Thermal Behavior in a Square Channel with Angled Ribs

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Abstract

This research presents a study of heat transfer enhancement and pressure drop in a square channel heat exchanger fitted with 45° and 90° inclined ribs. The rib to channel height ratio (*e/H*) of 0.1 and the rib pitch to channel height ratio, *PR*=1, 2 and 3 are introduced in the present work. The tested channel has a constant wall heat flux condition. The experiments are carried out by varying airflow rate in terms of Reynolds number ranging from 4000 to 26,000. The experimental result of heat transfer in the form of Nusselt number and pressure drop in terms of friction factor are compared between the channel mounted with inclined ribs and the smooth channel. The inclined rib with PR=1 gives higher heat transfer rate and friction factor than the one with PR=2, 3 and the smooth channel respectively the rib with 45° provides the higher value of heat transfer and pressure drop than 90° for all rib pitch ratio.

Keywords: inclined ribs, Nusselt number, friction factor, square channel.

1. Introduction

The increasing of heat transfer and thermal performance in a heat exchanger system is the target of heat transfer enhancement investigated. The techniques for increasing the thermal performance can be classified into two categories: active and passive methods. In the active methods, heat transfer is improved by supplying extra energy to the fluid or the equipment. Another one, the passive methods can be acquired without any external energy. Some examples of the passive methods are rough surfaces and insertion turbulator devices.

The use of ribs placing in the cooling channels or channel heat exchangers is one of the passive heat transfer enhancement technique in single-phase internal flows. Thus, the research work of fluid flow and heat transfer in ribbed channels has been rich so far. Several investigations have been conducted to study the effect of these parameters of ribs on heat transfer and friction factor for two opposite roughened surfaces. Han et al. [1, 2] studied experimentally the heat transfer in a square channel with different angled rib arrays on two walls for *P/e*=10 and *e/D*=0.0625. They reported that the angled ribs and 'V' ribs provided higher heat transfer enhancement than the continuous ribs and the highest value is at the 60° orientation amongst the angled ribs. For heating either only one of the ribbed walls or both of them, or all four channel walls, they reported that the former two conditions resulted in an increase in the heat transfer with respect to the latter one. For using broken ribs in a square channel with e/D=0.0625 and P/e=10. Murata [3] studied numerically the heat transfer distribution in a ribbed square channel with a large eddy simulation method. The ribs were placed at 60°, e/D=0.1 and P/e=10. Their numerical result indicated that the flow reattachment at the midpoint between ribs caused a significant increase in the local heat transfer. Lee et al. [4] studied experimentally the heat/mass transfer in rectangular channels with two different V-shaped ribs: continuous 60° V-shaped and multiple (staggered) 45° V-shaped ribs, and found that two pairs of counter-rotating vortices are generated in the channel. The effect of channel aspect ratio was more significant for the 60° V-shaped rib than for the multiple 45° V-shaped rib. Promvonge and Thianpong [5] studied the thermal performance of wedge ribs pointing upstream and downstream, triangular and rectangular ribs with e/H=0.3 and P/e=6.67 mounted on the two opposite walls of a channel with AR=15. They found that the inline wedge rib pointing downstream performed the highest heat transfer but the best thermal performance is the staggered triangular rib. Promvonge et al. [6] studied the numerical computations for three dimensional laminar periodic channel flows over a 45° inclined baffle mounted only on the lower square-channel wall and found that the 45° baffle with BR=0.4, the enhancement of heat transfer is about 2–3 fold higher than that for the 90° baffle while the friction loss is some 10–25% lower. Taslim et al. [7] reported that the heat transfer behavior in a ribbed square channel with three e/H ratios (e/H=0.083, 0.125 and 0.167) and a fixed P/e=10 using a liquid crystal technique. The average Nusselt number was increased with the rise in e/H ratio and the best one of the e/H ratios was found to lie between 0.083 and 0.125. Han and Zhang [8] also found that 60° broken 'V' ribs provide higher heat transfer at about 4.5 times the smooth channel and perform better than the continuous ribs. The performance of square, triangular and semi-circular ribs was experimentally investigated by Liou and Hwang [9, 10] using a real time Laser Holographic Interferometry to measure the local as well as average heat transfer coefficients. They found that the square ribs give the best heat transfer performance among them. This is contrary to the experimental result of Ahn [11] indicated that the triangular rib performs better than the square one.

