Effect of attack angles on air side thermal and pressure drop of the cross flow heat exchangers with staggered tube arrangement

Dr. SANJAT KUMAR MISHRA,NIT,BBSR, sanjatkumar@thenalanda.com

Mr.SUBHENDU MOHAN KUMAR BASANTIA, subhendumohan@thenalanda.com

Abstract

An experimental study on the air side heat transfer and pressure drop characteristics in steady state for flat tube design stepped crossflow heat exchangers is presented. The effect of angles of attack on the airside is an important study. In addition, the effect of the aspect ratio of the tubes was investigated. Nineteen heat exchangers with four aspect ratios (0.18, 0.39, 0.66 and 1) and six impact angles (0, 30, 60, 90, 120 and 150) are investigated. The water temperature was 75 °C and outside air flowed through the heat exchangers at a speed of 2-6 m/s. The results show that the heat and pressure loss characteristics had an effect on the impact angles and tubular proportions. For all values of air speed, for each tube aspect ratio and angle of attack (0 90), the heat transfer rate and pressure drop were increased, while the thermohydraulic performance decreased. Looking at the effect of the tube aspect ratio on the heat and pressure loss characteristics, the results showed that the change in the angle of attack of the smaller tube aspect ratio had more influence on the heat and pressure loss than the change . of the angle of attack of the dimensions of the larger tubes.

1. Introduction

Cross flow heat exchangers are widely used in many thermal devices such as production process, boiler economizer, air pre heater, air conditioning and automotive radiator, etc. Mostly, cross flow heat exchanger has two types: with fins (fin and tube heat exchanger) and without fin. If the hot fluid and the cold fluid have large heat transfer coefficient different such as gas and liquid, the heat transfer rate is limited by the lower heat transfer coefficient side in this case is the gas side. The fins require for increase the heat transfer surface area on the gas side due to increase the heat transfer rate. On the other hand the fin do not has necessary installation. If the fin and tube heat exchanger is uses for heat exchange between water and flue gas such as boiler economizer, to keep good heat transfer rate the fins are require frequently cleaning this causes the high operating cost, that some disadvantage of the fin and tube heat exchanger. In addition the fin caused pressure drop higher than the non-fin. However, the enhance heat transfer rate of the cross flow heat exchangers can use other methods such as tube arrangements, shapes of tube and etc. In the past investigations, the non-circular tubes have been considered to enhance heat transfer rate in cross flow heat exchangers.

The heat transfer on the tube surfaces around the tubes of different blocked ratios with the Reynolds range from 120 to 390 was investigated by E.Buyruk et al. [1]. They found the maximum Nusselt number occurred at $0^\circ$of the attack angle between air stream and the tube surfaces, while the attack angles range from 110° to 130° provided the minimum Nusselt number which depend the blocked ratios. In addition, the height pressure coefficient occurred at the same point of the height Nusselt number. A.Sufwat et al. [2] studied the heat transfer and friction factor of the cross flow heat exchangers with inline and staggered arrangements by used mathematical model. They shown the Nusselt number of the staggered tube arrangement more than the inline tube arrangement when the ratio of longitude pitch/tube diameter (Sp/D) < 4, while the Sp/D > 2.5 the inline tube arrangement give the friction factor more than the staggered tube arrangement. V.K.Mandhani et al. [3] founded in the range of Reynolds number from 1 to 500 the Nusselt number of the tube banks was decreased when the porosity of the tube banks increased. W.A.Khan et al. [4] reported the heat transfer rate of 16.4 mm tubes diameter with inline and staggered arrangements have approach when the distant between tubes was increased. Pitch of circular, oval and wing-shape tubes have effect on the air side heat transfer and pressure drop were studied by Andrej Horvat et al. [5]. They used mathematical model for studied, and found in range of the Reynolds number from 400 to 4500 and pitch to diameter tube ratios from 1.125 to 2 the circular tubes give the drag coefficient highest the oval and the wing-shape tubes, while the minimum drag coefficient due to the oval shape tube. For the heat transfer coefficient, the circular tube give the maximum value. A.Nouri-Borujerdi et al. [6] were investigated heat transfer and pressure drop on air side of the cam – shape tubes in the Raynolds number range from 15000 to 27000 and attack angles from 0° to 180°. They found the attack angle is 90° give maximum drag coefficient, and give maximum Nusselt number as well. The maximum thermal hydrualic performance to comparison attack angles found that the attack angle at 30° give maximum, and more than the attack angle at 90°about 50%. Toa et al. [7] presented the heat transfer and pressure drop of the oval tubes with finned higher the circular tube with finned about 30% and 10% respectively. Talaat et al. [8] used mathematical and experimental for studied effect of axis ratios and attack angles of the elliptic tubes on air side heat transfer performance in the Reynolds number range from 5600 to 40000. They reported the maximum of heat transfer and pressure drop have occurred on the attack angle of 90° and minimum axes ratio because it has the attack surface highest other attack angles. While the 0° of attack angle give the maximum thermal hydraulic performance. Khaled Al-Salem et al. [9] found the porous metal enhanced heat transfer of the tubes while the pressure drop had slightly increased.

