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Experimental Study on Heat Transfer and Friction Characteristics in Channel with Wavy Baffles

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Abstract

The paper presents an experimental study on heat transfer and turbulent flow friction characteristics in a rectangular channel fitted periodically with triangular and rectangular wavy baffles. The experiments are conducted by varying the airflow rate in order to obtain the Reynolds number from 5000 to 23,000 in the test section. The upper plate of the channel is uniformly heated with a constant heat-flux. The channel has an aspect ratio, AR = 10 and a height, H = 30 mm. The wavy baffles with a transverse pitch equal to two times of channel height (2H) and a single attack angle, α=30° are mounted on the lower plate only. The effect of three baffle- to channel-height ratios, (e/H= 0.2, 0.3, and 0.4) on the heat transfer in terms of Nusselt number and the friction loss in the form of friction factor is experimentally investigated. The results show a significant effect of the baffles on the heat transfer and friction loss over the smooth channel. The rectangular wavy baffle at e/H =0.4 provides the highest heat transfer and friction factor but the triangular wavy baffle at e/H =0.4 yields the best thermal performance.

***Keywords**:* rectangular channel; wavy baffles; friction loss; aspect ratio; attack angle.

1. Introduction

The use of turbulent promoters on the heat exchanger surfaces has been extensively in attention because it provides a substantial heat transfer enhancement. However, the heat transfer enhancement always accompanies a higher pressure drop penalty. For decades, heat transfer characteristics in channels with baffle-roughened walls have already been widely investigated. Several experiments have been conducted toward establishing an optimal baffle geometry, which gives the best heat transfer performance for a given pumping power or flow rate. Relevant geometric parameters such as channel aspect ratio, attack angle, flow blockage ratio (or baffle height), rib shape, and relative arrangement of the baffles, are found to affect pronouncedly on both local and overall heat transfer coefficients by enhancing turbulence and/or adding heat transfer surface area. Some of these effects have been carried out by several investigations [1-7]. The majority of these studies examined the overall heat transfer of the combination of ribs and the area between them. Some investigated the heat transfer on the surfaces between the ribs. The heat transfer of the ribs themselves has not been widely investigated. The ribs are served as turbulence promoters (called turbulators). In heat transfer experiments, the metallic ribs attached to the heat transfer surfaces are thermally active. Consequently, the heat transfer augmentation of the ribbed wall comes not only from the enhanced turbulence but also from the increased heat transfer area.

Several experiments were performed by placing the angled ribs according to a crossed arrangement or V-shaped rib arrangement with the apex of V pointing upstream or downstream of the flow. Results presented by Gao and Sunden [5] showed that, for a rectangular (AR = 8) channel ribbed on both sides, with e/D_h = 0.06, P/e = 10 and Re = 1000-6000, the 60° V-shaped ribs produced higher heat transfer enhancement when pointing downstream of the main flow direction, seemingly contradicting results of [3]. Sripattanapipat and Promvonge [7] conducted a numerical study of laminar periodic flow and thermal behaviors in a two dimensional channel fitted with staggered diamond-shaped baffles and reported that the diamond baffles with half apex angle of 5-10° provided slightly better thermal performance than the flat baffle. However, the increase in heat transfer was accompanied by an increase in the resistance of the fluid flow. An extensive literature review over hundred references on various rib turbulators was reported by Varun et al. [8].

Thus, the main objective of this study is to examine the heat transfer and friction characteristics in a rectangular channel with rectangular and triangular wavy baffles placed on the lower plate while the heated upper plate of the channel is used as the absorber plate. The experimental work is conducted for various blockage ratios (e/H = 0.20, 0.30, and 0.40) with two pitch ratios (PR = 2) and an attack angle of 30°. Experimental results using air as the test fluid are presented in turbulent channel flows in a range of Reynolds number from 5000 to 23,000.

2. Experimental setup

A schematic diagram of the experimental apparatus is shown in Fig. 1. In the figure, 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 rectangular duct including a calm section and a test section was employed following the settling tank. The rectangular duct configuration was characterized by the channel height, H = 30 mm and a longitudinal pitch equal to two times channel height (pitch ratio, PR = 2) and the rib attack angle of 30°. The overall length of the channel was 2000 mm which included length of the test section, L=600 mm with the channel width, W=300 mm. Each baffle was fabricated from the 0.3 mm thick aluminum strips. The baffle dimensions are 6, 9 and 12 mm high (e) and 0.3 mm thick (t) as shown in Fig. 2.

