mm before the test module to measure the temperature of free stream. Thermistors at the surfaces of the tube showed that the difference between the tubes in one element channel is negligible, and are within ±0.57 °C margin with respect to average three thermistors. Finally, the thermistor put in the extended region for measuring free stream temperatures within ±0.22 °C margin with respect to the average temperature measured inlet arrangement, in all the tests carried out in this work. The velocity measurements were taken with a vane type hot wire anemometer; model YK-2004AH which was placed in the extended flow region, as shown in Fig. 2, in the range of 0.2 m/s–20 m/s. The resolution 0.1 m/s of reading and uncertainty in free stream velocity U_∞ was ±(1–5% +0.1 m/s). The velocity of free stream U_∞ was varied 0.6 m/s–1.0 m/s in this study. The pressure drop measurements were taken using the differential pressure meter model TESTO 510. The operation range of 0–100 kPa with the resolution 1 Pa of reading and the accuracy ±0.3 Pa. The differential pressure measurements had the finality of measuring the pressure drop across each of change the free stream velocity in all experiments as shown in Fig. 2. The experimental work includes the attainment of temperature data using TESTO highly accurate thermometer, model TESTO 110 the nominal range of -50 °C to +150 °C the temperature thermometer resolution is 0.1°C and the accuracy around ±0.2 °C. The thermistors were calibrated in the laboratory for the purpose of finding the deviation limits. The details of the calibration are given in Ishak et al. [14].

Started each run by selecting the voltage and current supply to the cartridge heaters. The selected air velocity of free stream, then we waited for 1.5–2.0 hour while monitor the changes in voltage, current, T_{bef}, $T_{in,1}$–$T_{in,2}$, $T_{s,1}$–$T_{s,3}$ and $T_{out,1}$–$T_{out,2}$. The relative deviation in the voltage, current and temperature were range 0.3%-2.6%, 1.7%–3.6% and 0.16%–0.7%, respectively. These changes were estimated relatively by repeating the same value Re_{Dh} value within 7.5–8.0 hours. It should be noted that these relative changes are small when compared with the uncertainties in the relevant measurements.

3. Data reduction

In this experiment, it is assumed steady state flow. To investigate in the following relations for the relevant air properties and used in the following calculations. They are based on data and valid in the range of temperature. $250\text{K} \le 0.5(\overline{T_{\text{in}}} + \overline{T_{\text{out}}}) \le 275\text{K}$ [19];

$$\left.\begin{aligned}
\rho_{\text{a}} &= 2.209 - 3.414 \times 10^{-3}\left(\frac{\overline{T_{\text{in}}} + \overline{T_{\text{w}}}}{2}\right), && \text{kg/m}^3 \\
c_{P_{\text{a}}} &= \left[9.848 + 6.76 \times 10^{-4}\left(\frac{\overline{T_{\text{in}}} + \overline{T_{\text{w}}}}{2}\right)\right] \times 10^{2}, && \text{J/(kg K)} \\
k_{\text{a}} &= \left[3.479 + 7.58 \times 10^{-2}\left(\frac{\overline{T_{\text{in}}} + \overline{T_{\text{w}}}}{2}\right)\right] \times 10^{-3}, && \text{W/(m K)} \\
\mu_{\text{a}} &= \left[4.475 + 4.564 \times 10^{-2}\left(\frac{\overline{T_{\text{in}}} + \overline{T_{\text{w}}}}{2}\right)\right] \times 10^{-6}, && \text{kg/(m s)}
\end{aligned}\right\} \tag{1}$$

in expression,

$$\overline{T_{\text{in}}} = \frac{1}{n}\sum_{i=1}^{n}(T_{\text{in}})_{\text{i}}\ , \quad n = 2\,; \ \ \overline{T_{\text{w}}} = \frac{1}{n}\sum_{i=1}^{n}(T_{\text{w}})_{\text{i}}\ , \quad n = 3$$

The electrical heat gain rate [voltage (E) × current (I)], is uniform heat flux (UHF) from the outer tube surface can be evaluated as:

$$q_{\text{sup}} = I \times E \tag{2}$$

The steady state heat balance of the electrical heat test surface can be written as:

$$q_{\text{convection}} = q_{\text{sup}} - q_{\text{conduction}} - q_{\text{radiation}} \tag{3}$$