From the previous studies it can be seen that the heat transfer rate of cross flow heat exchangers can be enhanced by non-circular tube shapes selected method and modified tube arrangements method, it is well established that the cam shape tubes, the ellipse shape tubes and the wing shape tubes have thermal hydraulic better than the circular tubes shape. However, most of these previous studies do not have study on heat transfer and pressure drop of the flat tube shape. The purpose of this study is to investigate the heat transfer and pressure drop of the flat tube shape with different attack angles. The results may provide additional information for designs of cross flow heat exchanger to height performance without fin or reduce number of fin.

Nomenclature

A_c	cross section area (m^2)
A_s	heat transfer surface area (m^2)
Ar	aspect ratio
Cp	specific heat (kJ/kg °C)
d	diameter (m)
F	correction factor
f	friction factor
h	heat transfer coefficient (kW/m^2 °C)
k	thermal conductivity (W/m °C)
L	length of tube (m)
$\dot{m}$	mass flow rate (kg/s)
Nr	number of tube row
Nu	Nusselt number
P	perimeter (m)
p	pressure (N/m^2)
$\dot{Q}$	heat transfer rate (W)
Re	Reynolds number
S_t	transvers tube pitch (m)
T	temperature (C)
u_{max}	air velocity (m/s)
u	air velocity (m/s)

Greek symbol

μ	performance (W/Pascal)
	absolute viscosity (kg/s-m)
ρ	density (kg/m^3)

Subscript

a	air
ave	average
h	hydraulic
i	inlet
ln	logarithmic mean
max	maximum
o	outlet
s	heat transfer surface
w	hot water

2. Experimental setup and procedure

Experiments were conducted in an open circuit wind tunnel with a rectangular has cross section of 0.14 m × 0.18 m and 3.5 m length as shown in Fig.1, this consistent of main parts are as follows a centrifugal air blower, a heat exchanger test section, a hot baht and measuring system. The air at ambient conditions flowed into the wind tunnel by the 1500 cfm centrifugal air blower with a frequency inverter to control flow rate of the air. In the wind tunnel, the air flowed through a straightener which installed at a blower outlet with the purpose of laminarizing the flow before enters the test section. A distant between the straightener outlet and the test section inlet is 2.00 m to provide the fully developed of the airflow before enters the test section. The air free stream enters the test section with velocity in the range of 2.00 to 8.00 m/s. In the test section, the heat exchanger fabricated from the commercial copper tubes with 0.3 m length it was installed in a direction perpendicular to the air free stream, the hot water with constant temperature of 75 °C from the hot bath was supplied to the heat exchanger. The heat transfer between the test section and the surrounding was prevented by the insulation Aeroflex of 25 mm thickness.