The channel test section is consisted of the two parallel walls as shown in Fig. 2. The AC power supply was the source of power for the plate-type heater, used for heating the upper plate of the test section only in order to maintain a uniform surface heat flux. A conducting compound was applied to the heater and the principal upper wall to reduce contact resistance. Special wood bars, which have a much lower thermal conductivity than the metallic wall, were placed on the inlet and exit ends of the upper and lower walls to serve as a thermal barrier at the inlet and exit of the test section.

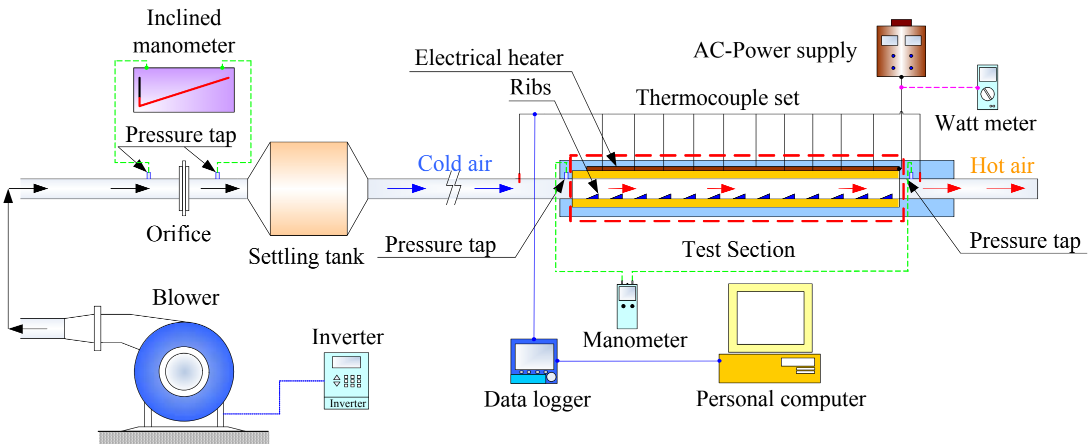

Fig. 1. Schematic diagram of experimental apparatus.

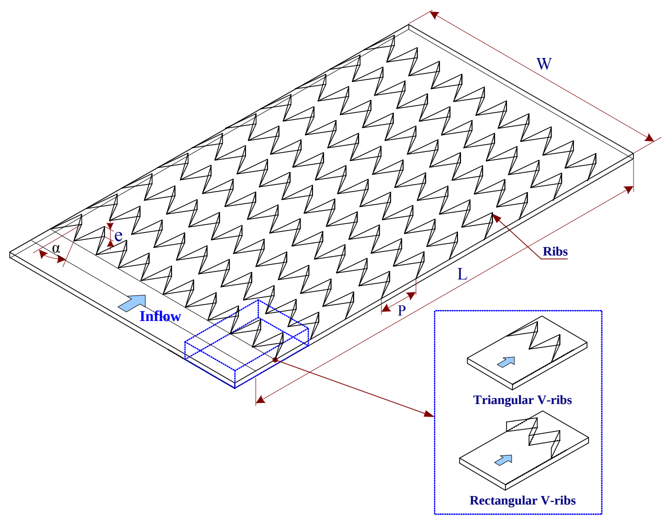

Fig. 2. Test section with rectangular and triangular wavy baffles.

Air as the tested fluid in 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 rate. 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 wall, twelve thermocouples were fitted to the wall. The thermocouples were attached 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 40 mm apart. To measure the inlet bulk temperature, two thermocouples were positioned upstream of duct inlet. All thermocouples were *K* type, 0.5 mm diameter wire. The thermocouple voltage outputs were fed into a data acquisition system and then recorded via a personal computer.

Two static pressure taps were located at the top of the principal channel 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 the test channel and the other is 30 mm downstream the test channel. The pressure drop was measured by a digital differential pressure connected to the 2 mm diameter taps.

