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Heat transfer and friction factor characteristics of rectangular channel solar air heater duct having protrusions as roughness element

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Abstract
An experimental investigation has been carried out to see the effect of roughness element on heat transfer and friction factor on the absorber plate of the solar air heater. The roughness provided is in the form of protrusions which are arranged in an arc pattern. This paper presents an investigation to study the effect of protruded geometry on heat transfer coefficient and friction factor in an artificially roughened solar air heater duct. The pair of protrusion geometry arc angle (α) of 45^o is mounted on the test section of duct to create a longitudinal flow through test section. Measurements are carried out for rectangular duct which has aspect ratio (W/H) of 11, relative roughness pitch (P/e) in the range of 12-24, relative roughness height (e/D) of 0.03, ratio of height of protrusion to print diameter (d) of 0.3° and Reynolds number (Re) ranges from 3600-18100.The results obtained for various relative roughness pitch (P/e) has also been compared with smooth one. And on comparison to smooth duct, the roughened duct enhances the heat transfer and friction factor by 2.96 and 2.73 times.

Keywords: Artificial roughness; Protrusions; Relative roughness pitch; Nusselt number; Friction factor.

1. Introduction
Solar energy is very large, inexhaustible source of energy. The power from the sun intercepted by the earth is approximately 1.8×10^{11} MW, which are many thousands of times larger than the present consumption rate on the earth of all commercial energy sources. Thus, in principle, solar energy could supply all the present and future energy needs of the world on a continuing basis and makes it one of the most promising of the unconventional energy sources [1]. The simplest and the most efficient way to utilize solar energy are to convert it into thermal energy for heating applications by using solar collectors. Solar air heaters, because of their simplicity are cheap and most widely used collection devices. Solar air heaters have been found to have a low thermal efficiency because of its low heat transfer coefficient between the absorber plate and air which leads to high absorber plate temperature and hence greater amount of heat losses to the surroundings.
The thermal efficiency of solar air heaters has been found to be generally poor because of low heat transfer capability between the absorber plate and air flowing through duct. Use of an artificial roughness below the absorber plate is a most effective technique to enhance the heat transfer rate [2]. However, it would result in an increase in frictional loss leading to more power required by fan or blower. To keep the friction losses at a minimum level, the turbulence must be created only in the region close to the duct surface (in laminar sub-layer region). The surface roughness can be produced by several methods, such

as sand blasting, machining, casting, forming, welding or providing ribs of small diameter wires. Nikuradse [3] investigated the effect of roughness on the friction factor and velocity distribution in pipes roughened by sand blasting. Dippery and Sabersky [4] developed a friction similarity law and a heat momentum transfer analogy for flow by sand blasting technique through roughened tubes. In case of solar air heaters, artificial roughness in the form of ribs has been investigated several times. The ribs can be discrete or continuous depending on whether ribs in pieces or one complete rib over the absorber plate. The orientation of the ribs can be as transverse, inclined, v-shaped and w-shaped ribs.
Varun et al., Hans et al. and Bhushan and Singh [5-7] presented a review of various types of roughness geometries used in solar air heaters. Webb et al. [8] developed heat transfer and friction factor correlations for turbulent flow in tubes having repeated rib roughness. Prasad and Saini [9] investigated the effect of relative roughness height and relative roughness pitch on heat transfer and friction factor. They developed the relations to calculate the average friction factor and Stanton number for artificial roughness of absorber plate by small diameter protrusion wire. Han and Zhang [10] found that ribs inclined at an angle of attack of 45º gives better results as compared to transverse ribs. But further studies by Lau et al. [11] and Taslim et al. [12] investigated the effect of v-shaped ribs and found that v-shaped ribs result in better enhancement in heat transfer in comparison of inclined and transverse ribs. Gupta et al. [13] investigated the thermo-hydraulic performance in terms of effective efficiency of solar air heater with rib roughened surface by using heat transfer and friction factor correlation developed by them. Muluwork et al. [14] compared the thermal performance of v-shaped staggered discrete ribs with transverse staggered discrete ribs. Momin et al. [15] experimentally investigated the effect of geometrical parameters of v-shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of a solar air heater. Saini and Saini [16] experimentally investigated the effect of geometrical parameters of arc-shaped wire as roughness on heat transfer and frictional characteristics of rectangular duct of a solar air heater. The various investigators have developed correlations for heat transfer and friction factor for solar air heater ducts having artificial roughness of different geometries. Several researchers have carried out the detailed experimental investigation for various types of ribs on the absorber plate, but none of study has been reported for their respective combination [17-20]. Recently, Bhushan and Singh [21] presented data on the influence of Reynolds number on heat transfer coefficient distribution on the surface having staggered array of protrusion geometry. In some cases, enhancement in heat transfer rate is about 2.5 times than smooth surface value over a range of Reynolds number. Hans et al. [22] investigated the effect of multiple v-ribs over the absorber plate. Sukhmeet et al. [23] performed study of discrete v-ribs and Kumar et al. [24] accomplished the study of discrete w-ribs. Each concluded that heat transfer and friction factor enhancement is more in discrete geometry as compared to continuous geometry.

