# Effect of Wavy Ribs on Thermal Performance in a Solar Air Heater Channel

C. Khanoknaiyakarn, W. Noothong, S. Kwankaomeng and P. Promvonge*

Department of Mechanical Engineering, Faculty of Engineering,
King Mongkut's Institute of Technology Ladkrabang, Bangkok 10520, Thailand
*Corresponding Author: Email kppongje@kmitl.ac.th
Tel: +662-3298350-1; Fax: +662-3298352

***Abstract***

The paper presents an experimental study on the thermal enhancement in a solar air heater channel fitted with 30° wavy and 90° ribs. The experiments have been carried out by varying the airflow rate for Reynolds number in the range of 4000 to 26000 in the test section with a constant heat flux on the upper channel plate. The effects of three rib- to channel-height ratios, (e/H= 0.1, 0.2, and 0.3) on the heat transfer in terms of Nusselt number and the friction loss in the form of friction factor are experimentally investigated. The wavy ribs with the rib pitch equal to three times of channel height (PR=P/H=3) are mounted only on the heated upper plate. The experimental result shows that the 30° wavy rib with e/H = 0.3 provides the highest heat transfer and friction factor for all cases. The mean Nusselt number values are found to be 2.34, 3.18, and 3.43 times above the smooth channel while the mean friction factor values are around 3.26, 6.22,and 8.06 times for using the 30° wavy rib with e/H = 0.1, 0.2 and 0.3, respectively. The 30° wavy rib with e/H = 0.2 provides the best thermal performance.

***Keywords****:* ribs, Reynolds number, friction factor, thermal enhancement, channel.

## 1. Introduction

Solar air heaters are widely used as the thermal collection equipment. The thermal performance of a conventional solar air heater is poor because of the low convective heat transfer coefficient between surface and air. It can be improved by using the turbulent promoter in the form of artificial roughness on the heat transfer surface. Several techniques have been advised for enhancing the rate of convective heat transfer from the heat transfer surface. In the channel heat exchanger design, fin, groove, baffle, and rib are often employed in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency. Periodic flow interruption generated by rib arrays fitted on the channel wall is an extensively used means for heat transfer augmentation. The rib turbulators increase not only heat transfer rate but also substantial the friction loss. In particular, the rib geometry, the rib to channel height ratio, and the rib pitch to height ratio and the rib angle are the parameters that affect the thermal performance.

Many experiments have been carried out to study the effect of pertinent parameters of turbulators on heat transfer and friction factor for roughened surface. Han et al. [1] studied experimentally the heat transfer in a square channel with ribs on two walls for nine different rib configurations. Average heat transfer and friction factor were reported for P/e =10 and e/H=0.0625. They reported that friction factor were reported for P/e=10 and e/H=0.0625. They reported that the angled ribs and 'V' ribs yield higher heat transfer enhancement than the continuous ribs. The heat transfer augmentations and the friction factor were highest for the 60° orientation amongst the angled ribs. Han et al. [2] also investigated the influence of the surface heat flux ratio on the heat transfer in a ribbed square channel with e/H=0.063 and P/e=10, by 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 and the average Nusselt number tends to decrease for increasing Reynolds numbers and the thermal boundary condition becomes less relevant at higher Reynolds number. Wright et al. [3] investigated the heat transfer distributions and frictional losses in rotating ribbed channels with an aspect ratio of 4:1. Angled, discrete angled, V-shaped, and discrete V-shaped ribs were investigated, as well as the newly proposed W-shaped and discrete W-shaped ribs. In all cases, the ribs were placed on both the leading and trailing surfaces of the channel, and they were oriented 45 deg to the mainstream flow. The rib height-to-hydraulic diameter ratio e/H was 0.078, and the rib pitch-to-height ratio P/e was 10. The channel orientation with respect to the direction of rotation was 135°. The range of flow parameters included Reynolds number (Re=10,000–40,000). It was determined that the W-shaped and discrete W-shaped ribs had the superior heat transfer performance in both non-rotating and rotating channels. The angled rib configuration resulted in the worst performance of the six configurations in their study.

