Heat Transfer Behavior in a Solar Air Heater Channel with V-Shaped Ribs

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Abstract

The research work presents the study of heat transfer enhancement in a solar air heater channel fitted with V-Shaped ribs. The experiments are carried out by varying airflow rate for Reynolds number ranging from 5000 to 25,000 in the test section with a constant surface heat flux on the upper plate of the channel which is similar to a solar air heater channel or solar collector. The V-shaped ribs with a transverse pitch value equal to two time of channel height and with the attack angle of 30° are mounted on the upper plate only. The effects of five rib to channel height ratios (e/H) of 0.05, 0.1, 0.15, 0.2 and 0.25 on heat transfer in terms of Nusselt number and friction loss in the form of friction factor are experimentally investigated. The experimental result shows that the V-Shaped rib with the e/H = 0.25 provides higher heat transfer and friction factor values than others. The mean Nusselt number values are found to be about 5.35, 4.75, 4.41, 3.72 and 2.74 times over the smooth channel while the mean friction factor values are around 22.58, 14.48, 9.98, 4.88 and 2.35 times for using the ribs with e/H = 0.25, 0.2, 0.15, 0.1 and 0.05, respectively.

Keywords*:* V-Shaped rib, Nusselt number, friction factor, solar air heater.

1. Introduction

The need for high-performance thermal systems in many engineering applications has stimulated considerable interest in finding various methods to improve heat transfer in the system. The conventional heat exchangers are generally improved by means of various augmentation techniques with emphasis on many types of surface enhancements. Augmented surfaces can create one or more combinations of the following conditions that are favorable for the increase in heat transfer rate with an undesirable rise of friction: (1) interruption of boundary layer development and increasing turbulence intensity; (2) increase in heat transfer area; and (3) generating of vortex and/or secondary flows. In the cooling channel or channel heat exchanger design, rib, fin, wing turbulators are often employed in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency. The use of wing turbulators completely results in the change of the flow field and hence the variation of the local convective heat transfer coefficient. Winglets have been successfully used for enhancement of heat transfer of modern thermal systems because they can generate longitudinal vortex flow and help to destabilize the main flow with less penalty of pressure loss.

Many investigations have been carried out to study the effect of these 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 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 square ribbed channel with e/H=0.0625 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. Taslim et al. [3] 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. Chandra et al. [4] 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. Sripattanapipat and Promvonge [5] 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 baffle with half apex angle of 5–10° provided slightly better thermal performance than the flat baffle. However, the increase in heat transfer is accompanied by an increase in the resistance of fluid flow. An extensive literature review over hundred references on various rib turbulators was reported by Varun et al. [6]

Thus, the main aim of this work is to extend the experimental data available on various blockage ratio (e/H=0.05, 0.10, 0.15, 0.20 and 0.25) with similar pitch ratio of (PR=2) and with the attack angle of 30°, placed on upper plate only are presented in turbulent channel flows in a range of Reynolds number from 5000-25,000

2. Experimental setup

A schematic diagram of the experimental apparatus is presented in Fig. 1 while the details 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 test section, the rectangular channel configuration is characterized by the channel height, *H* is 30 mm and transverse pitch value equal to three times of channel height (pitch ratio, PR=3) and with the attack angle of 30°. The overall length of the channel is 2000 mm of the test section with the channel width, *W*, of 300 mm. Each of the rib walls was fabricated from 6 mm thick aluminum plates, 300 mm wide and 400 mm long (L). The rib dimensions are 1.5, 3, 4.5, 6 and 7.5 mm high (e) and 0.3 mm thick (t).

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 (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%.