Heat transfer from the system may be;

i) Conduction between lab and the wall of the tubes was neglected because of the extremely low thermal conductivity of air (0.23 W/(m °C)) of Teflon and negligible temperature difference between the lab outer walls of the Teflon.
ii) Radiation heat transfer between surfaces of tubes and surrounding were also neglected. Based on the measurement of $0.5(\overline{T_{\text{in}}} + \overline{T_{\text{out}}})$ and mean $\overline{T_{\text{s}}}$ was estimated the radiation transfer coefficients as [10];

$$\overline{h_{\mathrm{rad}}} = \varepsilon \times \sigma \times \left[\left\{ \left(\frac{\overline{T_{\mathrm{in}}} + \overline{T_{\mathrm{out}}}}{2} \right)^2 + (\overline{T_{\mathrm{s}}})^2 \right\} \times \left\{ \left(\frac{\overline{T_{\mathrm{in}}} + \overline{T_{\mathrm{out}}}}{2} \right) + \overline{T_{\mathrm{s}}} \right\} \right] \tag{4}$$

here

$$\overline{T_{\mathrm{s}}} = \frac{1}{n} \sum_{i=1}^{n} (T_{\mathrm{s}})_{\mathrm{i}} \ , \quad n = 3$$

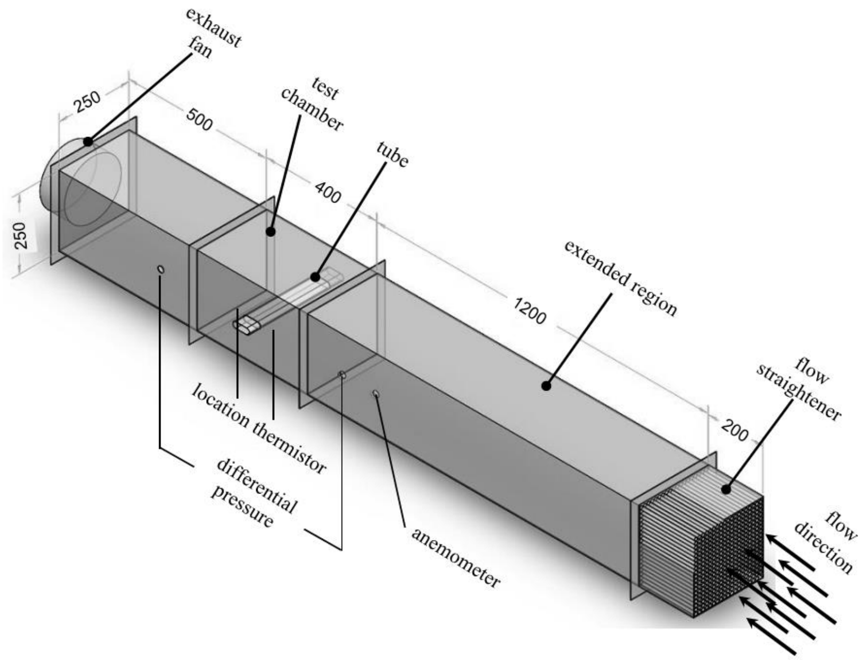

Fig. 2. The schematic diagram of the experimental setup, all dimensions (mm).

For a commercial aluminum tube with emissivity, $\varepsilon \cong 0.028$ [19], $\overline{h_{\mathrm{rad}}}$ was found in a range between (0.174–0.205) W/m^2 $^{\mathrm{o}}$C, in approximately 0.0102%–0.15% of the air side convection heat transfer coefficient, $\bar{h}$. Thus, the heat transfer between the air and the surface of the tubes was actually due to the convection and the mechanism of the Equation (3) was rewritten [20]:

$$q_{\mathrm{convection}} \cong q_{\mathrm{sup}} = \bar{h}\, A_{\mathrm{s}} \left[\overline{T_{\mathrm{s}}} - \left(\frac{\overline{T_{\mathrm{in}}} + \overline{T_{\mathrm{out}}}}{2} \right) \right] \tag{5}$$

Can be employed with

$$A_{\mathrm{s}} = [\pi d_{\mathrm{T}} + 2(d_{\mathrm{L}} - d_{\mathrm{T}})] \times L$$

The definition $\overline{T_{\mathrm{in}}}$ the inlet air temperature was the variable (24.5–25.3) $^{\mathrm{o}}$C and the average temperature of the surfaces of tube, $\overline{T_{\mathrm{s}}}$. For the steady state condition, the overall heat transfer rate, $q_{\mathrm{convection}}$ was equal to the electrical heat supply, q_{sup}. From Equation (3), the average heat transfer coefficient was determined as:

$$\bar{h} = \frac{q_{\mathrm{sup}}}{A_{\mathrm{s}} \left[\overline{T_{\mathrm{s}}} - \left(\frac{\overline{T_{\mathrm{in}}} + \overline{T_{\mathrm{out}}}}{2} \right) \right]} \tag{6}$$