For a system with only one roughened wall and three smooth walls, several investigations [4-8] have been carried out on rib roughened absorber plates of solar air heaters. Correlations for heat transfer coefficient and friction factor have been developed for such a system. However, the increase in heat transfer is accompanied by an increase in the resistance of fluid flow. Promvonge and Thianpong [9] 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. [10] 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 e/H =0.4, the enhancement of heat transfer is about 2–3 folds higher than that for the 90° baffle while the friction loss is some 10–25% lower. An extensive literature review over hundred references on various rib turbulators was reported by Varun et al. [11]

The present experimental work is to investigate the heat transfer and friction characteristics in the channel fitted with 30° wavy and 90° ribs fitted on the heated upper plate of the channel. The experimental data is conducted for three rib blockage ratios (e/H = 0.1, 0.2 and 0.3) and rib attack angle of 30° and 90° at PR = 3. The experiment results using air as the test fluid are presented for turbulent channel flows in a range of Reynolds number from 4000 to 26,000 in this investigation.

## 2. Experimental Apparatus

A schematic diagram of the experimental apparatus is presented in Fig. 1 while the details of 30° V-shaped rib arrays on the rectangular channel used in the heat transfer experiments are depicted in Fig.2. 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 rectangular channel including a calm section and a test section was employed following the settling tank. In the test section, the rectangular channel configuration was characterized by the channel height, *H* of 30 mm, transverse pitch equal to three times of channel height (pitch ratio, PR=3) and the attack angle of 30°. The overall length of the channel was 2000 mm in which the test section was 400 mm with the width, *W*, of 300 mm. Each of the principle channel walls was fabricated from 6 mm thick aluminum plates, 300 mm wide and 400 mm long (L). The rib strip dimensions were 1.5, 3, 4.5, 6 and 7.5 mm high (e) and 0.3 mm thick (t).