2. Experimental details

2.1 Experimental setup

An experimental set-up has been designed and fabricated as per the ASHRAE standard [25]. A schematic layout experimental setup is shown in Figure 1 and the cross sectional view of setup is shown in Figure 2. The schematic diagram of experimental setup consists of three sections, an entry section, test section and outlet section respectively. The rectangular duct is 2300 mm × 330 mm × 30 mm and fabricated with the wood. The test section is of 1500 mm long and 30 mm in height. The entry and exit section length were 530 mm and 270 mm respectively. ASHRAE standard 97-77 [25] recommends the minimum entry length is of 5√ (WH) and exit length is 2.5√ (WH). After outlet section of duct, a plenum was also provided for the proper mixing of air. The three walls (two sides and a bottom) of test section is finished with white sun mica and a 2 mm thick artificial roughened aluminium absorber plate acts as top wall of test section. The upper side of the absorber plate painted with dull black paint to absorb more heat radiations. The artificial roughness has been provided by using an indent on underside of plate and an arc shaped protruded geometry is created.
An electric heater having size of 1500 mm × 330 mm is fabricated by combining series and parallel loops of heating wire on a 5 mm thick asbestos sheet. A mica sheet of 1mm thickness was placed on electric heater wire in order to get uniform radiation between the electric heater and absorber plate. The heat flux in this case varies from 0 to 1000 W/m^2 with the help of variable transformer. The glass wool as an insulating material and 6 mm thick wooden panel was provided in order to reduce the heat losses from top side of heater assembly. The absorber plate is 2 mm thick and protrusion as roughness is provided at its bottom side in arc shape.

The mass flow rate of air was measured by means of calibrated orifice meter connected with an U-tube manometer. Control valves are provided to control the flow. An orifice plate was designed for flow measurement in the pipe having inner diameter 80 mm. A digital milli-voltmeter was used to indicate the output of the thermocouples. The pressure drop at the test section was measured with the help of micro-manometer.
The roughness element is produced by making protrusions of the desired size on the underside of the absorber plate. These dimples have been arranged as an arc fashion to maximize the heat transfer from the absorber surface. Figure 3 shows the geometry of roughness element used in the present work.

2.2 Experimental procedure
The roughened absorber plates were fabricated for different parameters of arc shaped protrusion as shown in Table 1. For determination of Nusselt Number (Nu) and Friction Factor (f) of roughened duct and also of smooth duct, data on smooth duct was also collected under similar operating conditions. For each experimental run, initially all the instruments viz., multi-voltmeter, micro-manometer, inclined U-tube manometer, voltmeter and ammeter, were checked for their correctness and all joints of duct were checked to avoid any air leakage. Data was recorded under quasi-steady state conditions for the air temperature at inlet to test section, air temperature after mixing section at three locations in transverse direction of duct, temperature of absorber plate at twelve locations, pressure drop across orifice and test section pressure drop. In the beginning, quasi-steady state was attained in 2-3 hours. The experimental setup was considered to be in quasi-steady state, when the temperatures of air after mixing section and absorber plate did not change appreciably for 10-15 minutes.