And is usually used in the average Nusselt number which is defined via Equation (7);

$$\overline{\mathrm{Nu}_{D_\mathrm{h}}} = D_\mathrm{h} \frac{\overline{h}}{k_\mathrm{f}} \tag{7}$$

in the expression, k_f is the thermal conductivity of air and the hydraulic diameter;

$$D_\mathrm{h} = 4 \times \frac{\left[\frac{\pi}{4}(d_\mathrm{T})^2 + (d_\mathrm{L} - d_\mathrm{T})d_\mathrm{T}\right]}{\pi d_\mathrm{T} + 2(d_\mathrm{L} - d_T)} \tag{8}$$

The Reynolds number based on the hydraulic diameter is defined;

$$\mathrm{Re}_{\mathrm{D}_h} = U_\infty \times D_\mathrm{h} \times \frac{\rho}{\mu} \tag{9}$$

The friction factor is calculated as follows [21];

$$f = \frac{2\Delta P}{\rho U_\infty^2}\left(\frac{D_\mathrm{h}}{4L}\right) \tag{10}$$

The experimental uncertainty issues were dealt by Editorial [22] and Holman [23]. There is more than one way to estimate the uncertainty in the experimental results and has presented by Kline and McClintock. The few sample calculations here. The independent parameters (such as tube dimensions, temperature, velocity, etc.) and the dependent parameters (like $\mathrm{D}_h, \mathrm{Re}_{\mathrm{D}_h}, q_{\mathrm{sup},} \overline{\mathrm{Nu}_{\mathrm{D}_h}}$, etc.). Which are independent functions of other measured parameters, an uncertainty for the independent variable spreads them, according to their functional relationship. The example, in case of the Reynolds number, $\mathrm{Re}_{\mathrm{D}_h}$;

$$\mathrm{Re}_{\mathrm{D}_h} = U_\infty \times D_h \times \frac{\rho}{\mu} \tag{11}$$

The uncertainties of density, free stream velocity; viscosity and hydraulic diameter propagate into $\mathrm{Re}_{\mathrm{D}_h}$, and can be estimated in terms of absolute values (%) as follows:

$$U_{\mathrm{Re}_{\mathrm{D}_h}} = \left[\left(\frac{\partial \mathrm{Re}_{\mathrm{D}_h}}{\partial \rho} U_\rho\right)^2 + \left(\frac{\partial \mathrm{Re}_{\mathrm{D}_h}}{\partial U_\infty} U_{U_\infty}\right)^2 + \left(\frac{\partial \mathrm{Re}_{\mathrm{D}_h}}{\partial \mu} U_\mu\right)^2 + \left(\frac{\partial \mathrm{Re}_{\mathrm{D}_h}}{\partial D_h} U_{D_h}\right)^2\right]^{1/2} \tag{12}$$

The entire description of the uncertainty calculation can be found in Ishak et al. [14]. The uncertainties in finding the$\mathrm{D}_h, \mathrm{Re}_{\mathrm{D}_h}, q_{\mathrm{sup},} \overline{\mathrm{Nu}_{\mathrm{D}_h}}, f$ were estimated and found to remain approximately within ±5.68%, ±2.19%, ±5.67%, ±4.10% and ±1.10%, respectively.

4. Results and discussion

In the present experimental work, the effects of the front free-steam velocity and heat flux supply on the air flow and heat transfer over flattened tube in the channel. The relationship $\overline{\mathrm{Nu}_{\mathrm{D}_h}} - q_{\mathrm{sup}}$for several $\mathrm{Re}_{\mathrm{D}_h}$number was plotted in Fig. 3. The results reveals that the $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number increased nearly linear always with an increasing $q_{\mathrm{sup},}$ as was expected. The heat transfer coefficient depended on the temperatures difference between the tube surface and free stream of air flow. In other words, increasing $q_{\mathrm{sup},}$ a leads to increased temperature gradient thus increase further in $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number.