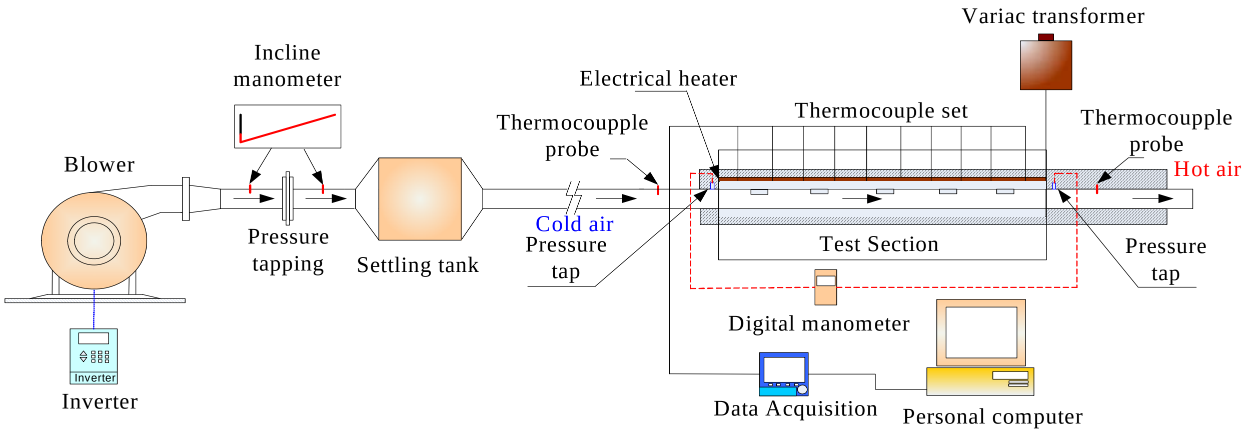


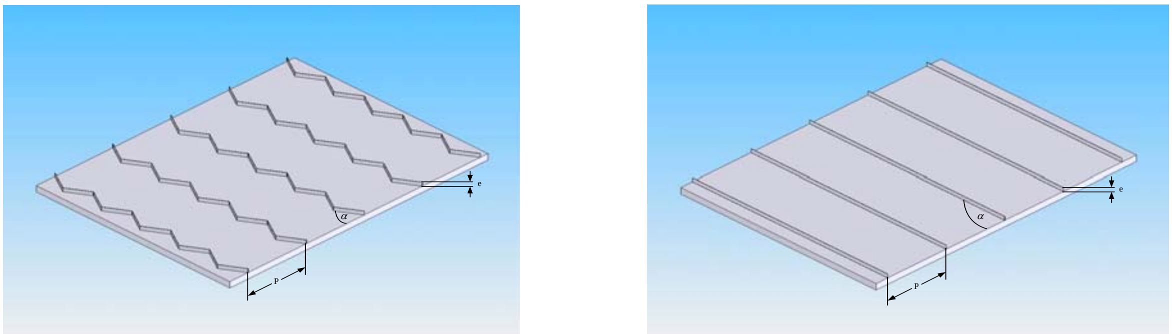

Fig.1 Schematic diagram of experimental apparatus.

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 to maintain uniform surface heat flux. A conducting compound was applied to the heater and the principal upper wall in order 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.

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, 1.5 mm diameter wire. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2650B) 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 120 mm downstream from the leading edge of the channel and the other is 50 mm upstream from the trailing edge. The pressure drop was measured by a digital differential pressure manometer (Testo 1445) 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%.



(a) (b)

Fig. 2 Test section with: (a) 30° wavy ribs and (b) 90° ribs.

## 3. Data reduction

The goal of this study 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:

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

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

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:

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

where $\Delta P$ is a pressure drop across the test section and $\rho$ 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,

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

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

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

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

The thermal enhancement factor, $\eta$, 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

### 4.1 Verification of smooth channel

The 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 data from the correlations of Dittus-Boelter and Blasius found in the open literature [10] for turbulent flow in ducts.

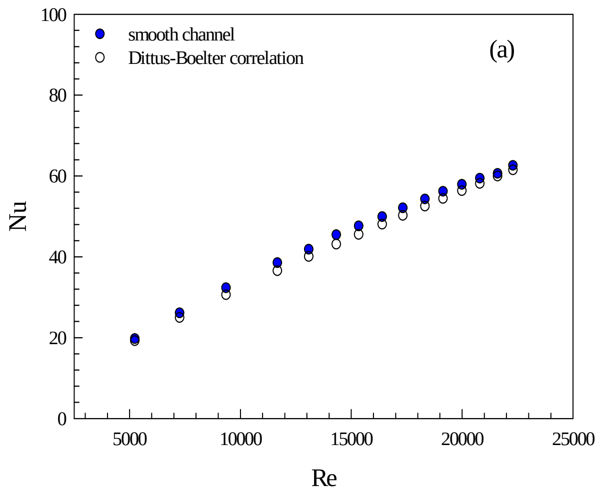
Correlation of Dittus-Boelter,

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

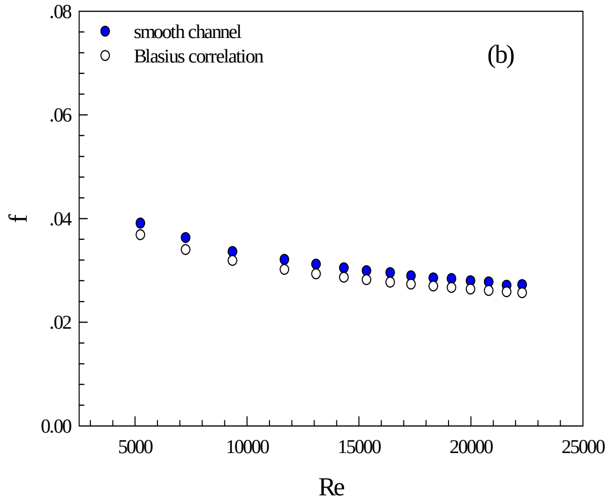
Correlation of Blasius,

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

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 $\pm 3\%$ for Nusselt number and friction factor correlations




(a)