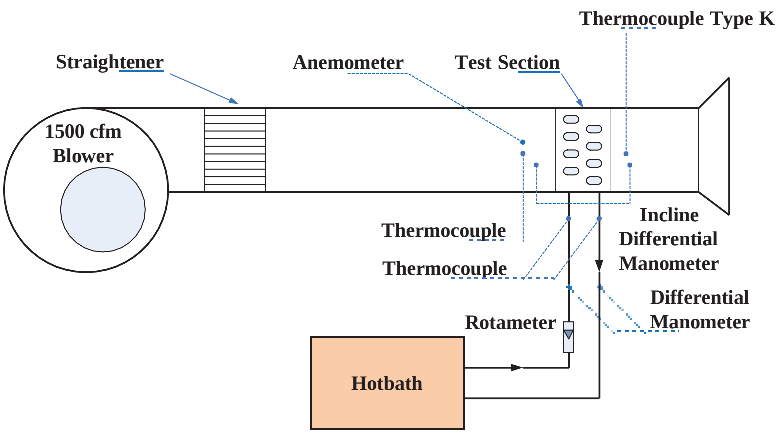


Fig. 1. Schematic diagram of the experimental apparatus

The inlet and the outlet temperatures of the air stream are measured by thermocouple type K with 9 points on the upstream and downstream cross section areas of the test section. The heat transfer of hot water is necessary to compare the air side, thus the temperature of hot water at inlet and outlet are also measured by the thermocouple type K with calibrated. The temperature of the air stream and the hot water were recorded by the data logger (Yokogawa mv 1000) with an accuracy of± 0.1 ℃. The velocity of the air stream was measured by the vane type anemometer (Testo 416) with an accuracy ± 0.2 m/s at the upstream with 9 points of grid. The volume flow rate of hot water was measured by the floating rotameter (Nitto instruments K 500 Series) with an accuracy of ± 5% of full scale. The pressure drop across the heat exchanger of the air stream and the hot water are measured by differential incline manometer with ± 1% accuracy of full scale and differential digital manometer ± 0.3% accuracy of full scale respectively. In the experimental, all data were recorded when the system was steady state.

In the study, the effect of attack angles (explain in Fig. 2) on the air side heat transfer and the pressure drop are examined. A total of 19 cross flow heat exchangers with staggered tube arrangement having four tube aspect ratios (0.18, 0.39, 0.66 and 1) and six attack angles (0°, 30°, 60°, 90°, 120° and 150°) were tested. The configurations of the flat tube, the tube arrangement and the heat exchanger are show in Fig.3 (a), Fig.3 (b) and Fig.4 respectively. To make sure of the experiments result, the different between heat transfer rates of the air side and the water side have to little

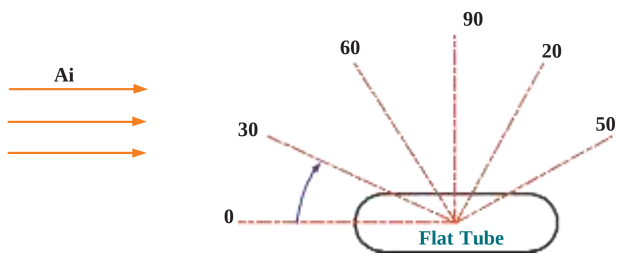


Fig. 2. Attack angles between the flat tube and the air stream

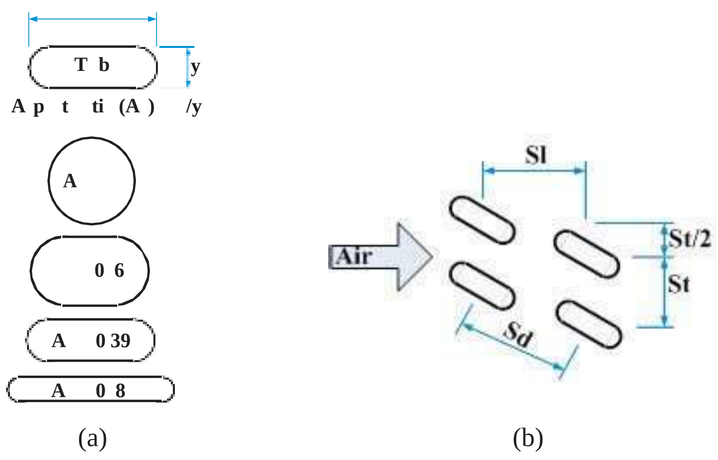


Fig. 3. (a) Flat tube cross sections with various aspect ratios (b) Tube arrangement