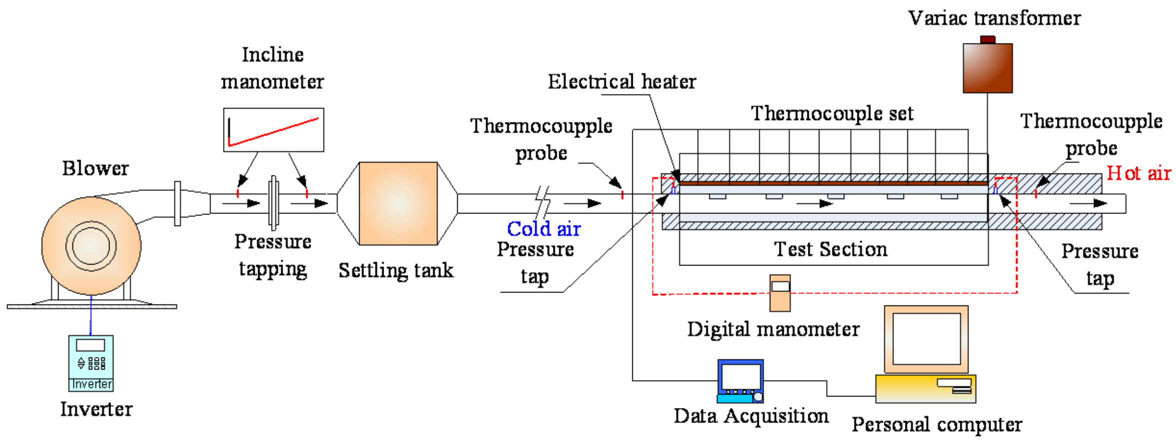

Fig. 1 Schematic diagram of experimental apparatus

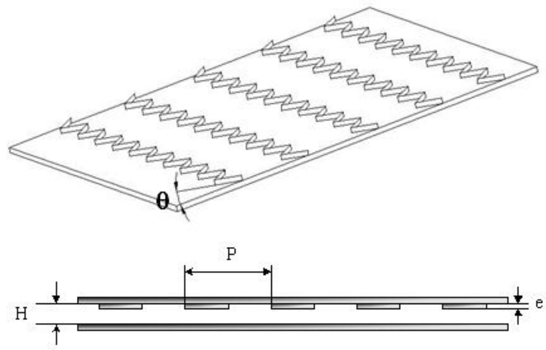

Fig. 2 Test section with wavy rib arrangement

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

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

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:

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

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

in which,

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

and

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

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 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,

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

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, η, defined as the ratio of heat transfer coefficient of an augmented surface, h to that of the smooth surface, h_o, 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 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,

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

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

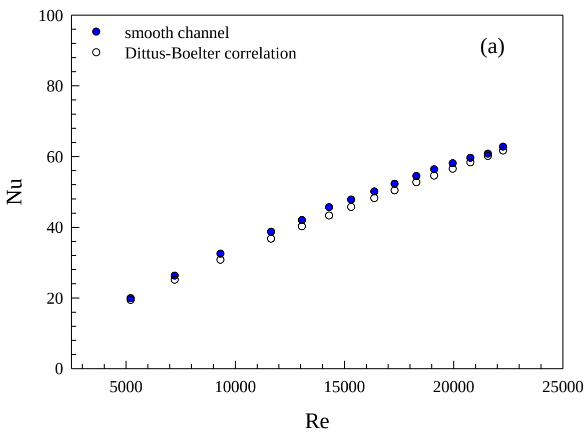

Fig. 3 (a) Verification of Nusselt number for smooth channel.

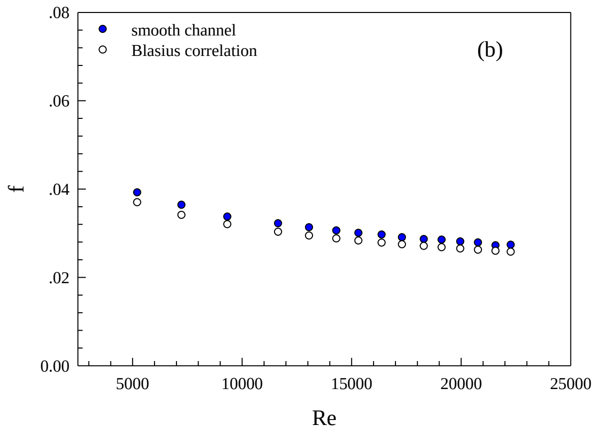

Fig. 3 (b) Verification of friction factor for smooth channel.

