Effect of Inclination Angle on Heat Transfer in a Square Channel with U-Shaped Ribs

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Abstract

The paper presents a study of heat transfer in a heat exchanger square channel inserted with 20°, 30° and 45° angled, U-shaped ribs. The test channel has a square section with uniform wall heat flux conditions. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the channel ranging from 4000 to 25,000. The U-Shaped ribs with axial pitch equal to three times of channel height and two rib-to-channel height ratios e/H = 0.1 and 0.2 are introduced. The experimental result of heat transfer in terms of Nusselt number and pressure loss in terms of friction factor are compared between the inserted channel and the smooth channel. The U-shaped rib with the attack angle of 45° gives higher heat transfer and friction factor than the one with the attack angle of 30°, 20° and the smooth channel. It is worth noting that the heat transfer and pressure loss for the rib with e/H = 0.2 provides higher Nusselt number and friction factor than these with e/H = 0.1 for all rib angles.

***Keywords**:* turbulator, angle rib, square channel, U-Shaped rib, pressure loss.

1. Introduction

Heat transfer coefficient in a duct flow can be increased by roughening the wall of the duct. Over the past decades, many engineering techniques have been devised for enhancing the rate of convective heat transfer from the wall surface. The uses of turbulators such as ribs, fins, grooves and baffles are often employed in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency.

Several investigations have been carried out to study the effect of these parameters of turbulators on heat transfer and friction factor for roughened surface. Taslim et al. [1] conducted measurements of the heat transfer in a straight 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. Various staggered rib configurations were studied, especially for the angle of 45°. Experimental data showed a significant increase in average Nusselt number for the increase of the e/H ratio. Murata and Mochizuki [2] 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. Chandra et al. [3] carried out measurements on heat transfer and pressure loss in a square channel with continuous ribs on four walls. Ribs were placed superimposed on walls at the rib height ratio e/D = 0.0625; and the rib pitch ratio, P/e = 8. They reported that the heat transfer augmentation found to increase with the rise in the number of ribbed walls was decreased with increasing Reynolds number while the friction factor augmentation increased with both cases. Promvonge and Thianpong [4] 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. Thianpong et al. [5] again investigated the thermal behaviors of isosceles triangular ribs attached on the two opposite channel walls with AR = 10 and suggested the optimum thermal performance of the staggered ribs could be at about e/H = 0.1 and P/H = 1.0. Promvonge et al. [6,7] studied the numerical computations for three dimensional laminar periodic channel flows over a 45° inclined baffle mounted on only the lower wall and on both the upper and lower channel walls. They found that for the baffle mounted on one wall, the BR = 0.4 gives the enhancement of heat transfer at about 2–3 fold higher than the 90° baffle while the friction loss is some 10–25% lower. For 45° inline baffles placed on two opposite walls, the BR = 0.2 provides the enhancement of heat transfer at about 150–850% higher than the 90° inline baffles but the friction loss at some 10–150% below. An extensive literature review over hundred references on various rib turbulators was reported by Varun et al. [8].

The study on U-shaped rib in square channels has never been reported since most ribs found in the literature are square, rectangular, triangular and wedge shaped-ribs. In the present work, the experimental data presented in turbulent channel flows with 20°, 30° and 45° angled, U-shaped ribs on the lower and side square-channel walls are conducted with the main aim being to study the changes in the flow pattern and heat transfer performance. The use of the U-shaped rib attached in tandem is expected to create a longitudinal vortex flow throughout the tested channel to better mixing of flows between the core and wall regimes leading to higher heat transfer rate.

2. Experimental Setup

A schematic diagram of the experimental apparatus is presented in Figure 1 while the detail of 45° U-shaped ribs placed on the lower and side channel walls is depicted in Figure 2. In Figure 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 configuration was characterized by the channel height, H of 45 mm and axial pitch equal to three times of channel height (pitch ratio, PR = 3) and the attack angle of 20°, 30° and 45°. The overall length of the channel was 3000 mm. The test square channel made of 3 mm thick aluminum plates has a cross section of 45×45 mm^2 and 1000 mm long (L). The rib strip dimensions were 4.5 and 9 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.