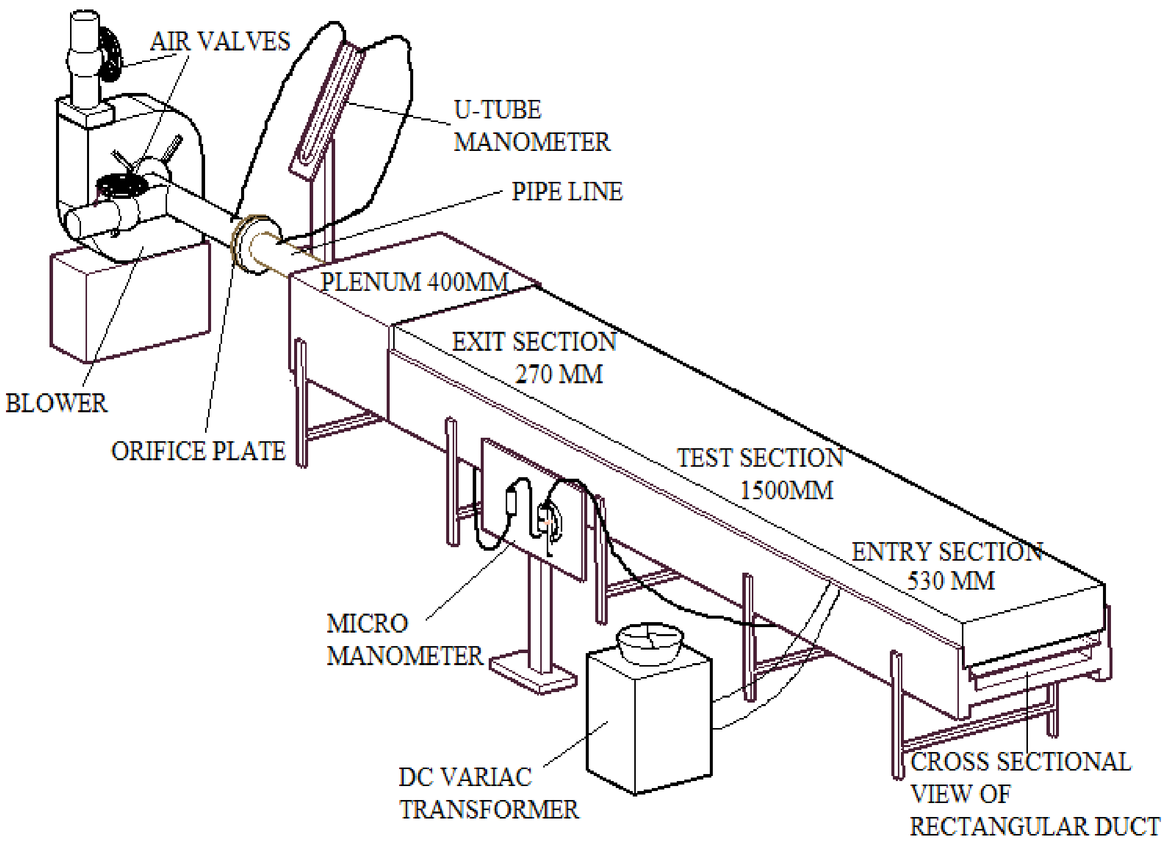

Figure 1. Schematic diagram of rectangular duct

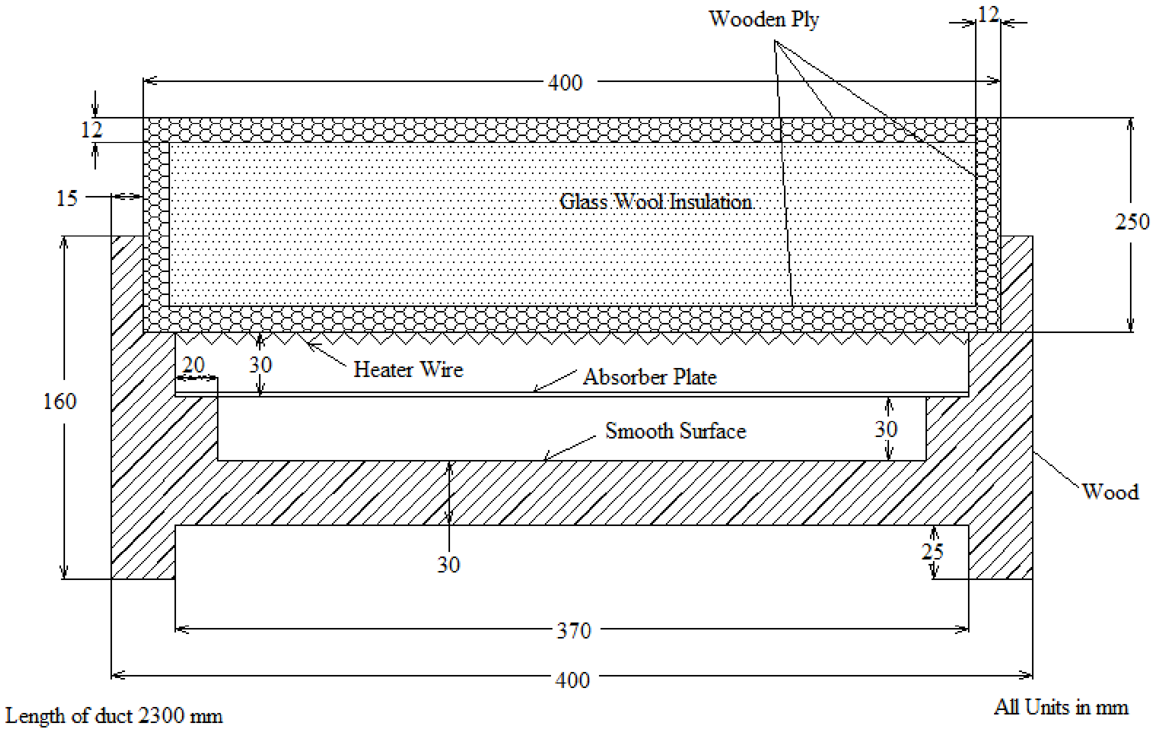

Figure 2. Cross sectional view of rectangular duct

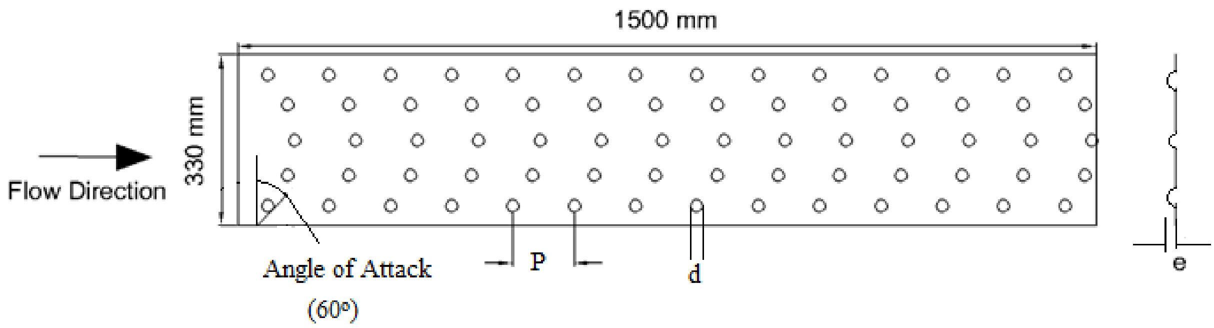

Figure 3. Diagram of the absorber plate

Table 1. Values of flow and roughness parameter

S. No.	Parameter	Range
1.	Aspect ratio W/H	11
2.	Relative roughness height (e/D)	0.03
3.	Relative roughness pitch (P/e)	12-24
4.	Angle of attack (α)	45°
5.	Ratio of height to print diameter of protrusion (e/d)	0.3
6.	Reynolds number	3600-18100

