Heat Transfer and Friction Behavior in a Channel Fitted with Triangular and Rectangular V-shaped Ribs

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Abstract

An experimental investigation has been conducted to study the turbulent flow friction and heat transfer behavior in a rectangular channel fitted with triangular and rectangular V-shaped ribs mounted periodically on the lower plate of the channel while the upper plate was heated at a constant heat flux condition. The channel has an aspect ratio (channel width to height ratio, AR = W/H) of 10 and height (H) of 30 mm. The thin rib characteristics are the rib to channel height ratio (e/H) of 0.2, 0.3 and 0.4; rib pitch to channel height ratio (PR) of 4 and the attack angle (α) of 30° relative to the flow direction. Measurements were carried out by varying the airflow rate for Reynolds number in the range of 5000-24,000. The whole test section was insulated with thermal insulation in order to reduce heat loss to surrounding. The experimental results show that the use of the rib leads to considerable heat transfer rate and friction factor in comparison with the smooth channel. In addition, the rectangular V-shaped rib performs higher heat transfer rate and friction factor than the triangular one at a similar e/H ratio. The rectangular V-shaped rib with e/H = 0.4 yields the highest heat transfer rate but one with e/H = 0.2 provides the best thermal performance.

***Keywords**:* periodic; triangular; V-shaped rib; turbulent heat transfer; friction factor.

1. Introduction

The application of rib turbulators on the surfaces of typical heat exchangers is widely considered because it provides a substantial heat transfer enhancement. However, the heat transfer enhancement always accompanies a higher pressure drop penalty. Heat transfer characteristics in channels or ducts with rib-roughened walls have already been extensively investigated. Many investigations have been conducted toward establishing an optimal rib geometry, which gives the best heat transfer performance for a given pumping power or flow rate. Relevant geometric parameters are channel aspect ratio, rib angle of attack, flow blockage ratio (or rib height), rib shape, and relative arrangement of the ribs 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 investigators [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 a V-shaped ribs 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].

The objective of this experimental study is to investigate the heat transfer and friction behavior in a rectangular channel with rectangular and triangular v-shaped ribs fitted only on the lower plate but heating only on the upper plate of the channel. The experiment is carried out for various blockage ratios (e/H = 0.20, 0.30 and 0.40) with PR = 4 and the 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 24,000.

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 longitudinal pitch value equal to three and four times of channel height (pitch ratio, PR = 4) 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, of 300 mm. Each of rectangular and triangular geometric V-ribs was fabricated on the 0.3 mm thick aluminum tapes. The rib dimensions are 6, 9 and 12 mm high (e) and 0.3 mm thick (t) as shown in Fig. 2.

The channel test section 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.

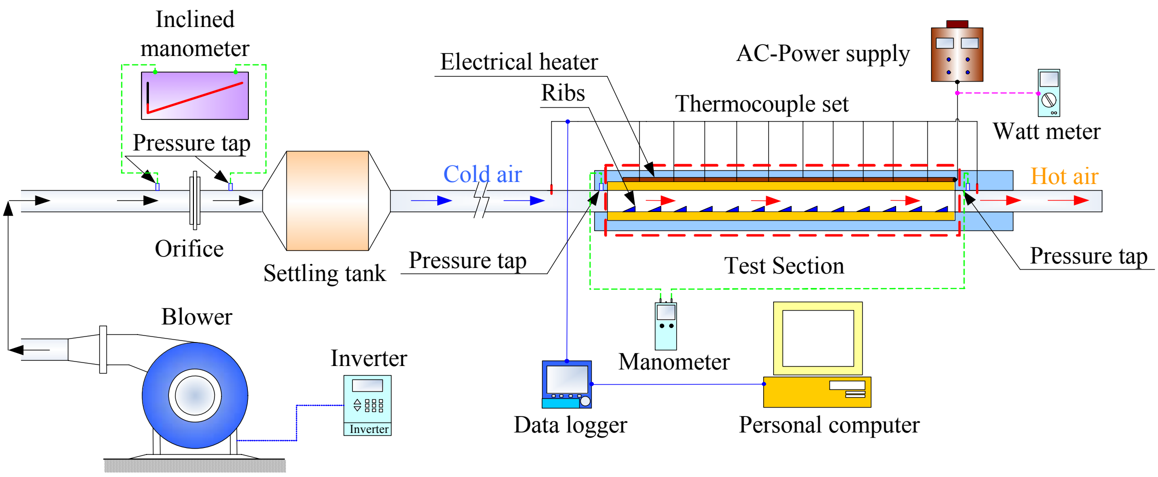

Fig. 1 Schematic diagram of experimental apparatus.