4.2 Effect of rib height

The present experimental results on heat and flow friction characteristics in a uniform heat flux channel with rectangular v-shaped rib, 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 presented in Fig. 4. In the figure, the rectangular v-shaped rib turbulators yield considerable heat transfer enhancements 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 rectangular v-shaped rib turbulators interrupt the development of the boundary layer of the fluid flow and increase the turbulence degree of flow. It is worth nothing that the heat transfer coefficient for the 30° rectangular v-shaped rib with pitch ratio of 2 (PR=2) and rib to channel height ratios (e/H) of 0.25 provides the highest value of Nusselt number while the e/H = 0.2 is found to perform better than e/H = 0.15, 0.10 and 0.05 for all types of rib. This caused by e/H = 0.25 interrupting the flow and diverting its direction thus promoting high levels of mixing over others. A close examination reveals that the rib to channel height ratios of 0.25 produces the highest heat transfer coefficient than other blockage ratio of rectangular v-shaped rib.

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 flow blockage, higher surface area and the act caused by the reverse flow. As expected, the friction factor of rib to channel height ratios (e/H) of 0.25 is considerably higher than those of 0.2, 0.15 and 0.1. For the rib of e/H = 0.25, the increase in friction factor is in the range of 156-960% over the e/H = 0.2, 0.15, 0.10, and 0.05 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 blockage ratios because of the presence of the ribs.

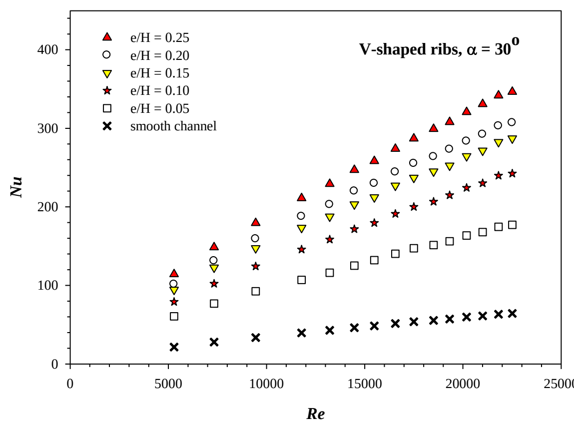

Fig. 4 Variation of Nusselt number with Reynolds number for various rib heights.

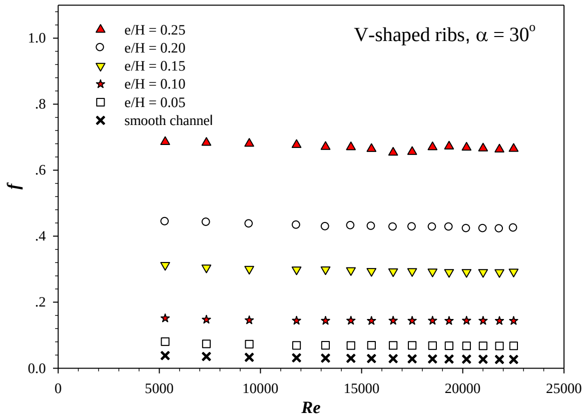

Fig. 5 Variation of friction factor with Reynolds number for various rib heights.

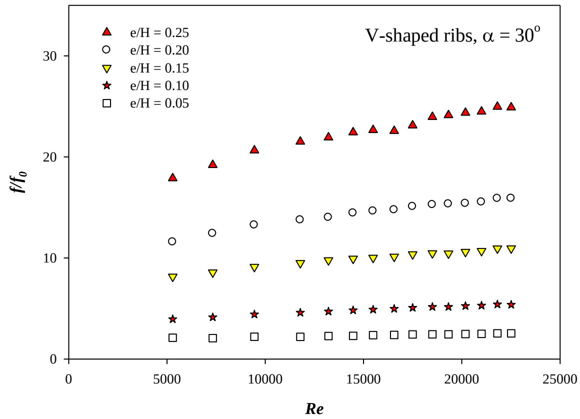

Fig. 7 Variation of friction factor ratio, f/f_0 with Reynolds Number.