3. Data reduction

The experimental data such as pressure drop across orifice and test section, air and plate temperatures at various locations in the duct was recorded under quasi-steady state conditions at different mass flow rates of air. This data was used to calculate heat transfer rate of air flowing in the duct. Nusselt number and friction factor are also computed to see the effect of roughness geometry and operating parameters on heat transfer and friction characteristics. The following equations were used to calculate the mass flow rate ‘m’, heat gained by air ‘Q_u’ and heat transfer coefficient ‘h’:

$$m = C_d A_o \sqrt{\frac{2\rho \Delta P_o}{1-\beta^4}} \tag{1}$$

The calibration of orifice-plate is done against a standard Pitot tube which gives a value of 0.608 for coefficient of discharge (C_d).Where, $\Delta P_\circ = 9.81 \rho_m \Delta h_\circ \sin\theta$
whereas θ = 90°, for this case.

$$Q_u = mC_p(T_o - T_i) \tag{2}$$

and

$$h = \frac{Q_u}{A_p(T_p - T_f)} \tag{3}$$

where A_p is the heat transfer area (area of absorber plate), T_f and T_p are average values of air and absorber plate temperatures respectively. The Nusselt number (Nu) and friction factor (f) were calculated by using the following relationships.

$$Nu = \frac{hD}{k} \tag{4}$$

$$f = \frac{2\Delta P_t D}{4\rho L V^2} \tag{5}$$

where, $D = \frac{4WH}{2(W+H)}$ and $\Delta P_t = \rho g \Delta h_t$

For the above stated calculations, the air properties corresponding to bulk mean air temperature were used. The uncertainty analysis is done as per the method proposed by Kline and McClintock [26]. For all the protruded roughened plates investigated, the maximum uncertainty values for non-dimensional number are given below:
Reynolds Number: ±2.27%
Nusselt Number: ±4.91%
Friction Factor: ±4.37%

4. Validity test

To validate the experimental data, Nusselt number and friction factor for smooth duct were experimentally determined. The experimentally obtained values for smooth duct are compared with Dittus-Boelter [27] and Nikuradse-Karman [28] equation for Nusselt number and friction factor respectively.
Dittus-Boelter and Nikuradse-Karman equation is given as below:

$$Nu_s = 0.023 \,\mathrm{Re}^{0.8} \,\mathrm{Pr}^{0.4} \tag{6}$$

$$f_s = 0.046 \,\mathrm{Re}^{-0.2} \tag{7}$$

Comparison of experimental and predicted values of Nusselt number and friction factor for smooth duct is shown in Figures 4 and 5. A reasonable good agreement between the experimental and predicted data ensures accuracy of the data being collected with the help of experimental set up.

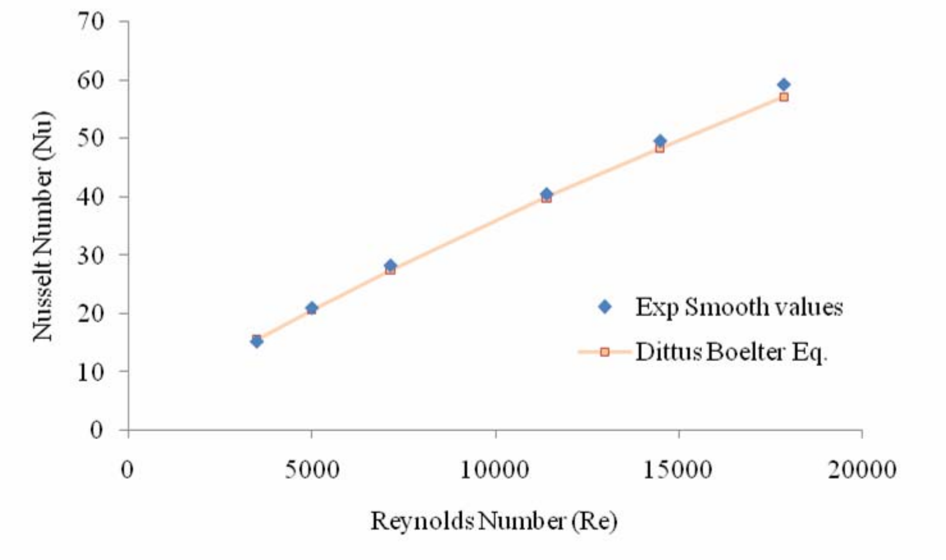

Figure 4. Comparison of experimental and predicted data of Nusselt number for smooth plate

