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Experimental study of Nusselt number and Friction factor in solar air heater duct with Diamond shaped rib roughness on absorber plate

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Abstract:- Solar air heater is used to heat air but it has low thermal efficiency because of low thermal conductivity between air and absorber plate. Thermal efficiency of solar air heater can be improved by creating artificial roughness on absorber plate which causes higher temperature to absorber plate and hence maximum thermal losses occurs to atmosphere. There are number of parameters which enhances the thermal conductivity such as relative roughness height (e/D_h), relative roughness pitch (P/e), Reynolds number (Re), and angle of attack (α).Experimental investigations were carried out to study heat transfer enhancement using diamond shape rib on absorber plate of solar air heater. Absorber plate is heated with the solar radiation in outdoor experiment whereas electric heater is used for indoor experiment. Setup is isolated from the three sides with Thermocol. The relative roughness pitch (p/e) varies from 10 to 25 mm. The roughened wall has relative roughness height (e/D_h) of 0.023mm and 0.028mm, angle of attack (α) is 0° degree, rib height (e) is 1 mm and 1.25 mm. Duct aspect ratio (W/H=8), rate of air flow corresponds to Reynolds no. (Re) ranging from 3000-14000.Finally comparison of heat transfer and friction factor from both smooth and roughened plate under the similar condition of air flow is made.

Keyword: - *Solar air heater, diamond shape rib, heat transfer enhancement, pitch and Reynolds number, rib height, friction factor.*