2. Methodology

2.1 Experimental set-up and Materials

The experimental set-up used in this study is shown detail in Fig 1. A circular pipe was used for connecting a high-pressure blower to a settling tank, which an orifice flow meter was mounted in this pipeline while a square channel including a calm section and a test section was employed following the settling tank. The square channel was made of 3 mm thick an aluminum plate has a cross section of 45×45 mm^2 *(HxW)* and 3,000 mm length, it separate the test section 1,000 mm.

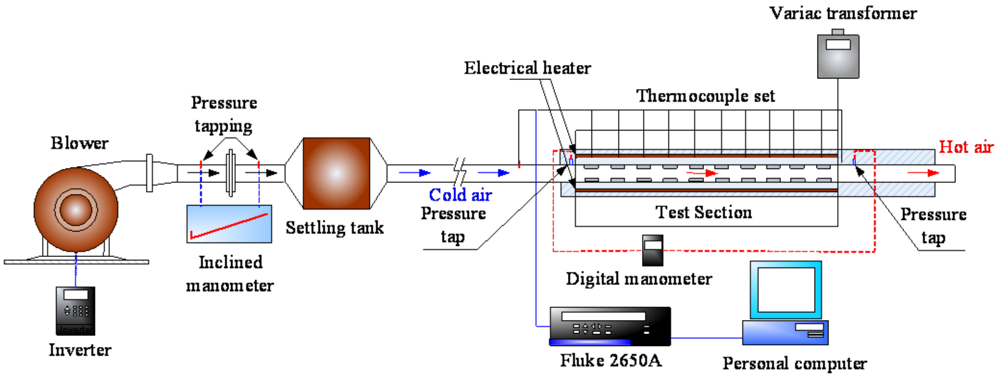


Fig. 1 Schematic diagram of experimental apparatus.

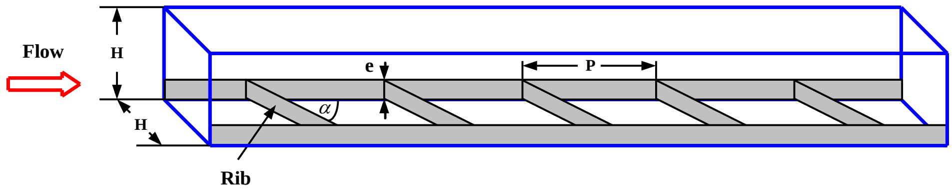


Fig 2. A detail of square channel with rids.

The pitch ratio (*PR*) is defined as the ratio of the length between ribs to the channel height *(P/H)*. The rib strip dimensions was 4.5 mm high *(e)* and 0.3 mm thick (*t*).

The test section consisted of the four walls. The AC power supply was the source of power for the plate-type heater, used for heating all walls of the test section in order to maintain a uniform surface heat flux. All details of square channel with rids are demonstrated in Fig 2.

2.2 Procedures

In the experiments, the cold air with ambient condition was passed through the test section by means of a 1.5 kW high speed blower, and its inlet flow rate was measured and controlled by an orifice plate pre-calibrated by using hot wire and vane-type anemometers (Testo 445) and a motor inverter, respectively. The pressure across the orifice was measured using inclined manometer. The surface temperatures (T_s) on the principal upper, lower and side walls were measured by 28 thermocouples type K located along the test section. To measure the inlet and outlet bulk temperatures by type K thermocouples were positioned upstream and downstream of the test channel. All thermocouples were type K, 1.5 mm diameter wire. All of the temperatures data from the system were recorded using a Fluke 2650A. Two static pressure taps were located at the top of the principal wall to measure axial pressure drops across the test section, used to evaluate average friction factor. These were located at the centre line of the channel. One of these taps is 50 mm upstream of the test channel and the other is 50 mm downstream. The pressure drop was measured by a digital differential pressure and a data logger (Testo 1445 and Testo 350XL) connected to the 2 mm diameter taps and recorded via a personal computer. Reynolds numbers for the air flowing through the test section were controlled in the range of 4000 to 26,000 for turbulent flow region.