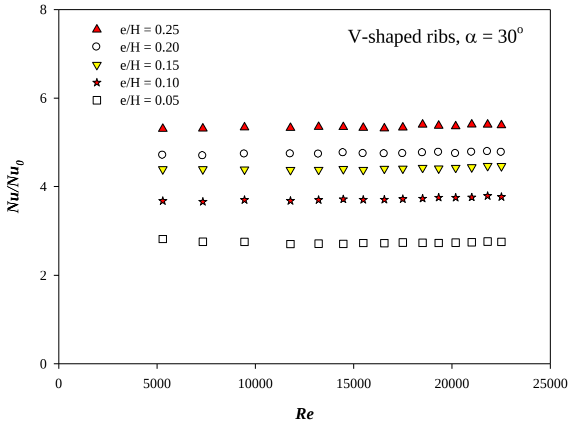

Fig. 6 Variation of Nu/Nu_0 with Reynolds number.

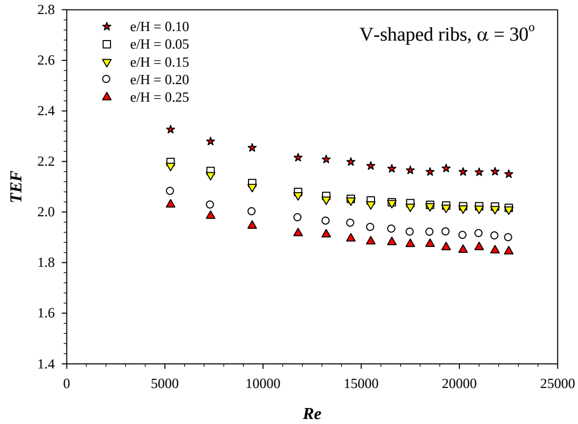

Fig. 8 Variation of thermal enhancement factor with Reynolds number.

4.3 Performance evaluation

The Nusselt number ratio, Nu_a/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 be nearly uniform with the rise of Reynolds number from 5000 to 25,000 for all cases of e/H of 0.25, 0.20, 0.15, 0.10, and 0.05. The mean Nusselt number ratio values are found to be about 5.35, 4.75, 4.41, 3.72 and 2.74 times over the smooth channel for using the 30° wavy ribs with e/H = 0.25, 0.20, 0.15 0.10 and 0.05, respectively.

The variation of isothermal friction factor ratio value with Reynolds number for five heights of v-shaped ribs case is also depicted in Figure 7. In the figure, the friction factor value is found to be increased with increasing the Reynolds number and the blockage ratio. The mean friction factor values are around 22.58, 14.48, 9.98, 4.88 and 2.35 fold for using the inclined ribs with e/H = 0.25, 0.20, 0.15, 0.10 and 0.05, 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 cases. For all, the data obtained by

Nusselt number and friction factor values are compared at similar pumping power. The enhancement factor tends to decrease with the rise of Reynolds number values for all. It is seen that the blockage ratio of 0.10 shows the highest value of mean the thermal enhancement factor. The mean thermal enhancement factor values are around 1.90, 1.95, 2.05, 2.20, and 2.06 times for using the inclined ribs with e/H = 0.25, 0.20, 0.15, 0.10 and 0.05, respectively. The results are for Reynolds number of 5000-25,000 for the inclined rib 30°, the maximum thermal enhancement factor is found at e/H = 0.10. This can be attributed to considerably lower friction loss.

5. Conclusions

Experimental study has been carried out to investigate airflow friction and heat transfer characteristics in a high aspect ratio channel (AR=10) fitted with different blockage ratio turbulators for the turbulent regime, Reynolds number of 5000-25,000. The use of the ribs with e/H = 0.25 causes a very high pressure drop increase and also provides considerable heat transfer augmentations, Nu/Nu_0 = 4.98. 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 e/H = 0.10 provides the higher thermal enhancement factor due to lower friction loss.

6. Acknowledgement

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7. References

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