I. INTRODUCTION

The Artificial roughness is used as turbulence promoters on a surface. It is also the technique to enhance the rate of heat transfer to the flowing fluid in a testing duct. The surface roughness can be created by number of methods such as welding, fixing small ribs, fixing small diameter of wires, machining, and sand blasting, casting and forming. Several investigators [15, 21, 22, and 24] create artificial roughness in the form of fine wires and ribs of different shapes to enhance the heat transfer coefficient. It results to increase in frictional losses which cause more power required by blower. To keep friction losses at a minimum level, the turbulence should be created very close to the duct surface i.e. laminar sub layer. Flowing air strike with ribs and break laminar sub layer which creates local wall turbulence causes flow separation and reattachment between consecutive ribs which reduce the thermal resistance and increases the heat transfer. Various studies [7, 8, 11, 13, 18, 20, 21, and 22] have shown that V-shaped ribs perform better than angled ribs. Formation of two secondary flow cells in case of V-ribs instead of one cell in case of angled rib has been cited as reason for the superior performance of V-ribs. Application of the artificial roughness in a solar air heater owes its origin to several investigations carried out for enhancement of cooling of turbine blades' passage. Several investigations have been carried out to study effect of artificial roughness on heat transfer and friction factor for two opposite roughened surfaces by Han [6], Han et al. [4,6,7], Lau et al. [8,9], Han and Zhang [11] , Taslim et al. [13] and Wright et al. [21] have developed correlations. Prasad and Saini [5], Gupta et al. [14], Karwa et al. [20], Bhagoria et al. [23], Momin et al. [25], Karwa [18] have carried out investigations on rib roughened absorber plates of solar air heaters which have only one roughened heated wall and three smooth walls. Correlations for heat transfer coefficient and friction factor have been developed for such systems. Prasad and Saini [5] used transverse small diameter wire as roughness element. Gupta et al. [14] investigated effect of relative roughness height, angle of attack on heat transfer and friction factor for inclined circular wire ribs. Karwa et al. [20] investigated effect of rib chamfer angle (U), duct aspect ratio on heat transfer and friction factor using integral chamfered ribs. Prasad and Saini [5] investigated the effect of relative roughness height (e/D_h) and relative roughness pitch (p/e) on heat transfer and friction factor using circular wire roughness. It also observed that increase in the relative roughness height results decrease in the rate of heat transfer. Increase in the relative roughness pitch results in a decrease in the rate of both heat transfer and friction factor. Nusselt number and friction factor were enhancing maximum as 2.38 and 4.25 times than that of smooth duct, respectively. Gupta et al. [12] investigated the effect of relative roughness height, angle of attack and Reynolds number on heat transfer and friction factor in rectangular duct having circular wire ribs on the absorber plate. It was found as result that the heat transfer coefficient in roughened duct improved by a factor up to 1.8 and the friction factor found as result is increased by 2.7 times that of the smooth duct. The heat transfer coefficient and friction factor were found maximum at an angle of attack of 60 and 70 respectively.Saini and Saini [14] investigate the effect of metal matrix geometry on the heat transfer coefficient and friction factor in a large aspect ratio rectangular duct, having one wall artificially roughened by an expanded metal matrix. The maximum values of Nusselt number and friction factor corresponds to angle of attack values of 61.9 and 72.The maximum enhancement in Nusselt number and friction factor values are of the order of 4 and 5, respectively. Muluwork et al. [15] compared the thermal performance of staggered discrete V-apex up and down with corresponding transverse staggered discrete ribs. The relative roughness length ratio (B/S) had been considered as dimensionless geometric parameter of roughness element to compare three different configurations. It was observed that the Stanton number increases with the increase of relative roughness length ratio. The Stanton number for V-down discrete ribs was higher than the corresponding V-up and transverse discrete roughened surfaces. The Stanton number ratio enhancement was found 1.32–2.47 in the range of parameters covered in the investigation. It was also observed that the friction factor increases with an increase in the relative roughness length ratio. Further for Stanton number, it was seen that the ribbed surface friction factor for V-down discrete ribs was highest among the three configurations investigated.Karwa et al. [16] performed experimental study to predict the effect of rib chamfer angle (Uc) and duct aspect ratio on heat transfer and friction factor in a rectangular duct roughened with integral chamfered ribs. As compared to the smooth duct, the presence of chamfered ribs on the wall of duct yields up to about two fold and three fold increases in the Stanton number and the friction factor in the range of parameters investigated. The highest heat transfer as well as highest friction factor exists for a chamfer angle (Uc) of 15°. The minima of the heat transfer function occur at roughness Reynolds number of about 20. As the aspect ratio (H/D) increases from 4.65 to 9.66, the heat transfer function also increases and then attains nearly a constant value. The roughness function decreases with the increase in the aspect ratio (H/D) from 4.65 to 7.75 and then attains nearly a constant value.Verma and Prasad [17] investigated the effect of geometrical parameters of circular wire ribs on heat transfer and friction factor. It was observed that the value of heat transfer enhancement factor (Nur/Nus) varies from 1.25 to 2.08 within the range of parameters.Momin et al. [20] experimentally investigate the effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow characteristics in rectangular duct of solar air heater. The investigation covered Reynolds number range of 2500–18,000, relative roughness height of 0.02–0.034 and angle of attack of flow of 30– 90 for a fixed relative pitch of 10. For this geometry it was observed that the rate of increase of Nusselt number with an increase in Reynolds number is lower than the rate of increase of friction factor. The maximum enhancement of Nusselt number and friction factor as result of providing artificial roughness had been found as 2.30 and 2.83 times to smooth surface respectively, for an angle of attack of 60_. It was also found that for relative roughness height of 0.034 and angle of attack of 60, the V-shaped ribs enhance the value of Nusselt number by 1.14 and 2.30 times over inclined ribs and smooth plate, respectively. It was concluded that V shaped ribs gave better heat transfer performance than the inclined ribs for similar operating conditions. Bhagoria et al. [19] performed experiments to determine the effect of relative roughness pitch, relative roughness height and wedge angle on the heat transfer and friction factor in a solar air heater roughened duct. The presence of ribs yields Nusselt number up to 2.4 times while the friction factor rises up to 5.3 times as compared to smooth duct in the range of parameters investigated. A maximum enhancement in heat transfer was obtained at a wedge angle of about 10_. The heat transfer was found maximum for a relative roughness pitch of about 7.57. The friction factor decreased as the relative roughness pitch increased. M.M.Sahu and Bhagoria [22] experimentally investigate on broken transverse ribs in solar air heaters that Reynolds number range of 3000–12,000, roughness pitch of 10–30 mm, height of the rib 1.5 mm and the aspect ratio of 8. It was found that the maximum Nusselt number attained for roughness pitch (p/e) of 20 and decreased with the increase in roughness pitch. Roughened absorber plates increased the heat transfer coefficient by 1.25–1.4 times as compared to smooth rectangular duct under similar operating conditions at higher Reynolds number.