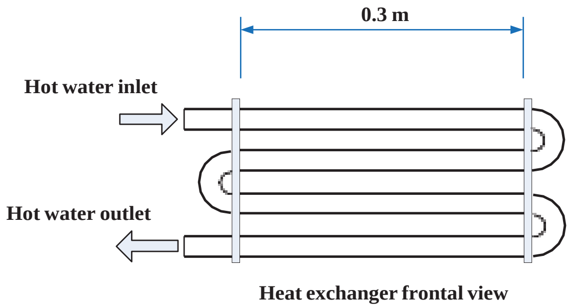


Fig. 4. Frontal view of heat exchanger

3. Data reduction

In this study, heat transfer rate from the hot water to the air is steady state. The heat exchange of both fluids can be calculated by energy balance following Eq. (1) and Eq. (2) for the air and the hot water respectively;

$$\dot{Q}_a = \dot{m}_a Cp_a (T_{ao} - T_{ai}) \tag{1}$$

$$\dot{Q}_w = \dot{m}_w Cp_w (T_{wi} - T_{wo}) \tag{2}$$

The properties of the air and the hot water are used at the bulk temperature of inlet and outlet heat exchangers.

In experiments, the energy changed of both fluids have little different because some heat loss to the surroundings. Thus, to analyse the heat transfer characteristics in the heat exchanger were used the average heat transfer of both fluids which can be express as;

$$\dot{Q}_{ave} = \frac{\dot{Q}_a + \dot{Q}_w}{2} \tag{3}$$

The average heat transfer coefficient and the average Nusselt number in the airside can be obtained from Eq. (4) [8] and Eq. (6) [8], respectively;

$$h_a = \frac{\dot{Q}_{ave}}{A_s \Delta T_{\ln}} \tag{4}$$

$$A_s = \pi dLN \tag{5}$$

$$Nu_a = \frac{h_a d_h}{k_a} \tag{6}$$

where

$$\Delta T_{\ln} = F \frac{(T_{wi} - T_{ao}) - (T_{wo} - T_{ai})}{\ln\left(\frac{T_{wi} - T_{ao}}{T_{wo} - T_{ai}}\right)} \quad (7)$$

and

$$d_h = \frac{4A_c}{P} \quad (8)$$

The airside friction factor can be obtained from Eq.(9)

$$f = \frac{2\Delta p}{\rho u_{max}^2 N_r} \quad (9)$$

Where

$$u_{max} = \frac{u_\infty S_t}{S_t - d} \quad (10)$$

The Reynolds number express in Eq.(11) is based on the circular tube

$$\mathrm{Re} = \frac{\rho_a u_{max} d}{\mu_a} \quad (11)$$

In the practical, the heat exchanger requires the fan power for delivers the air mass through it. Thus, to indicate the heat exchanger performance may be considers to the pressure drop across it. The heat exchanger performance can be defines in form the ratio of the average heat transfer rate to the air side pressure drop follow as Eq. (12) [8];

$$\eta = \frac{\dot{Q}_{ave}}{\Delta p} \quad (12)$$

4. Results and discussion

Verification of the experiment results

Before using the results obtained from the experiments, it is necessary to verification. In the experiments, the heat transfer and pressure drop in the heat exchanger are verified in terms of the Nusselt number and the friction factor, both are compared from the correlations of Sukuakas (Yonus A. Cengel [10]). Fig. 5 -6 show the data obtained from the experiments for the plain tube are reasonable agreement with the predicted results from the proposed correlations with the discrepancy of less than 2.95%, 4.61% for the Nusselt number and friction factor, respectively.