(b)

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

### 4.2 Effect of rib height

The experimental results on thermal characteristics in a uniform heat flux absorber plate with both 30° wavy and 90° ribs placed on upper plate only are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for all case are depicted in Fig. 4. In the figure, the rib turbulators provide considerable heat transfer enhancements with a similar trend in comparison with the smooth channel. The Nusselt number increases with the rise of Reynolds number. This is because the rib turbulators interrupt the development of thermal boundary layer thickness of the fluid flow and help to increase the turbulence degree of flow. It is worth nothing that the 30° wavy rib with PR=3 and e/H=0.3 provides the highest Nusselt number while the one with e/H = 0.2 performs better than the one with e/H = 0.1. The 30° wavy rib is also found to perform better than the 90° rib at a similar e/H ratio. This caused that the 30° wavy rib with e/H=0.3 interrupting the flow and diverting its direction thus promoting high levels of mixing over others. A close examination reveals that the 30° wavy rib with e/H=0.3 produces the highest heat transfer coefficient than the one with other blockage ratio and all the 90° ribs.

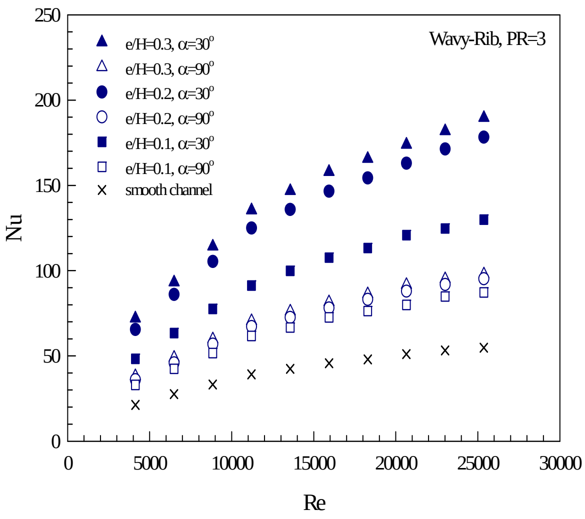



Fig. 4 Variation of Nusselt number with Reynolds number for various rib heights.

The effect of using the rib turbulators on the isothermal pressure drop across the tested channel is presented in Fig. 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 blocking flow of ribs, larger surface area and the act caused by the reverse flow. As expected, the friction factor of the rib with e/H=0.3 is considerably higher than the one with e/H=0.2 and 0.1, and the 90° ribs with all blockage ratios. For the wavy rib with e/H=0.3, the increase in friction factor is in the range of 128% and 240% over the e/H=0.2, and 0.1 and of 111%-154% above all the 90° ribs, respectively. 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 flow block because of the presence of the ribs.

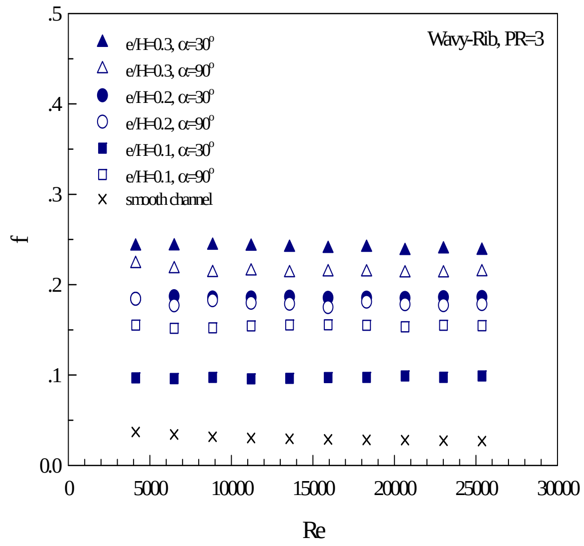



Fig. 5 Variation of friction factor with Reynolds number for various rib heights.