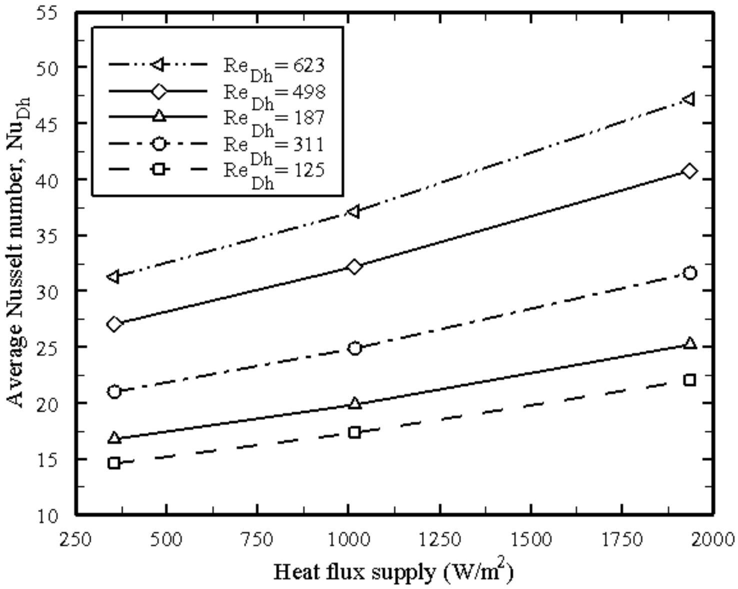

Fig. 3. The $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number variation with q_{sup} at different $\mathrm{Re}_{\mathrm{D}_h}$ number.

The $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number various with $\mathrm{Re}_{\mathrm{D}_h}$ number at a different heat flux supply is plotted in Fig. 4. As might be expected, the $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number increases with increasing of the $\mathrm{Re}_{\mathrm{D}_h}$ number. The reason for this issue to the reduction in thickness of a boundary layer and the powerful periodical vortices of air flow in the detached zones near to the behind wall of the tube. In general, $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number increases with $\mathrm{Re}_{\mathrm{D}_h}$ number. The heat transfer coefficient is largest at the stagnation point and increase with a long time contact of fluid with the surface of tube because of the increased thermal boundary layer. On the other hand, when $\mathrm{Re}_{\mathrm{D}_h}$ number increases the level of turbulence strength caused by secondary flow was increased. This enables be construed increase $\overline{\mathrm{Nu}_{\mathrm{D}_h}}$ number. In this regard, the $\overline{\mathrm{Nu}_{\mathrm{D}_h}} - \mathrm{Re}_{\mathrm{D}_h}$ correlation was using a non–linear regression analysis of the empirical data is defined as:

$$\overline{\mathrm{Nu}_{\mathrm{D}_h}} = C \times \left(\mathrm{Re}_{\mathrm{D}_h}\right)^m \tag{13}$$

The constant (C) and (m) are estimated by statistical processing for experimental results, can be seen in Table 1. Corresponding trends are given in Fig. 5. It is evident from the figure, the experimental data of Žukauskas [1] with circular tube, the analytical solution by Khan et al. [24] for elliptic tube and for $100 \leq \mathrm{Re} \leq 5000$ and the numerical study by Bahaidarah et al. [25] using flat tube shape for $100 \leq \mathrm{Re} \leq 350$. In general, current work satisfactorily agreed with trends of the previous studies. For the sake of comparison, the trends of result near to the experimental result by Žukauskas [1] are illustrated in the same figure. Noticeable, there was a deviation of about 4.4–16.7% between current results and numerical results from Bahaidarah et al.[25].

Table 1. Summary of estimated parameters in Eq. (13).

q_{sup}, W/m^2	C	m	R^2 (%)
354.90	1.313	0.500	100
1016.32	1.557	0.500	100
1935.84	1.921	0.503	99.7

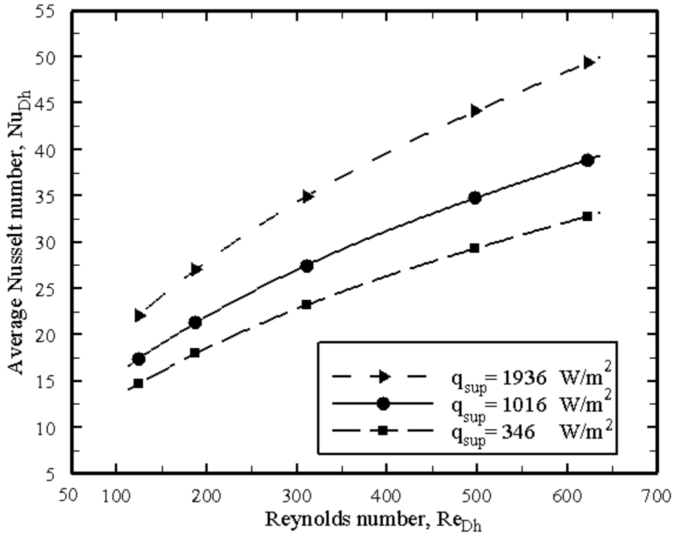

Fig. 4. The $\overline{Nu_{D_h}}$ number versus with Re_{D_h} number at several q_{sup}.