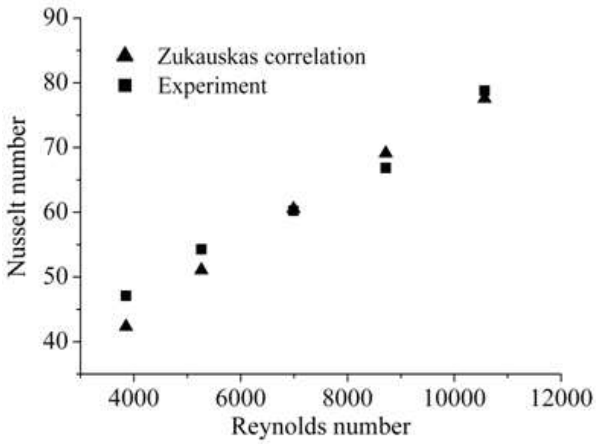


Fig. 5. Verification of the Nusselt number for the circular tube

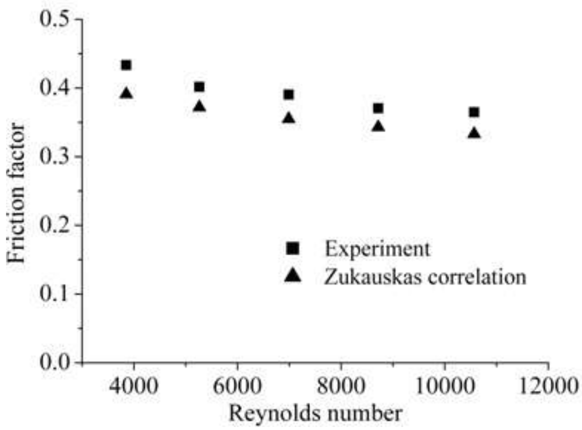
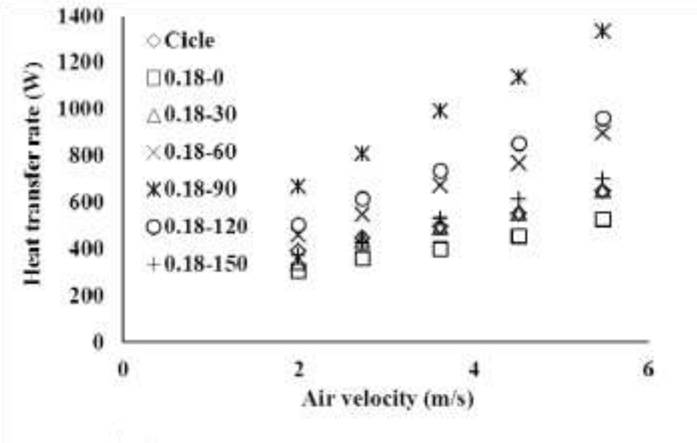


Fig. 6. Verification of the friction factor for the circular tube

Effects on heat transfer characteristic

All of tube aspect ratios and attack angles the heat transfer rate increased following as air velocity increasing as show in Fig. 7 – Fig.9. The 90° of attack angle give the maximum heat transfer rate, while the minimum heat transfer rate occurred in the zero degree of attack angle. The different of heat transfer rate between the 0° and 90° have increased with increasing the aspect ratios because effect of change attack angle on heat transfer surfaces of the low aspect ratio more than the height aspect ratio. For the tube aspect ratios 0.18, 0.36 and 0.66, the heat transfer rate of 90° of attack angles higher than the 0° of attack angles by 140.0%, 82.2% and 41.4% respectively. In this studied, the maximum and minimum heat transfer rates occurred at the 90° and 0° of tube aspect ratio is 0.18.