Air as the tested fluid in both the heat transfer and pressure drop experiments, was directed into the systems by a 1.45 kW high-pressure blower. The operating speed of the blower was varied by using an inverter to provide desired air flow rates. The flow rate of air in the systems was measured by an orifice plate pre-calibrated by using hot wire and vane-type anemometers (Testo 445). The pressure across the orifice was measured using inclined manometer. In order to measure temperature distributions on the principal upper, lower and side walls, twenty eight thermocouples were fitted to the walls. The thermocouples were installed in holes drilled from the rear face and centered of the walls with the respective junctions positioned within 2 mm of the inside wall and axial separation was 100 mm apart. To measure the inlet and outlet bulk temperatures, two thermocouples were positioned upstream and downstream of the test channel. All thermocouples were K type, 1.5 mm diameter wire. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2650A) and then recorded via a personal computer.

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.

To quantify the uncertainties of measurements, the reduced data obtained experimentally were determined. The uncertainty in the data calculation was based on ref. [9]. The maximum uncertainties of non-dimensional parameters were ±5% for Reynolds number, ±8% for Nusselt number and ±10% for friction. The uncertainty in the axial velocity measurement was estimated to be less than ±7%, and pressure has a corresponding estimated uncertainty of ±5%, whereas the uncertainty in temperature measurement at the channel wall was about ±0.5%.

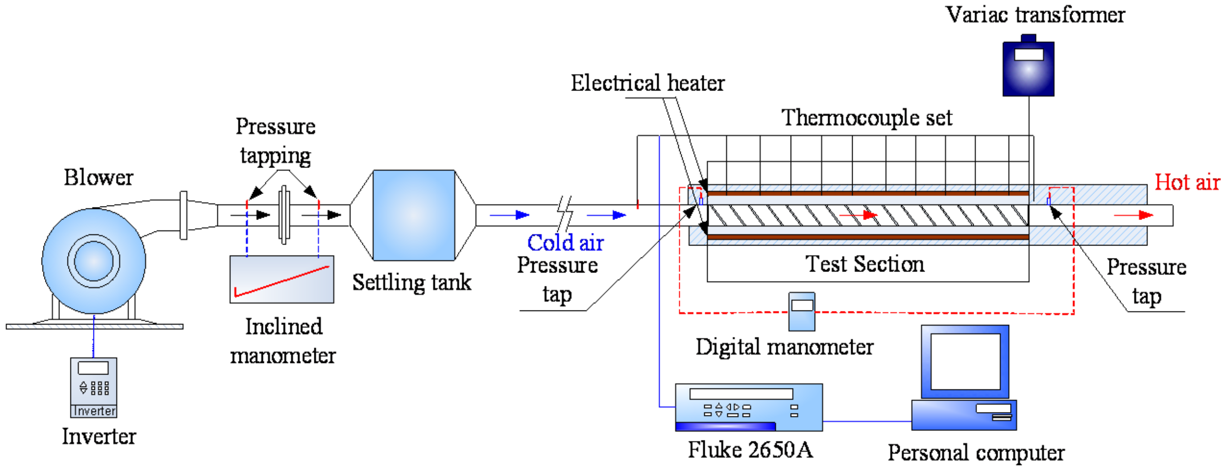


Fig. 1 Schematic diagram of experimental apparatus.

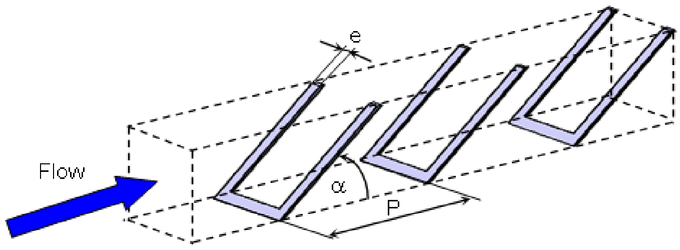


Fig. 2 Test section with U-Shaped rib inserts.