2.3 Data Reduction

The target of this experiment is to investigate the Nusselt number, the friction factor and the thermal enhancement factor. The independent parameters are Reynolds number, pitch ratios and ribs fitted angle. The Reynolds number based on the channel hydraulic diameter is given by

$$\mathrm{Re} = UD_h / \nu \tag{1}$$

The average heat transfer coefficients are evaluated from the measured temperatures and heat inputs. With heat added uniformly to fluid (Q_{air}) and the temperature difference of wall and fluid ($T_w - T_b$), average heat transfer coefficient will be evaluated from the experimental data via the following equations:

$$Q_{air} = Q_{conv} = \dot{m}C_p(T_o - T_i) = VI \tag{2}$$

$$h = \frac{Q_{conv}}{A(\tilde{T}_s - T_b)} \tag{3}$$

in which,

$$T_b = (T_o + T_i)/2 \tag{4}$$

and

$$\tilde{T}_s = \sum T_s / 28 \tag{5}$$

The term A is the convective heat transfer area of the heated upper channel wall whereas $\tilde{T}_s$ is the average surface temperature obtained from local surface temperatures along the axial length of the heated channel. Then, average Nusselt number is written as:

$$Nu = \frac{hD_h}{k} \tag{6}$$

The friction factor is evaluated by:

$$f = \frac{2}{(L/D_h)} \frac{\Delta P}{\rho U^2} \tag{7}$$

where ΔP is a pressure drop across the test section and U is mean air velocity of the tube. All of thermo-physical properties of the air are determined at the overall bulk air temperature from Eq. (4). For a constant pumping power and the relationship between friction and Reynolds number can be expressed as:

$$(f\,\mathrm{Re}^3)_0 = (f\,\mathrm{Re}^3)$$

$$\mathrm{Re}_0 = \mathrm{Re}(f/f_0)^{1/3} \tag{8}$$

The thermal enhancement factor defined as the ratio of the heat transfer coefficient, *h* of an augmented surface to that of a smooth surface, h_0, at a constant pumping power:

$$\eta = \left.\frac{h_a}{h_0}\right|_{pp} = \left.\frac{Nu_a}{Nu_0}\right|_{pp} = \left(\frac{Nu_a}{Nu_0}\right)\left(\frac{f_a}{f_0}\right)^{-1/3} \tag{9}$$

3. Results and Discussion

3.1 Confirmatory of the smooth channel

The Nusselt number and friction factor obtained from the present smooth channel are compared with the foundation correlations of Gnielinski and Petukhov found in the open literature [12] for turbulent flow in ducts as showed below.

Correlation of Gnielinski,

$$Nu = \frac{(f/8)(\mathrm{Re} - 1000)\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \tag{10}$$

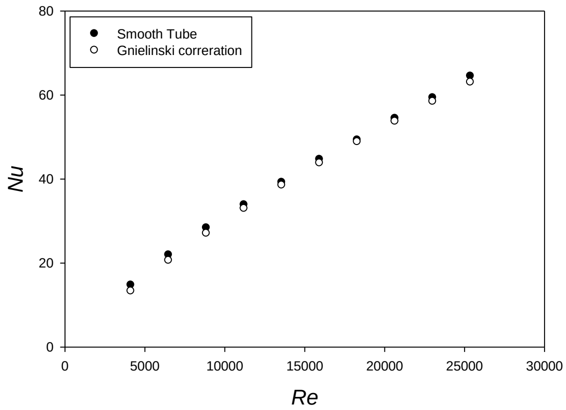
Correlation of Petukhov,

$$f = (0.79\ln \mathrm{Re} - 1.64)^{-2} \tag{11}$$

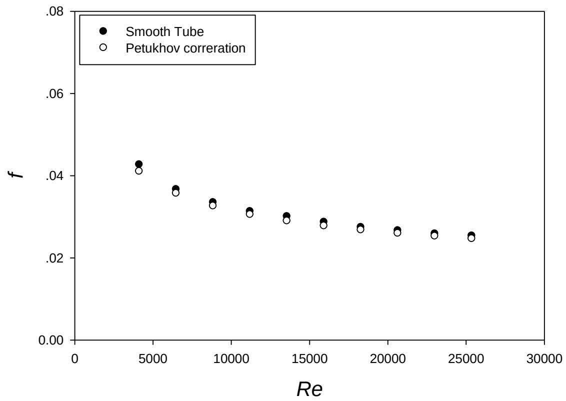
Figs 3a and b are shows a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (10) and (11) respectively. From the figure 3, it is obvious that the present data are in good agreement with those obtained from the Gnielinski and Petukhov equation for the whole range studied with deviation falls within ±3%.