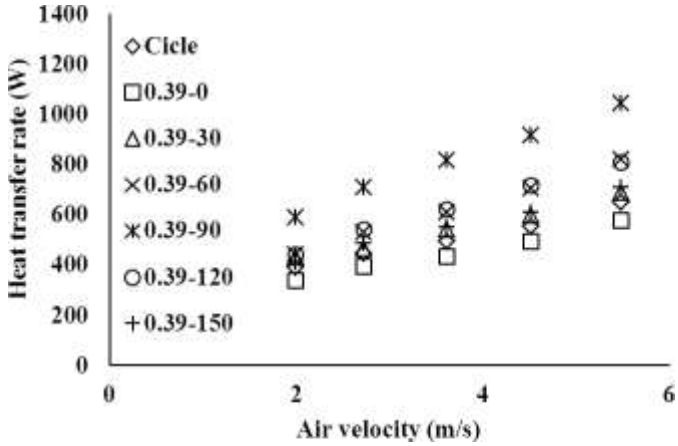


Fig. 8. Effect of attack angles on heat transfer of tube aspect ratio 0.39

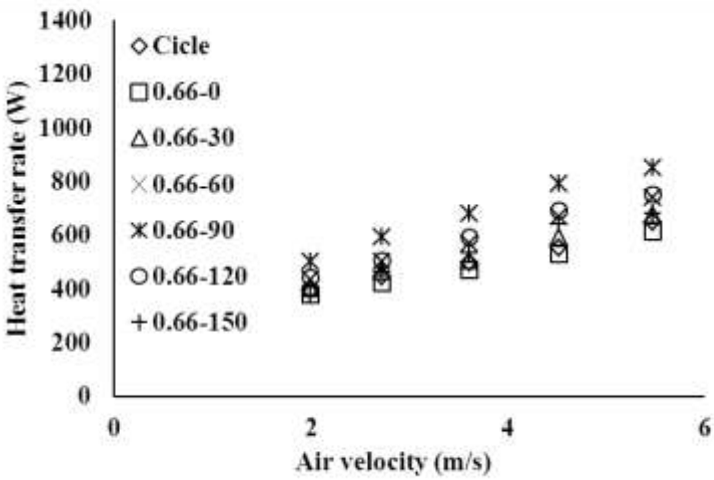


Fig. 9. Effect of attack angles on heat transfer of tube aspect ratio 0.66

Effect on pressure d op

The pressure drop on air side of tube aspect ratios have 0.18, 0.39 and 0.66 shown in the Fig.10, Fig.11 and Fig.12 respectively. The characteristics of pressure drop have similar to heat transfer characteristics as mentioned above. For the tube aspect ratios 0.18, 0.36 and 0.66, the pressure drop of 90° of attack angles higher than the 0°of attack angles by 17.8 times, 5.5 times and 1.8 times respectively. In this studied, the maximum and minimum pressure drop occurred at the 90°and 0 °of tube aspect ratio is 0.18. When considering relations between heat transfer and pressure drop of the heat exchangers founded the attack angle to give height heat transfer it give height pressure drop as well.