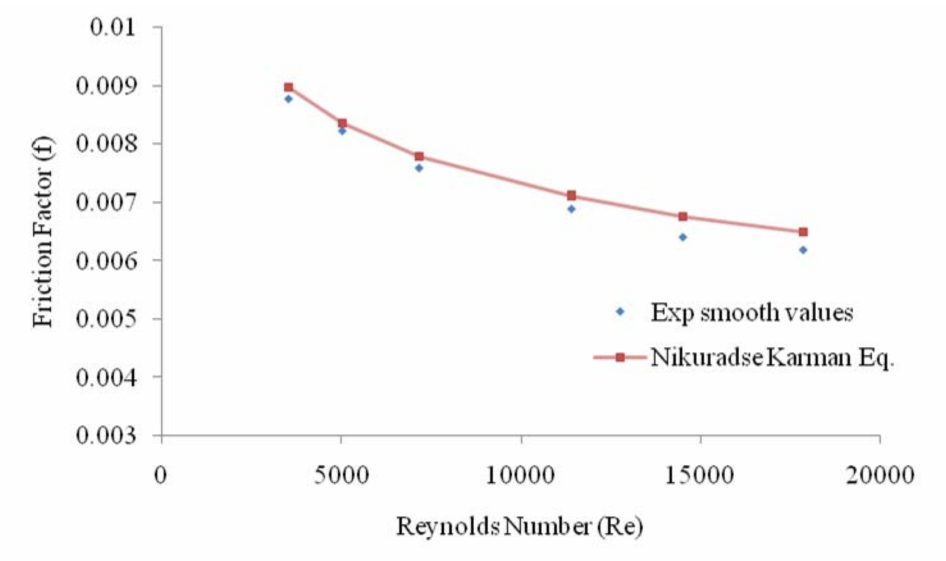

Figure 5. Comparison of experimental and predicted data of friction factor for smooth plate

5. Results and discussion

The results on Nusselt number and friction factor of protrusion arc shaped roughness provided in the duct for the parameters range given in Table 1 are reported and discussed. Under similar conditions, Nusselt number and friction factor of smooth duct are also reported for the comparison.

Figure 6 shows the variation of Nusselt number with Reynolds number for different values of relative roughness pitch (P/e). The Nusselt number is observed to increase for all values of relative roughness pitch (P/e). It is due to the fact that increase in Reynolds number causes the increase in turbulence which leads to increase in heat transfer. From Figure 6, it is also seen that the Nusselt number varies with the variation in the relative roughness pitch (P/e). To clearly show the variation in the Nusselt number for different values of Reynolds number, these results have been replotted in Figure 7. It is also seen that the value of Nusselt number decreases with the increase in relative roughness pitch (P/e). It is due to the fact with the increase in relative roughness pitch (P/e), the distance between the protrusions increases which results in the generation of lesser reattachment points.