3.2 Effect of ribs on Nusselt numbers

The experimental results of smooth square channel and square channel fitted ribs on the bottom side wall in heat transfer and flow friction factor in a uniform heat flux. The Nusselt numbers were presented under turbulent flow conditions for all cases are showed in Fig 4.



(a)



(b)

Fig. 3 Verification of (a) Nusselt number and (b) friction factor for smooth channel.

In the figure, the square channel fitted with ribs yield considerable heat transfer rate with a similar trend in comparison with the smooth channel and the Nusselt number increases with the rise of Reynolds number. This is because the ribs interrupt the development of the boundary layer of the fluid flow and increase the turbulence degree of flow. Fig 5 showed variation of Nusselt number ratio with Reynolds number. The Nusselt number ratio, Nu/Nu_0 is defined as a ratio of augmented Nusselt number in each case study to Nusselt number of smooth channel. In the figure, the Nusselt number ratio tends to slightly decrease with the rise of Reynolds number from 4000 to 26,000 for all of cases study. The mean Nusselt number ratio values are found to be about 1.93, 1.73 and 1.54 time over the smooth channel for using the 45° inclined ribs with PR = 1, 2 and 3, respectively. The 90° inclined ribs have given the mean Nusselt number ratio values are found to be about 1.75, 1.65 and 1.51 time at PR = 1, 2 and 3, respectively.

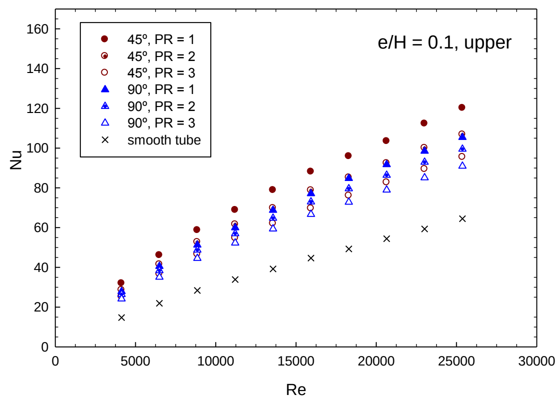


Fig. 4 Variation of Nusselt number with Reynolds number.

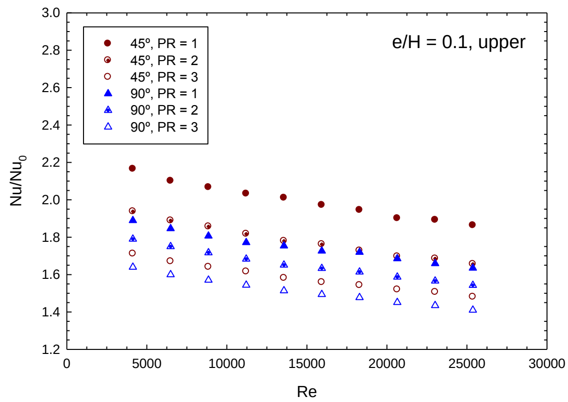


Fig. 5 Variation of Nusselt number ratio, Nu/Nu_0 with Reynolds number.

3.3 Effect of ribs on friction factor

The results are presented in Fig 6 that the effect of using the ribs turbulators on the pressure drop across the tested channel as showed in terms friction factor. In the figure, it is apparent that the use of ribs turbulators leads to a substantial increase in friction factor over the smooth channel.

The variation of friction factor ratio value with Reynolds number for this investigated is presented in Fig 7. In the figure, the friction factor ratio value is found to be increased with the rise of Reynolds number.