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%.

3. Data reduction

The goal of this experiment is to investigate the Nusselt number in the channel. The Reynolds number based on the channel hydraulic diameter, D_h, is given by

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

where U and ν are the mean air velocity of the channel and kinematics viscosity of air, respectively. The average heat transfer coefficient, h, is 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), the average heat transfer coefficient will be evaluated from the experimental data via the following equations:

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

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

in which,

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

and

$$\tilde{T}_s = \sum T_s / 10 \,. \tag{5}$$

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, T_s, along the axial length of the heated channel. The terms $\dot{m}$, C_p, V and I are the air mass flow rate, specific heat, voltage and current, respectively. Then, average Nusselt number, Nu, is written as:

$$\mathrm{Nu} = \frac{hD_h}{k} \,. \tag{6}$$

The friction factor, f, is evaluated by:

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

where ΔP is a pressure drop across the test section and ρ is density. All of thermo-physical properties of the air are determined at the overall bulk air temperature, T_b, from Eq. (4).

For equal pumping power,

$$\left(\dot{V}\Delta P\right)_0 = \left(\dot{V}\Delta P\right), \quad (8)$$

in which $\dot{V}$ is volumetric airflow rate and the relationship between friction and Reynolds number can be expressed as:

$$\left(f\,\mathrm{Re}^3\right)_0 = \left(f\,\mathrm{Re}^3\right),$$
$$\mathrm{Re}_0 = \mathrm{Re}\left(f/f_0\right)^{1/3}. \quad (9)$$

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

$$\mathrm{TEF} = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right|_{pp} = \left(\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right)\left(\frac{f}{f_0}\right)^{-1/3}. \quad (10)$$

4. Result 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 (Nu) and friction factor (f). The Nu and f obtained from the present smooth channel are, respectively, compared with the correlations of Dittus-Boelter and Blasius found in the open literature [10] for turbulent flow in ducts.

Correlation of Dittus-Boelter,

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \quad \text{for heating.} \quad (11)$$

Correlation of Blasius,

$$f = 0.316\mathrm{Re}^{-0.25} \text{ for } 3000 \le \mathrm{Re} \le 20{,}000. \quad (12)$$

Fig. 3a and 3b shows, respectively, a comparison of Nu and f obtained from the present work with those from correlations of Eqs. (11) and (12). In the figure, the present results agree very well within $\pm$ 6% for Nu and f correlations.

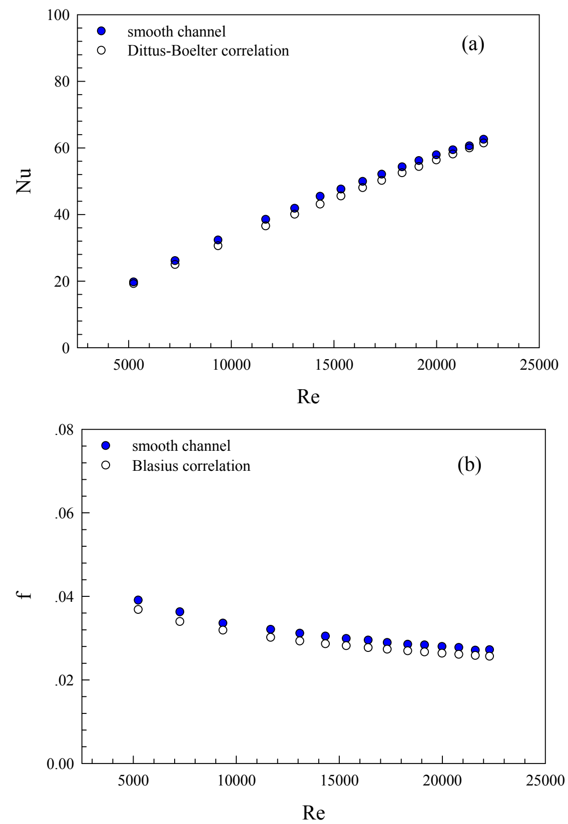

Fig. 3. Verification of (a) Nusselt number and (b) friction factor for smooth channel.