3. Data Reduction

The goal of this experiment is to investigate the Nusselt number in ribbed channels. The independent parameters were Reynolds number and rib types. The Reynolds number based on the channel hydraulic diameter (D_h) is given by:

$$\mathrm{Re} = UD_h / \nu \quad (1)$$

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:

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

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

in which,

$$T_b = (T_o + T_i)/2 \quad (4)$$

and

$$\tilde{T}_s = \sum T_s / 28 \quad (5)$$

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:

$$Nu = \frac{hD_h}{k} \quad (6)$$

The friction factor is evaluated by:

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

The thermal enhancement factor, η, defined as the ratio of the, h of an augmented surface to that of a smooth surface, h_0, at a constant pumping power, wedd [10].

$$\eta = \left.\frac{h_a}{h_0}\right|_{pp} = \left.\frac{Nu_a}{Nu_0}\right|_{pp} = \left(\frac{Nu_a}{Nu_0}\right)\left(\frac{f_a}{f_0}\right)^{-1/3} \qquad (8)$$

4. Results and Discussion

4.1 Verification of smooth channel

The present experimental results on heat transfer and friction characteristics in a smooth wall channel are first validated in terms of Nusselt number and friction factor. The Nusselt number and friction factor obtained from the present smooth channel are, respectively, compared with the correlations of Gnielinski and Petukhov found in the open literature [11] for turbulent flow in ducts.

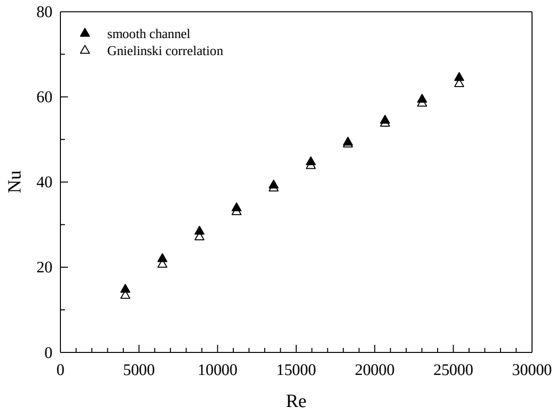
Correlation of Gnielinski,

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)} \qquad (9)$$

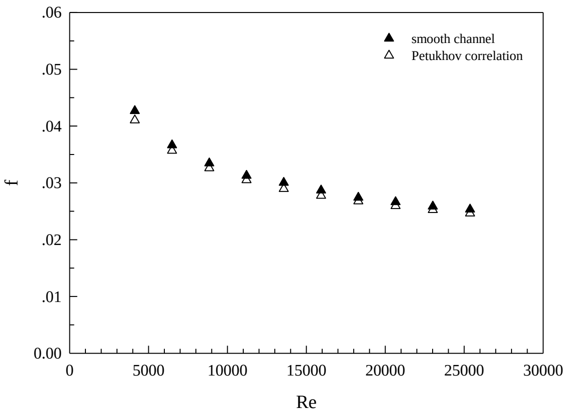
Correlation of Petukhov,

$$f = (0.79\ln \mathrm{Re} - 1.64)^{-2} \qquad (10)$$

Figure 3a and b shows, respectively, a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (9) and (10). In the figure, the present results agree very well within ±3% for both friction factor and Nusselt number correlations.



(a)



(b)

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

4.2 Effect of blockage ratio (e/H)

The present experimental results on heat and flow friction characteristics in a uniform heat flux channel with U-shaped rib, placed on the lower and side wall are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for all cases are presented in Figure 4. In the figure, the Nusselt number increases with the rise of Reynolds number. This is because the U-shaped rib turbulators interrupt the development of the boundary layer of the fluid flow and increase the turbulence degree of flow. The rib along with the U-shaped rib of higher α value also provides higher heat transfer than that of lower α value, as can be seen in Figure 4. It is worth noting that the heat transfer coefficient for rib-to-channel height ratio, e/H = 0.2 is considerably higher than those for e/H = 0.1. This is caused by higher blockage of using e/H = 0.2 interrupting the flow and diverting its direction thus promoting high levels of mixing over the others.