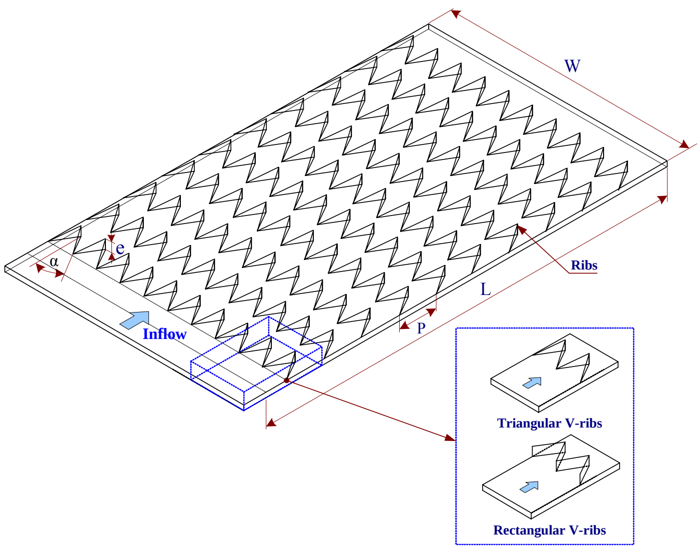

Fig. 2 Test section with downstream (PD) of rectangular and triangular V-ribs arrangement.

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 wall, twelve thermocouples were fitted to the wall. 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 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 of the leading edge of the test channel and the other is 30 mm downstream of 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:

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

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,

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

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

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

The thermal enhancement factor, η, 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:

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

4. Result and Discussion

In the present investigation, experimental measurements of both heat transfer and pressure loss in channels with periodical rectangular and triangular V-ribbed turbulators are presented. Measurements were conducted in a channel of aspect ratio, AR = 10 for two rib types and an attack angle of 30° over a range of Reynolds numbers as mentioned earlier.

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 Dittus-Boelter and Blasius found in the open literature [10] for turbulent flow in ducts.

Correlation of Dittus-Boelter,

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

Correlation of Blasius,

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

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

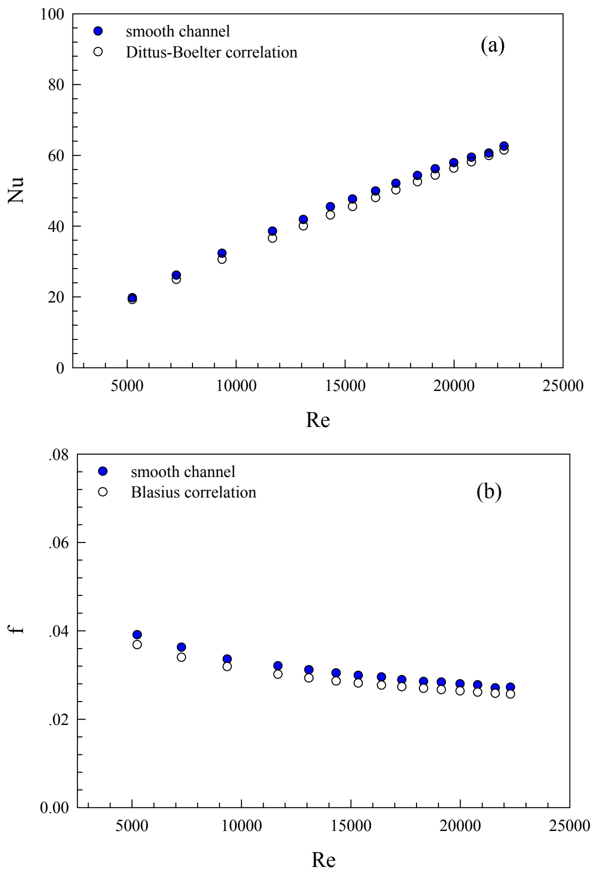

Fig. 3 Verification of (a) Nusselt number and (b) friction factor for smooth channel.

4.2 Effect of geometry

The experimental results on heat transfer and friction behavior in a uniform heat flux channel with rectangular and triangular V-shaped ribs, placed on the lower plate only are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for all cases are depicted in Fig. 4. In the figure, the rectangular and triangular V-rib turbulators provide considerable heat transfer enhancements with a similar trend in comparison with the smooth channel. The Nusselt number increases with the rise in Reynolds number for all the ribs. This is because the rib turbulators interrupt the development of the boundary layer thickness of the fluid flow and help to increase the turbulence degree of the flow. It is worth nothing that the rectangular V-rib with *e*/*H* = 0.4 provides the highest value of Nusselt number while the *e*/*H* = 0.3 performs higher than the *e*/*H* = 0.2. The rectangular V-rib is also found to perform better than the triangular V-ribs at a given *e*/*H* ratios. This caused that the rectangular rib provides a higher flow blockage, interrupts the flow and diverts its direction thus promoting high levels of flow mixing over the other. A close examination reveals that the rectangular V-ribs *e*/*H* = 0.4, produces the highest heat transfer rate than the others.