4.2 Effect of geometry

The experimental results on heat transfer and friction behavior in a uniform heat flux channel fitted with rectangular and triangular wavy baffles placed only on the lower plate are presented in the form of Nusselt number (Nu) and friction factor (f). The Nu values obtained under turbulent flow condition for all cases are depicted in Fig. 4. In the figure, the rectangular and triangular wavy baffles provide considerable heat transfer increases with a similar trend in comparison with the smooth channel. The Nu increases with the rise in Re values. This is because the baffles interrupt the development of the boundary layer thickness of the fluid flow and help to increase the turbulence degree of the flow. It is worthy to note that the rectangular baffle at e/H=0.4 provides the highest Nu while the one at e/H=0.3 performs higher than the one at e/H=0.2. The rectangular baffle is also found to perform better than the triangular wavy baffle at a given e/H ratio. This caused that the rectangular baffle provides a higher flow blockage, interrupts the flow and diverts its direction and thus promotes higher level of flow mixing over the other. A close investigation reveals that the rectangular baffle at *e*/*H*=0.4 produces the highest heat transfer rate.

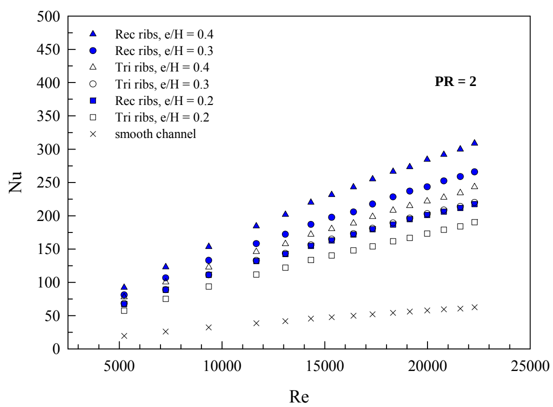

Fig. 4. Variation of Nusselt number with Reynolds number for various e/H values.

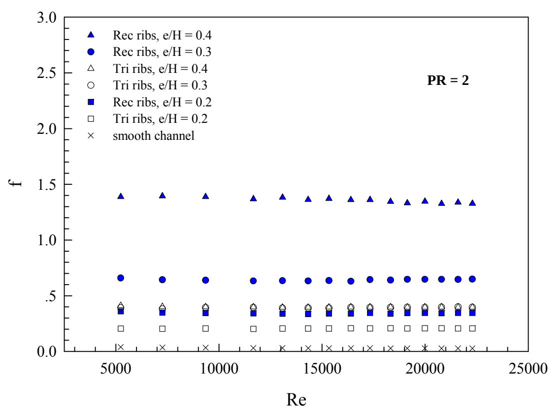

Fig. 5. Variation of friction factor with Reynolds number for various e/H values.

The effect of using the rectangular and triangular wavy baffles on the isothermal pressure drop across the test channel is displayed in Fig. 5. The variation of the pressure drop is shown in the form of friction factor with Reynolds number. In the figure, it is apparent that the use of triangular baffles leads to a substantial increase in friction factor over the smooth channel with no baffle. As expected, the friction factor of the rectangular wavy baffle at e/H=0.4 is considerably higher than that at e/H=0.3 and 0.2; and than that of the triangular wavy baffles at all e/H values. For the rectangular wavy baffle at e/H = 0.4, the increase in friction factor is in the range of 197- 368% above the one at e/H = 0.3 and 0.2 and of 320 - 610% over the triangular baffles, depending on Reynolds number and e/H values.

The Nusselt number ratio, Nu/Nu_0, defined as a ratio of augmented Nu to Nu of smooth channel plotted against the Re value is depicted in Fig. 6. In the figure, the Nu/Nu_0 tends to be nearly uniform with the rise of Re for all cases. The Nu/Nu_0 values for the rectangular and triangular wavy baffles at e/H=0.4, 0.3 and 0.2 are found to be 4.85, 4.16 and 3.44; and 3.83, 3.67 and 2.96, respectively.