II. INDOOR EXPERIMENTAL PROGRAM

2.1 Experimental apparatus

An indoor setup consist of long duct 2040 mm which is divided into number of parts .These are inlet section 177 mm, test section 1500 mm, mixing section and exit section 353 mm, baffles spacing 87 mm. A blower, single phase 240 volts, control valve, orifice plate and other devices such as milli voltmeter measures temperature, micro manometers measures pressure head for Reynolds no. (ASHRAE 1977) and inclined manometer for pressure measurement .A roughened absorber plate placed of length 1500 mm, on the top of the test section. Actual experimental setup shown in fig.-1.

Fig.1 Actual setup of experiment

Exit section is provided after this test section i.e.354 mm in length. The reason is to provide the exit section is to reduce the end effect in the test section to find uniform temperature at the out let, three baffles at 87 mm at equidistance are provided to mix the hot air coming out from duct. Inclined manometer is placed between blower and exit section of the duct and a control valve also provided beside the orifice plate to control the Reynolds no. The setup is covered with 25 mm thick thermocol sheet from inlet section of the duct to orifice plate to avoid heat losses. The heated GI 1mm thick sheet have diamond shaped roughness create with the help of pasting repeated diamond shaped on one side of the sheet and other side painting with black paint and fixed thermocouples on the back side of the sheet. These couples give the temperature at the different locations, the mass flow rate measures with the help of providing the inclined manometer across the orifice plate. A schematic diagram shown in fig.-2.

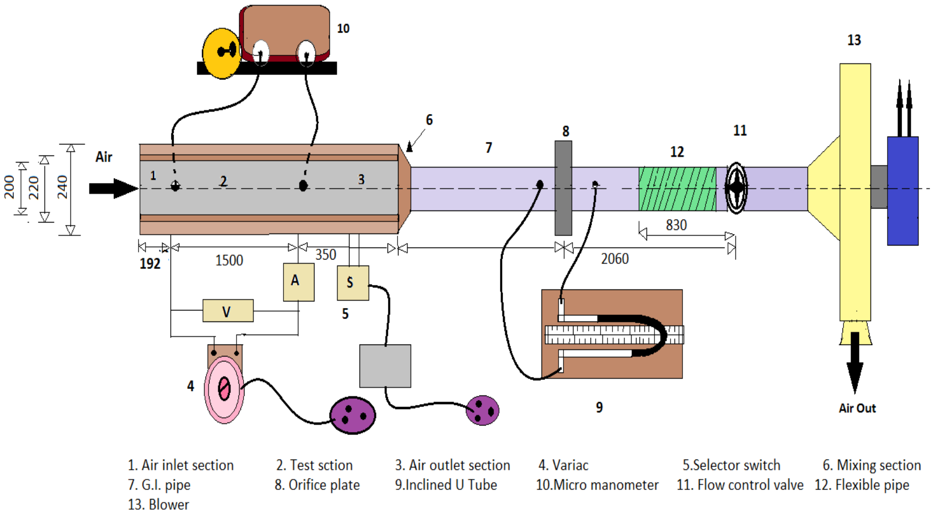


Fig.2 Schematic diagram of setup

TABLE 1 EXPERIMENTAL CONDITION

Parameter	Values
Reynolds number(Re)	3000 – 14,000
Channel aspect ratio(W/H)	8.0
Test length (L)mm	1500
Roughness height(e)mm	1 and 1.25
Relative roughness height (e/ D_h)	0.023 and 0.034
Hydraulic Diameter(D_h)mm	44.44
Roughness pitch(P)mm	10,15,20 and 25
Insulation(I) W/m^2	900-950

2.2 Absorber plates

Absorber plate made up of G.I. 1mm thickness. Roughness created by pasting regular diamond shaped of 5mm pieces and 1mm rib height on the absorber plate. Other side of the absorber plate painted by black paint also affixed thermocouples on this side. Some difficulties were experienced in getting plates manufactured to the exact dimensions. One smooth and other was artificially roughness. Figure no.-3 shows the photograph of rough plate. Fig no.4 shows the geometry of the absorber plate with diamond shape of roughness.