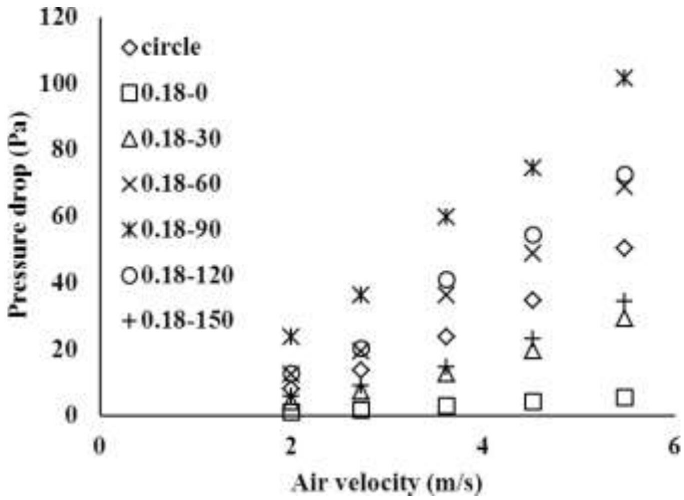


Fig. 10. Effect of attack angles on pressure drop of tube aspect ratio 0.18

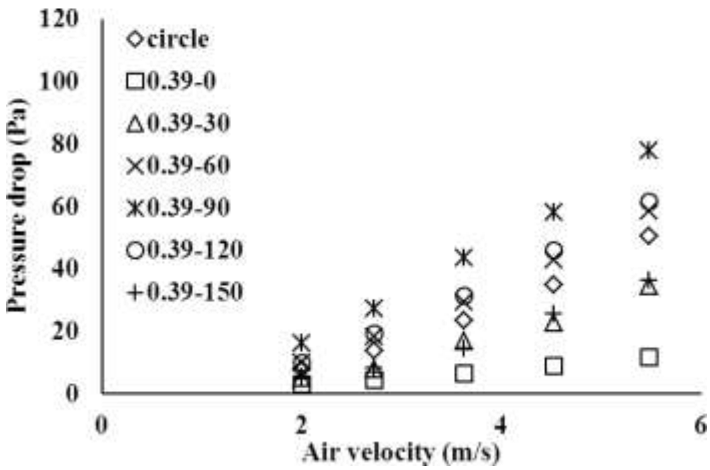


Fig. 11. Effect of attack angles on pressure drop of tube aspect ratio 0.39

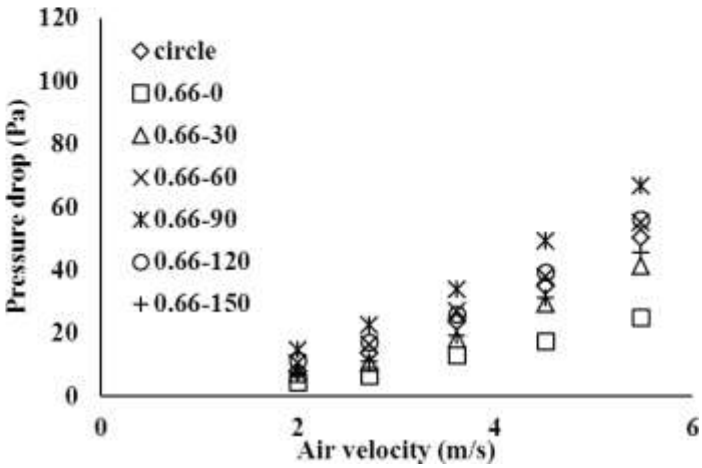


Fig. 12. Effect of attack angles on pressure drop of tube aspect ratio 0.66

Effect on performance

From above, the experiments indicated that the heat transfer can be promoted by increasing the attack angles from 0°to 90 °while the pressure drop of the air had increased as well. Thus, the thermal hydraulic performance is necessary to compare between heat transfer and pressure drop. Fig.13, Fig.14 and Fig.15 represent the thermal hydraulic performance of tube aspect ratios are 0.18, 0.39 and 0.66 respectively.

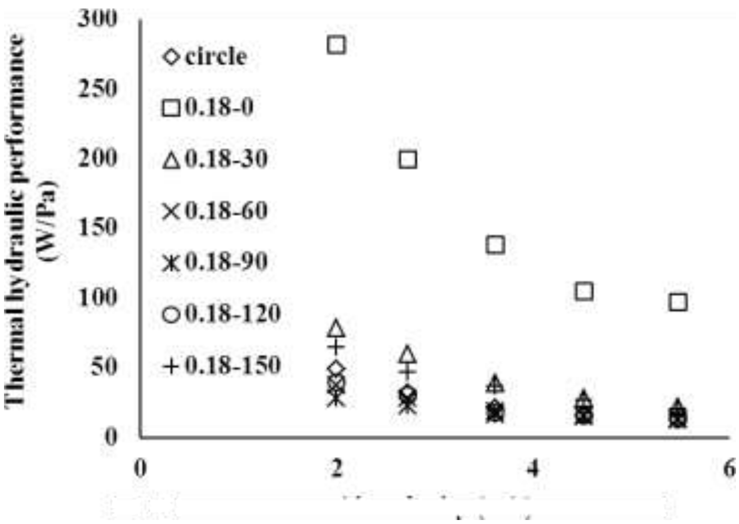


Fig 13. Effect of attack angles on performance of tube aspect ratio 0.18

The thermal hydraulic performance of all aspect ratios has decreased with increase the air velocity and angles of attack from 0 90°. The maximum thermal hydraulic performance occurred on the 0° of each tube aspect ratios while the 90° give the minimum thermal hydraulic performance, because the 0° of attack angle give very low pressure drop where compared another angles. At attack angles of 90°, the tube aspect ratio of 0.18 give the best thermal hydraulic performance, this greater than that of 0.39 and 0.66 by 1.5 times and 3.5 times respectively, while more than the circular tube 5.3 times.