The effect of using the rib turbulators on the isothermal pressure drop across the tested channel is presented in Figure 5. The variation of the pressure drop is shown in terms of friction factor with Reynolds number. In the figure, it is apparent that the use of rib turbulators leads to a substantial increase in friction factor over the smooth channel. This can be attributed to flow blockage, higher surface area and the act caused by the reverse flow. As expected, the friction factor of rib-to-channel height ratio, e/H = 0.2 is considerably higher than e/H = 0.1, especially for higher the attack angle. For the rib-to-channel height ratio e/H = 0.2, the losses mainly come from the dissipation of the dynamical pressure of the air due to high viscous losses near the wall, to higher friction of increasing surface area and the blockage ratios because of the presence of the ribs.

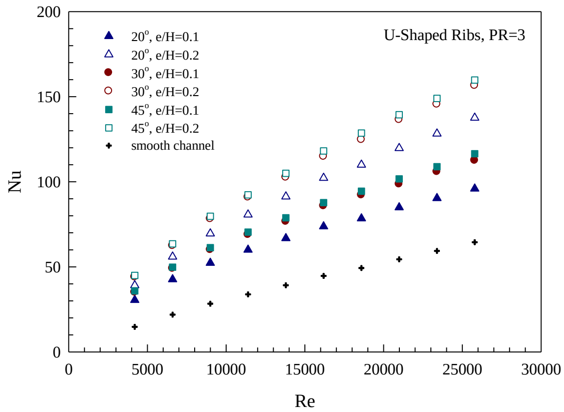


Fig. 4 Variation of Nusselt number with Reynolds number for various rib heights.

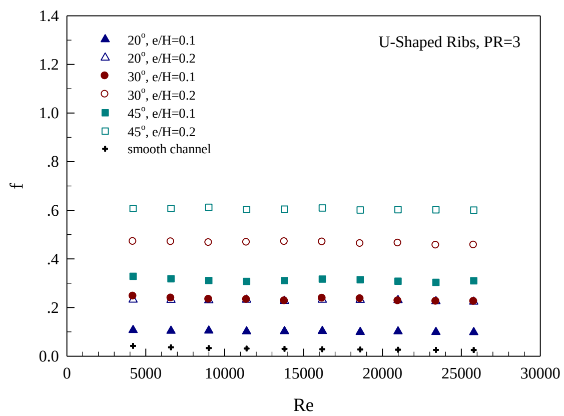


Fig. 5 Variation of friction factor with Reynolds number for various rib heights.

4.3 Performance evaluation

The Nusselt number ratio, Nu/Nu_0, defined as a ratio of augmented Nusselt number to Nusselt number of smooth channel, plotted against the Reynolds number value is displayed in Figure 6. In the figure, the Nusselt number ratio tends to slightly decrease with the rise of Reynolds number from 4000 to 25,000 for all cases of U-shaped rib. The average Nu/Nu_0 values for the U-shaped rib with α = 20°, 30° and 45° are, respectively, around 2.3, 2.6 and 2.7 for the e/H = 0.2 and about 1.7, 1.9 and 2.0 for the e/H = 0.1 one.

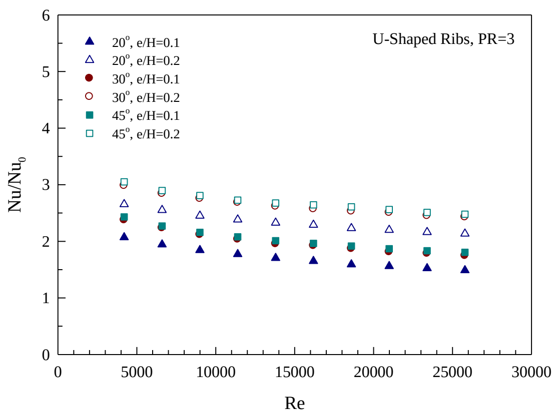


Fig. 6 Variation of Nusselt number ratio, Nu/Nu_0 with Reynolds number.