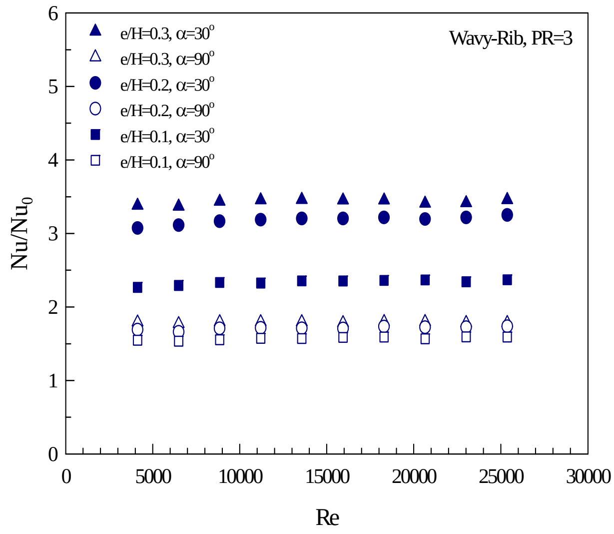



Fig. 6 Variation of Nusselt number ratio, $Nu/Nu_0$ with Reynolds number.

### 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 depicted in Fig. 6. In the figure, the Nusselt number ratio tends to be nearly uniform with the rise of Reynolds number for all cases. The mean $Nu/Nu_0$ values for using the 30° wavy and 90° ribs at e/H = 0.3, 0.2 and 0.1 are found to be about 3.43, 3.18, and 2.74; and 1.79, 1.71, and 1.57 times higher than the smooth channel, respectively.

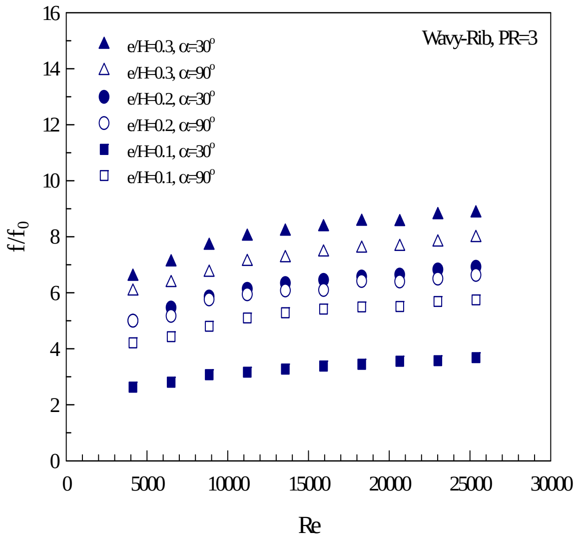



Fig. 7 Variation of friction factor ratio, $f/f_0$ with Reynolds Number.

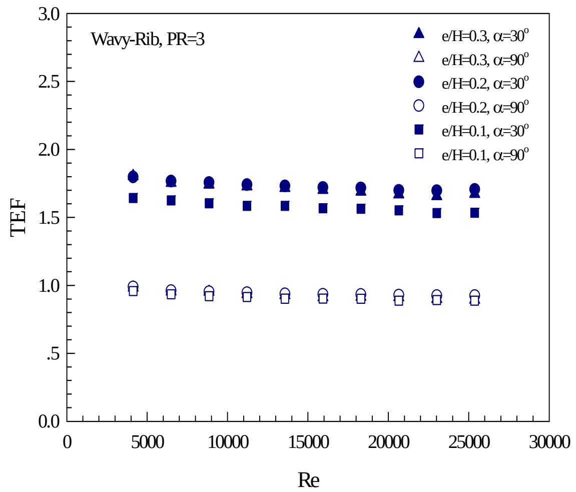



Fig. 8 Variation of thermal enhancement factor with Reynolds number.

The variation of isothermal friction factor ratio value, $f/f_0$, with Reynolds number for both 30° wavy rib and 90° rib cases is shown in Fig. 7. In the figure, the friction factor is found to be increased with increasing the Reynolds number and the blockage ratio. The mean friction factor values are around 8.06, 6.22, and 3.26 times; and 7.19, 5.99 and 5.17 times for using the 30° wavy ribs and 90° ribs with e/H = 0.3, 0.2, and 0.1, respectively. This result indicates that the use of low blockage ratio can help to reduce the pressure loss considerably.