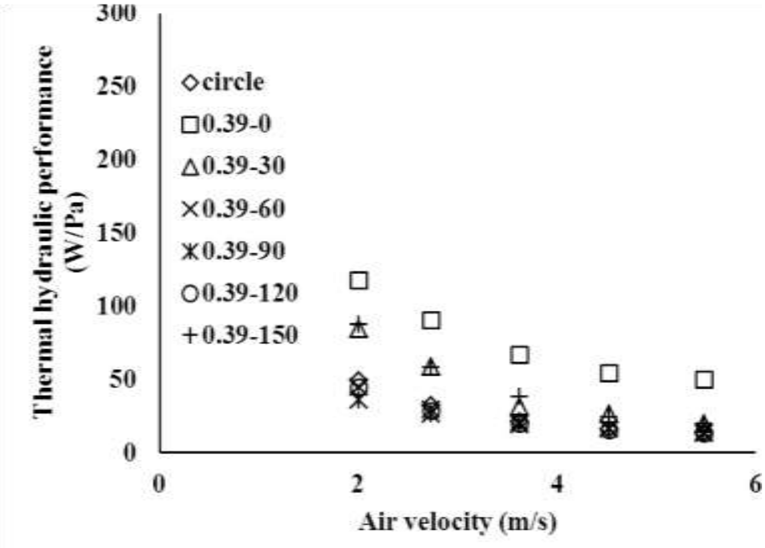


Fig a a k angles on performance of tube aspect ratio 0.39

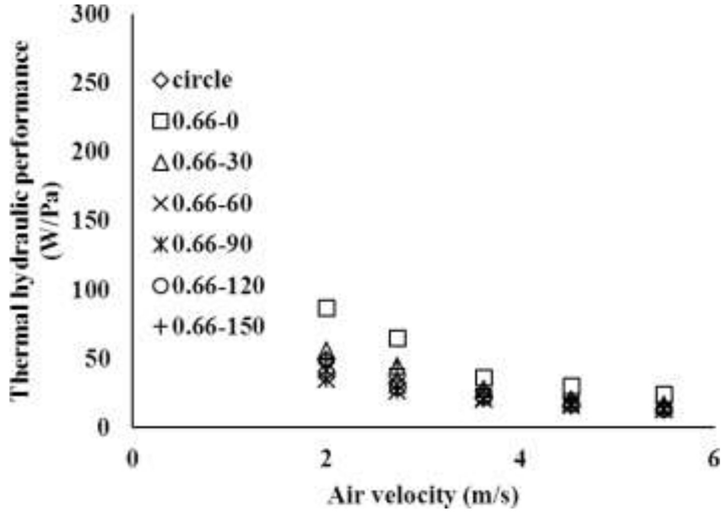


Fig. 15. Effect of attack angles on performance of tube aspect ratio 0.66

4. Conclusions

The effect of the attack angles on heat transfer and the pressure drop characteristics in the air side of fined cross flow heat exchangers with the flat tubes have different aspect ratios have been investigated experimentally. In the experiment, six attack angles (0°, 30°, 60°, 90°, 120°, and 150°) with four aspect ratios (0.18, 0.39, 0.66 and 1) have been performed. The main conclusions that can be drawn from the previous discussions are the following:

- The heat transfer rate and the pressure drop increase with increasing the air velocity and angles of attack ($0^\circ$to 90°) while the thermal hydraulic performance decreased.
- The attack angle of $90^\circ$of each tube aspect ratios give the maximum heat transfer and pressure drop, but provided lowest thermal hydraulic performance, while the best thermal hydraulic performance occurred at the $0^\circ$of attack angles.
- To change of angle of lower aspect had effected on thermal more than angle change of higher aspect ratios.

In case of value of thermal energy more than pumping energy, energy can saving by cross flow heat exchanger with flat tube of 90°attack angle.

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