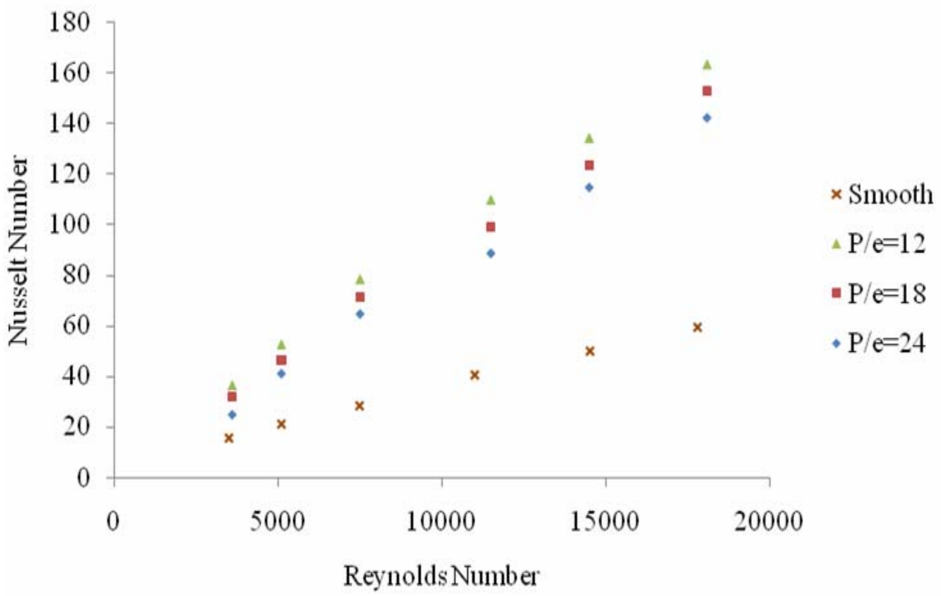

Figure 6. Variation of the Nusselt number with the Reynolds number for different values of P/e

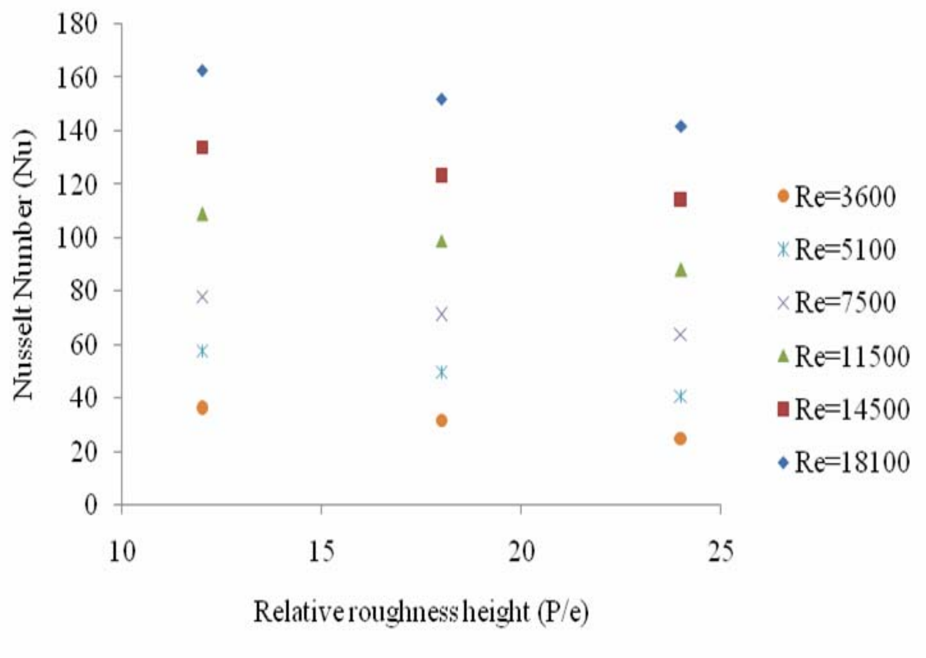

Figure 7. Variation of the Nusselt number for various relative roughness pitch (P/e) at different Reynolds number

The effect of relative roughness pitch (P/e) on friction factor with Reynolds number is shown in Figure 8. For all the values of relative roughness pitch (P/e), the friction factor decreases with the increase of Reynolds number.

In order to show the effect of relative roughness pitch (P/e) on friction factor, the results have been re-plotted in Figure 9 for different values of Reynolds number. The friction factor decreases with the increase in relative roughness height (P/e) for different values of Reynolds number. The maximum value of friction factor is found at a relative roughness height (P/e) of 12 which is also maximum in the case of heat transfer.