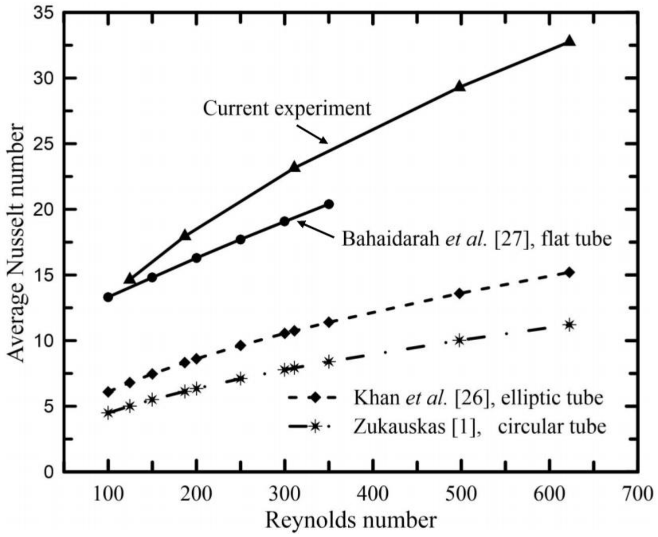

Fig. 5. Comparison of the present results and foretelling with published literature [27].

The pressure drop and friction factor, *f* for isoflux flat tube in the air versus free stream velocity are illustrated in Fig. 6. It is clear that ΔP increased nearly linearly and *f* decreased with increase the free stream velocity. Although, the *f* is directly proportional with the output the ΔP decrease with increase air velocity. With the rise in free stream velocity, relatively contributes to increase the inertial forces whereas was decreases of the *f* (that is the control factor on the friction). This can be attributed for a reason, to the preferable aerodynamic shape for the flat tube which generates less drag force. The friction factor value appears to reduction approximately to 0.018 according to the maximum velocity is 1 m/s.

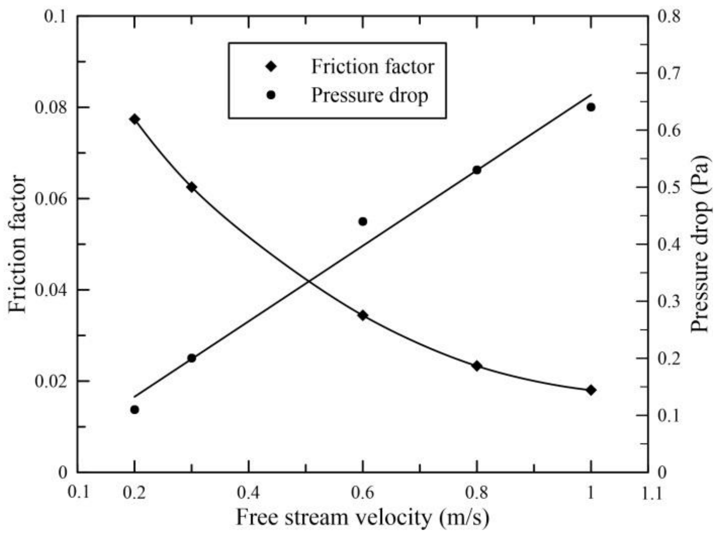

Fig. 6. Variation of ΔP parameter and f with free stream velocity of air.

5. Conclusions

The heat transfer and pressure drop a flat tube in a cross flow of air have been experimentally study. A heat flux supply in the range of 354.9 W/m^2 –1935.84 W/m^2 with the Reynolds number ranging are $125 \leq \mathrm{Re}_{\mathrm{D}_h} \leq 623$. This paper examines the impact of two parameters on the average heat transfer coefficient and friction factor for air flow over single flat tube. It can be summarized that the experimental results are drawn:

i) The heat transfer coefficient increase almost linear with increase heat flux supply for all free stream velocity.
ii) The average Nusselt number increase with increase Reynolds number with any heat flux supply tested.
iii) The pressure drop increased and friction factor decrease with increasing of free stream velocity.
iv) The compared of the results with similar earlier studies as though the good approximation to the similar trends.
v) Lastly, this study clearly show that the flat tube has the pressure drop is small and the heat transfer coefficient is higher compare with circular and elliptic shapes of tube.

Acknowledgements

The authors would like to acknowledgements Ministry of Education Malaysia and Universiti Malaysia Pahang for providing laboratory facilities and financial support under project no. RDU120103 .

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