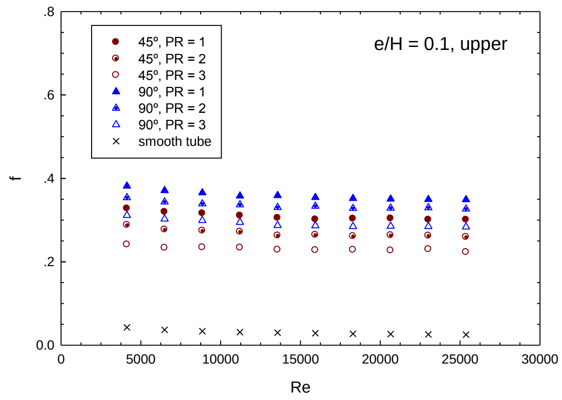


Fig. 6 Variation of friction factor with Reynolds number

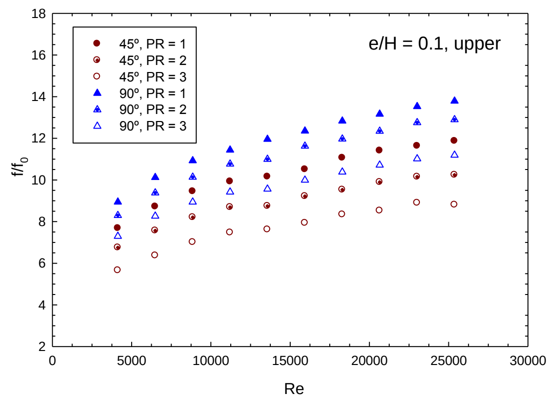


Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds number.

The mean friction factor ratio values are found to be about 11.05, 9.62 and 8.24 time over the smooth channel for using the 45° inclined ribs with PR = 1, 2 and 3, respectively. The 90° inclined ribs have given the mean friction factor ratio values are found to be about 11.90, 11.17 and 9.68 time at PR = 1, 2 and 3, respectively. The 90° inclined ribs were showed the greater friction factor than the 45° inclined ribs. This result indicates that the use of low blockage ratio can help to reduce the pressure loss considerably.

3.4 The thermal enhancement factor

In the present work, the effectiveness of heat transfer enhancement in terms of thermal performance factor is defined using the Nusselt number and friction factor in the tube fitted with the enhancement device as shown in Eq. (9). The thermal performance factor for the channel with various ribs fitted is compared at the same pumping power in Fig 8. Apparently, the performance factor tends to decrease with the increasing Reynolds number. The mean thermal enhancement factor values are between 0.71 – 0.87 times. The 45° inclined ribs with PR = 1, is found to give a maximum thermal enhancement factor of 1.10 at low Reynolds number.

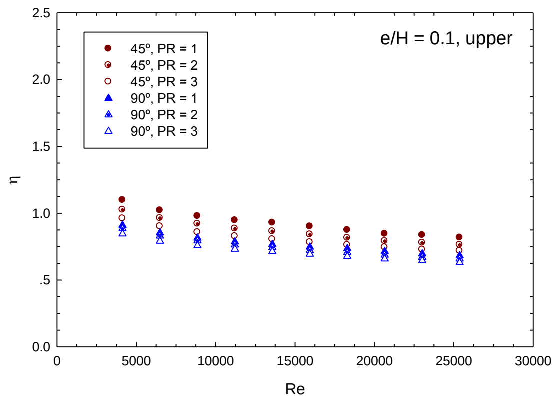


Fig. 8 Variation of thermal enhancement factor with Reynolds number.

4. Conclusions

In this paper, the heat transfer enhancement, friction factor and thermal performance factor behaviors in a square channel for the turbulent regime, Reynolds number ranging from 4000 to 26,000. The experimental results are compared between the channel mounted with inclined ribs and the smooth channel. The channel fitted rib gives higher heat transfer rate and friction factor than the smooth channel. The inclined rib with PR=1 gives higher heat transfer rate and friction factor than one with PR= 2, 3 and the smooth channel respectively the rib with 45° provides the higher value of heat transfer and pressure drop than 90° for all rib pitch ratio.

5. Acknowledgement

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