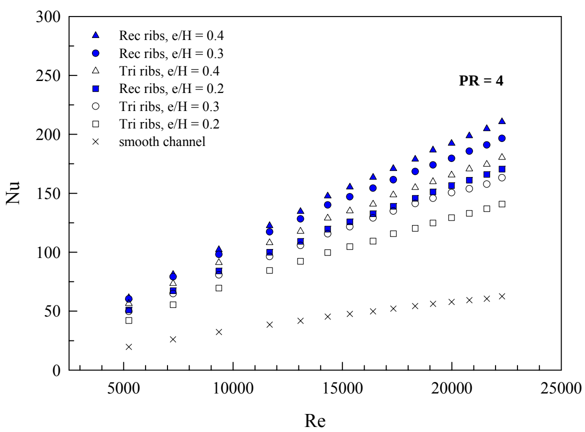

Fig. 4 Variation of Nusselt number with Reynolds number for various rib heights.

The effect of using the rectangular and triangular V-ribs on the isothermal pressure drop across the tested 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 V-ribs leads to a substantial increase in friction factor value over the smooth channel with no rib. As expected, the friction factor values of the rectangular V-rib with *e*/*H* = 0.4 are considerably higher than those with *e*/*H* = 0.3 and 0.2; and those of triangular V-ribs with *e*/*H* = 0.4, 0.3 and 0.2. For the rectangular rib with *e*/*H* = 0.4, the increase in friction factor is in the range of 110-275% above one with *e*/*H* = 0.3 or 0.2 and of 160 -430% over the triangular ribs depending on Reynolds number and *e*/*H* values.

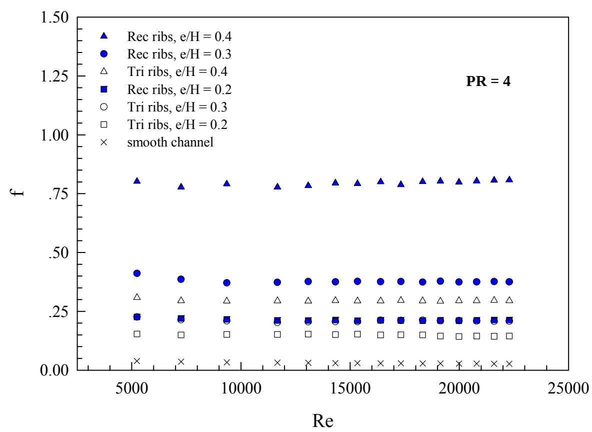

Fig. 5 Variation of friction factor with Reynolds number for various rib heights.

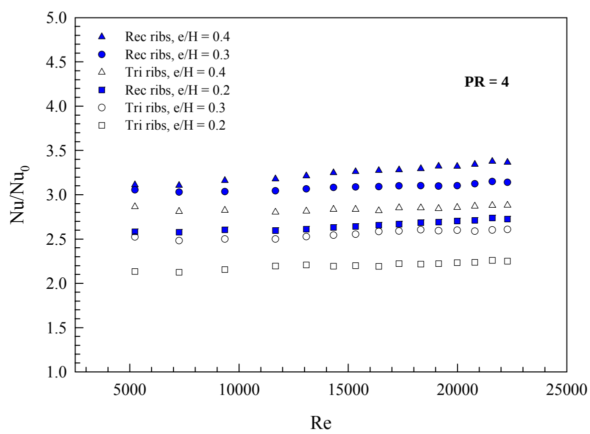

Fig. 6 Variation of Nu/Nu_0 with Reynolds number.

4.3 Effect of arrangement

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 depicted in Fig. 6. In the figure, the Nusselt number ratio tends to be nearly uniform with the rise of Reynolds number from 5000 to 24,000 for all cases. The Nusselt number ratio values of using the rectangular and triangular V-ribs are found to be about 3.26, 3.09 and 2.65; and 2.84, 2.56 and 2.20 times higher than the smooth channel for the e/H = 0.4, 0.3 and 0.2, respectively.

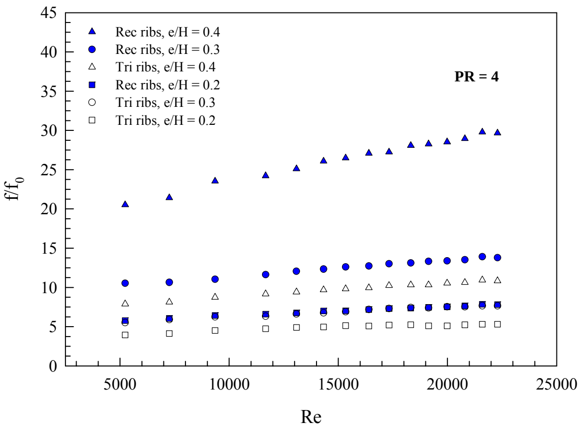

Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds Number.