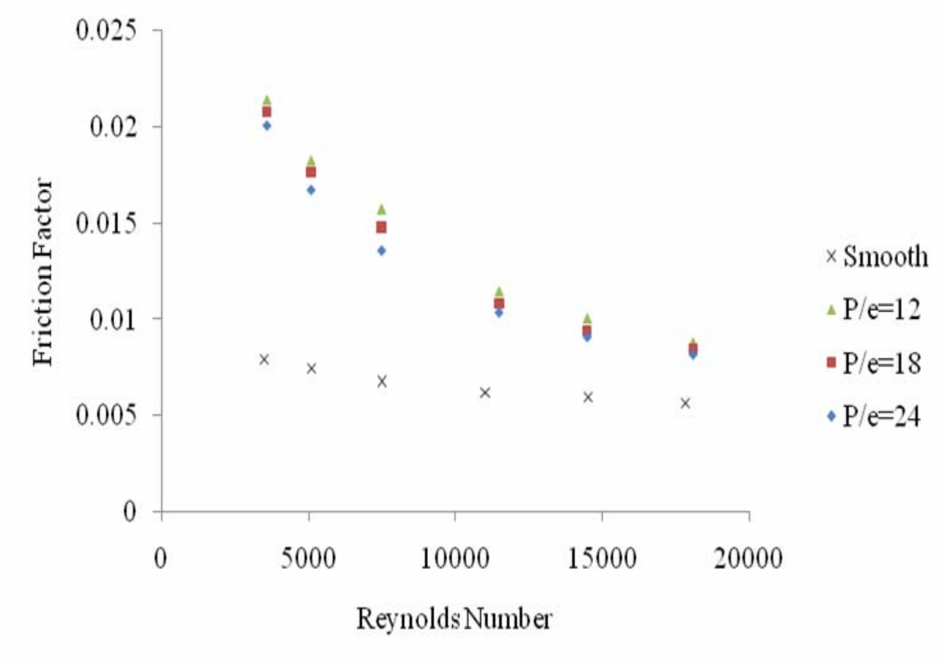

Figure 8. Variation of friction factor with the Reynolds number for different values of P/e

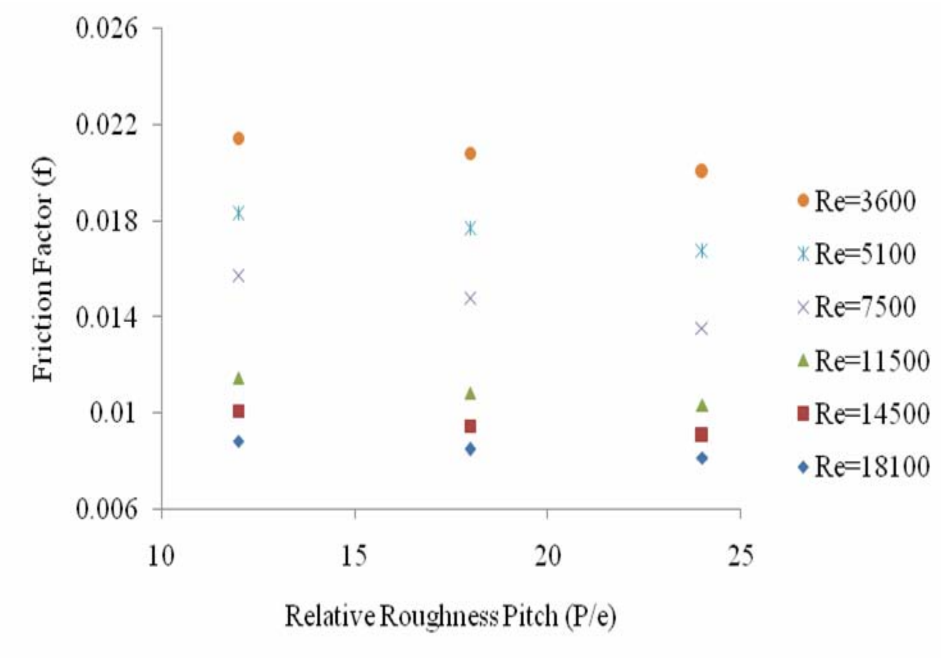

Figure 9. Variation of the friction factor for various relative roughness pitch (P/e) at different Reynolds number

6. Conclusion

On the basis of experimental investigation on heat transfer and friction factor of solar air heater ducts having rectangular cross-section provided protrusion-shaped roughness element on the absorber plate, the following conclusions are drawn:

1. The friction factor and Nusselt number of roughened duct are strong functions of flow attack angle.
2. The results have also been compared with smooth duct also to show a significant increase in the heat transfer through roughened duct..

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