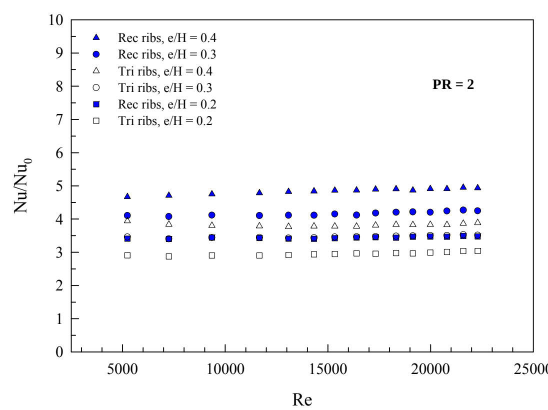

Fig. 6. Variation of Nu/Nu_0 with Reynolds number.

The variation of the friction factor ratio, f/f_0 with Re values for rectangular and triangular wavy baffles is shown in Fig. 7. The f/f_0 values are increased with the increase in the Re and the e/H values. The mean f/f_0 values for the rectangular and triangular wavy baffles at e/H=0.4, 0.3 and 0.2 are around 44.92, 21.23, and 11.38; and 13.18, 13.03 and 6.84, respectively. The result shows that using lower e/H help to reduce considerably the friction loss.

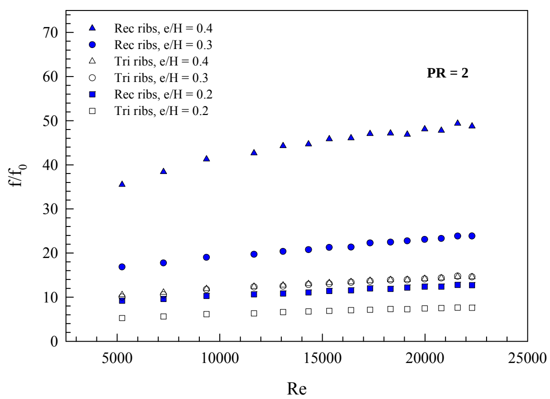

Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds Number.

4.3 Performance evaluation

Fig. 8 shows the variation of the thermal enhancement factor (TEF) with Reynolds number. For all, the data obtained from the Nu and *f* values are compared at a similar pumping power. The TEF tends to decrease with the rise of Re. It is seen that for the rectangular baffle, the e/H=0.2 provides the highest TEF. The mean TEF values are around 1.36, 1.51 and 1.53; and 1.62, 1.48 and 1.56 for the rectangular and triangular wavy baffles at e/H=0.4, 0.3 and 0.2, respectively. The maximum TEF is found for the triangular wavy baffle at e/H=0.40 and lower Re. This can be attributed to considerably lower friction loss from using the triangular wavy baffle having lower mean e/H value.

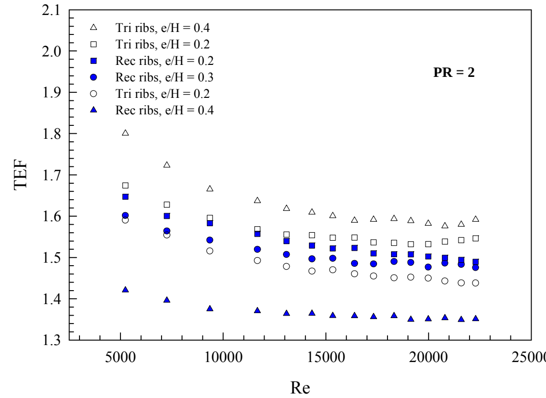

Fig. 8. Variation of thermal enhancement factor with Reynolds number.

5. Conclusions

An experimental study has been carried out to investigate airflow friction and heat transfer characteristics in a high aspect ratio channel (AR=10) mounted periodically with rectangular and triangular wavy baffles on the lower wall at different e/H values in the turbulent regime, Reynolds number of 5000-23,000. The use of the rectangular wavy baffle at e/H=0.4 causes a very high pressure drop increase and also gives considerable heat transfer augmentations, Nu/Nu_0 =4.85. The Nu tends to increase with the rise in Re values. In comparison, the use of the rectangular and triangular wavy baffles lead to the higher heat transfer rate but the triangular baffle at e/H=0.4 provides the highest TEF due to lower friction loss.

6. References

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