Figure 8 shows the variation of the thermal enhancement factor (TEF) with Reynolds number. For all, the data obtained by Nusselt number and friction factor values are compared at similar pumping power. The TEF tends to decrease with the increment of Reynolds number for all cases. It is seen that the 30° wavy rib with e/H = 0.2 gives the highest TEF. The mean TEF values are around 1.73, 1.71, and 1.58; and 0.94, 0.93, and 0.91 for both the 30° wavy rib and 90° rib at e/H = 0.3, 0.2, and 0.1, respectively. The 30° wavy rib at e/H = 0.2 provides the maximum TEF. This can be attributed to considerably lower friction loss for using the lower blocking 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 with 30° wavy and 90° ribs on the upper plate wall at different blockage ratios in the turbulent regime, Reynolds number of 4000-26,000. The 30° wavy rib at e/H=0.3 causes a very high pressure drop and also provides considerable heat transfer augmentation. Nusselt number augmentation tends to increase with the rise of Reynolds number. In comparison, the use of 30° wavy rib and 90° rib leads to much higher heat transfer rate but the 30° wavy rib with e/H = 0.2 provides higher TEF due to the higher Nu ratio and the lower friction loss for the same pumping power and can be proven by using the equation (10).

## 6. References

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

[2] Han, J.C., Zhang, Y.M. and Lee, C.P. (1992). Influence of surface heat flux ratio on heat transfer augmentation in square channels with parallel, crossed, and V-shaped angled ribs, *ASME, Journal of Turbomachinery*, vol.114 pp. 872–880.

[3] L.M. Wright, W.L. Fu, J.C. Han. 2004. Thermal performance of angled, V-shaped rib, and W-shaped rib turbulators in rotation rectangular cooling channels (AR=4:1), *ASME, Journal of Turbomachinery*, vol. 126 pp. 604-614.

[4] K. Prasad, S.C. Mullick. (1983). Heat transfer characteristics of a solar air heater used for drying purposes, *Applied Energy*, vol. 13 pp. 83–93.

[5] D. Gupta, S.C. Solanki, J.S. Saini. (1997). Thermo-hydraulic performance of solar air heaters with roughened absorber plates, *Solar Energy*, vol. 61 pp. 33–42.

[6] J.L. Bhagoria, J.S. Saini, S.C. Solanki. (2002). Heat Transfer coefficient and friction factor correlation for rectangular solar air heater duct having transverse wedge shaped rib roughness on the absorber plate, *Renewable Energy*, vol. 25 pp. 341–369.

[7] R. Karwa. (2003). Experimental studies of augmented heat transfer and friction in asymmetrically heated rectangular ducts with ribs on the heated wall in transverse, inclined, V-continuous and V-discrete pattern, *International. Journal of Heat and Mass Transfer*, vol. 30 pp. 241–250.

[8] J.L. Bhagoria, M.M. Sahu. (2005). Augmentation of heat transfer coefficient by using 901 broken transverse ribs on absorber plate of solar air heater, *Renewable Energy*, vol. 25 vol. 2057–2073.

[9] P. Promvonge, C. Thianpong. (2008). Thermal performance assessment of turbulent channel flow over different shaped ribs, *International. Communications.in Heat and Mass Transfer*, vol. 35 pp. 1327–1334.

[10] P. Promvonge, S. Sripattanapipat, S. Tamna, S. Kwankaomeng, C. Thianpong. (2010). Numerical investigation of laminar heat transfer in a square channel with 45° inclined baffles, *Int. Communications.in Heat and Mass Transfer*, vol. 37 pp. 170–177.

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

[12] ANSI/ASME, Measurement uncertainty, PTC 19, 1-1985. Part I, 1986.

[13] F. Incropera, P.D. Dewitt, Fundamentals of heat and mass transfer, 6th edition, John Wiley & Sons Inc; 2007.