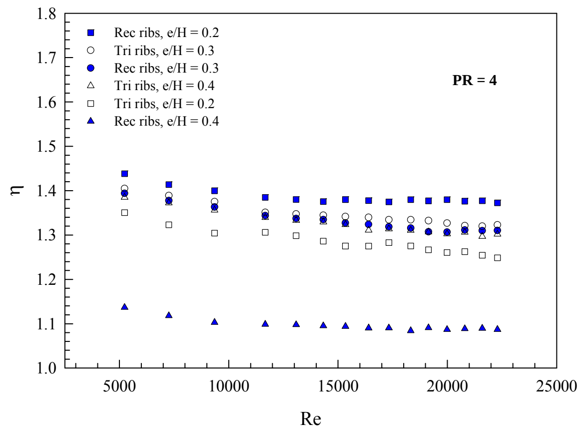

Fig. 8 Variation of thermal enhancement factor with Reynolds number.

The variation of the friction factor ratio, f/f_0 with Reynolds number for both three heights of rectangular and triangular V-ribs is also shown in Fig. 7. The friction factor ratio value is found to be increased with the increase in the Reynolds number and the blockage ratio values. The mean friction factor ratio values are around 26.33, 12.50, and 7.05-fold; and 9.78, 6.93 and 4.93-fold increases for using rectangular and triangular V-ribs with e/H = 0.4, 0.3 and 0.2, respectively. This result indicates that the use of lower blockage ratio can help to reduce the pressure loss considerably.

Fig. 8 shows the variation of the thermal enhancement factor (η) with Reynolds number. For all, the data obtained by Nusselt number and friction factor values are compared at a similar pumping power. The enhancement factor tends to decrease with the rise of Reynolds number for all. It is seen that the rectangular rib with e/H = 0.2 performs the highest thermal enhancement factor. The mean thermal enhancement factor values are around 1.39, 1.33 and 1.10; and around 1.325, 1.35 and 1.28 for both the rectangular and triangular V-ribs with e/H = 0.4, 0.3 and 0.2, respectively. The maximum thermal enhancement factor is found at e/H = 0.20 and lower Re for the rectangular V-ribs. This can be attributed to considerably lower friction loss from using the lower e/H rib.

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 V-shaped ribs on the lower wall at different flow blockage ratios in the turbulent regime, Reynolds number of 5000-24,000. The use of the rectangular ribs with *e*/*H* = 0.4 causes a very high pressure drop increase and also gives considerable heat transfer augmentations, Nu/Nu_0 = 3.45. The Nusselt number enhancement tends to increase with the rise in Reynolds number. In comparison, use of the rectangular and triangular ribs leads to the higher heat transfer rate but the rectangular rib with *e*/*H* = 0.2 provides the highest thermal enhancement factor due to lower friction loss.

6. References

[1] Han, J.C. (1988). Heat transfer and friction characteristics in rectangular channels with rib turbulators, *ASME J. Heat Transfer,* vol. 110, pp. 321-328.

[2] Chanda, P.R. and Han, J.C. (1989). Pressure drop and mass transfer in two-pass ribbed channels, *AIAA J. Thermophysics,* vol. 3, pp. 315-320.

[3] Han, J.C., Zhang, Y.M. and Lee, C.P. (1991). Augmented heat transfer in square channel with parallel crossed, and V-shaped angled ribs, *ASME J. Heat Transfer,* vol. 113, pp. 590-596.

[4] Hwang, J.J., and Liou, T.M., (1994) Augmented heat transfer in rectangular channel with permeable ribs mounted on the wall, *ASME J. Heat Transfer,* vol. 116, pp. 912-920.

[5] Gao, X. and Sunden, B. (2001). Heat transfer distribution in rectangular ducts with V-shaped ribs, *Heat and Mass Transfer*, vol. 37, 315-320.

[6] Chandra, P.R., Alexander, C.R. and Han, J.C. (2003). Heat transfer and friction behaviour in rectangular channels with varying number of ribbed walls. *Int. J. Heat Mass Transfer*, vol. 46, pp. 481–495.

[7] Sripattanapipat, S. and Promvonge, P. (2009), Numerical analysis of laminar heat transfer in a channel with diamond-shaped baffles, *Int. Commun. Heat Mass Transfer*, vol. 36, pp. 32-38.

[8] Varun, Saini, R.P. and Singal, S.K. (2007). A review on rib-roughness geometry used in solar air heaters, *Solar Energy,* vol. 81, pp. 1340-1350.

[9] ANSI/ASME, (1986). *Measurement uncertainty*, PTC 19, 1-1985, Part I.

[10] Incropera, F. and Dewitt, P.D. (2006). *Introduction to heat transfer*, 5th edition, John Wiley & Sons Inc.