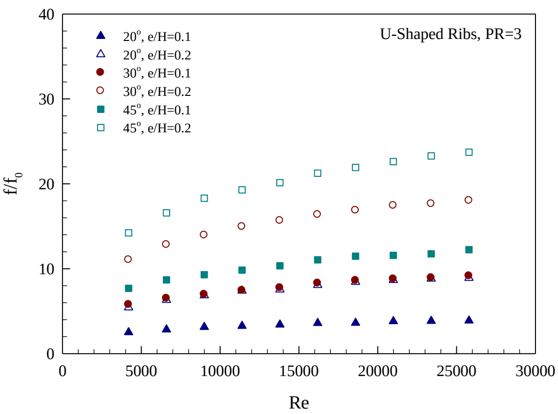


Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds number.

The variation of isothermal friction factor ratio, f/f_0 value with Reynolds number for U-shaped rib is also depicted in Figure 7. In the figure, the friction factor ratio value is found to increase with the rise of Reynolds number. The U-shaped rib with e/H = 0.2 and the higher attack angle provide a considerable increase in the friction factor ratio than the e/H = 0.1 and the lower angle under the same operating condition. The average f/f_0 values for the U-shaped rib with α = 20°, 30° and 45° are, respectively, around 7.7, 15.5 and 20.1 for the e/H = 0.2 and about 3.4, 7.6 and 10.4 for the e/H = 0.1 in the range of Re studied. This implies that the use of lower angle of attack and the rib e/H = 0.1 can help to reduce considerably the pressure loss.

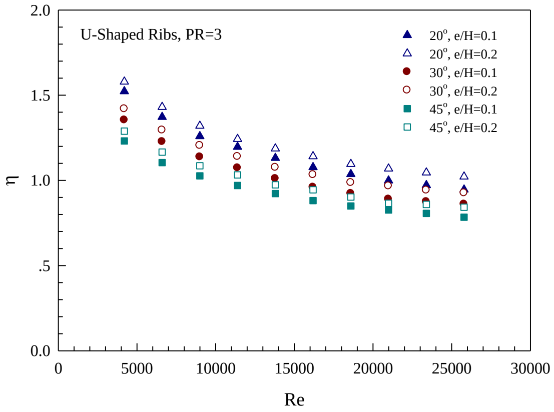


Fig. 8 Variation of thermal enhancement factor with Reynolds number.

Figure 8. shows the variation of the thermal enhancement factor η with Reynolds number for all cases. For all, the data obtained by Nusselt number and friction factor values are compared at similar pumping power. The thermal enhancement factor tends to decrease with the rise of Reynolds number values for all. It is seen that the e/H = 0.2 and the lower attack angle provide a considerable increase in the η than the e/H = 0.1 and the higher angle under the same operating condition. The average η values for the U-shaped rib with α = 20°, 30° and 45° are, respectively, around 1.2, 1.07 and 1.0 for the e/H = 0.2 and about 1.15, 1.02 and 0.94 for the e/H=0.1 in the range of Re studied. The results are for Reynolds number of 4000-25,000 for the α = 20° U-shaped rib.

5. Conclusions

An experimental study has been carried out to investigate the airflow friction and heat transfer characteristics in a square channel fitted with different attack angle turbulators for the turbulent regime, Reynolds number from 4000-25,000. The use of the U-Shaped rib with e/H = 2 and α = 45° causes a very high pressure drop increase and also provides considerable heat transfer augmentations, Nu/Nu_0 = 2.7. Nusselt number augmentation tends to increase with the rise of Reynolds number. In comparison, the use of rib leads to the higher heat transfer rate but the α = 20° provides the higher thermal enhancement factor due to lower friction loss.

6. Acknowledgement

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7. References

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