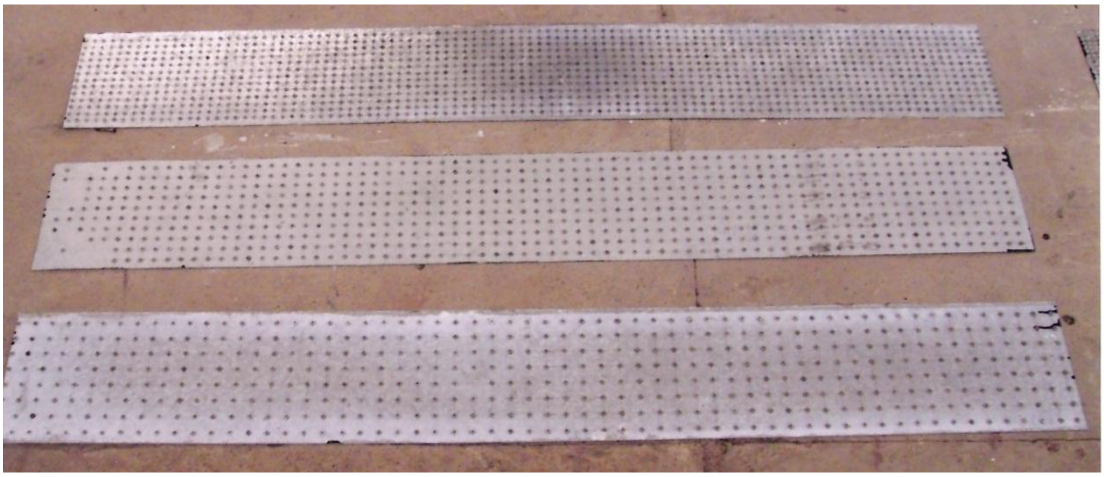
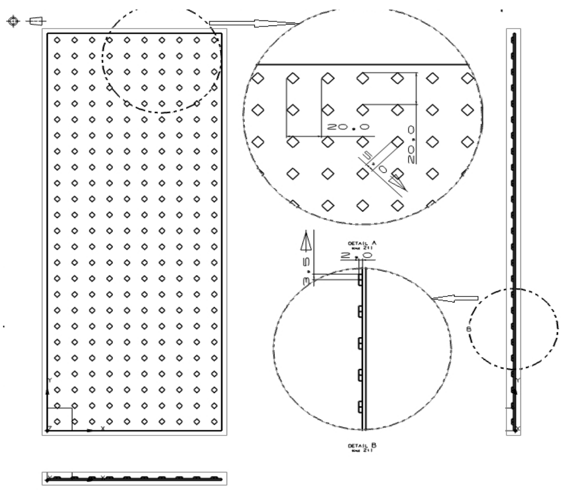


Fig.3 Rough plate of different pitches



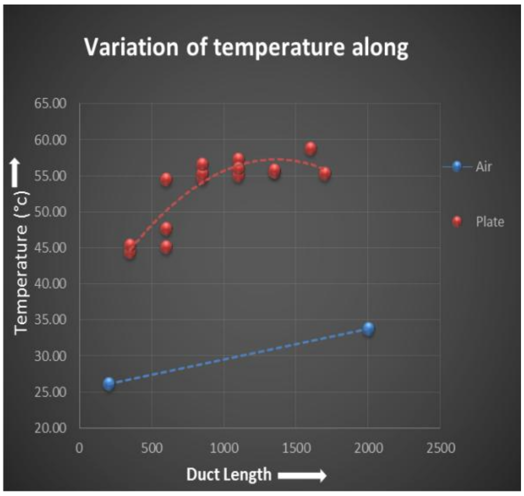


Fig. 4 Geometry of rough plate with 20mm pitch and Graph for variation of temperature along the test length

2.3 Experimental procedure

Instrument should check whether all equipment is in proper working before starting the setup. Also check all the instruments are connecting in proper way and correct. The leakage of the joints can be checked by soap and bubble treatment .Micro manometer is connected to measure pressure drop across the duct. Thermocouples are used to measure temperature of plate. Flow of air can be control with the help of control valve for the proper value of the Reynolds no. Switch on to run the blower and heater. Set the rate of flow of air according to Reynolds no. Wait for half an hour if the steady state condition reaches. Collect all the relevant data concern with setup and required data for each rib configuration and the various Reynolds no.3000-14000. This gives the various parameters which are to be measured during experiments. The parameters are needed to record is

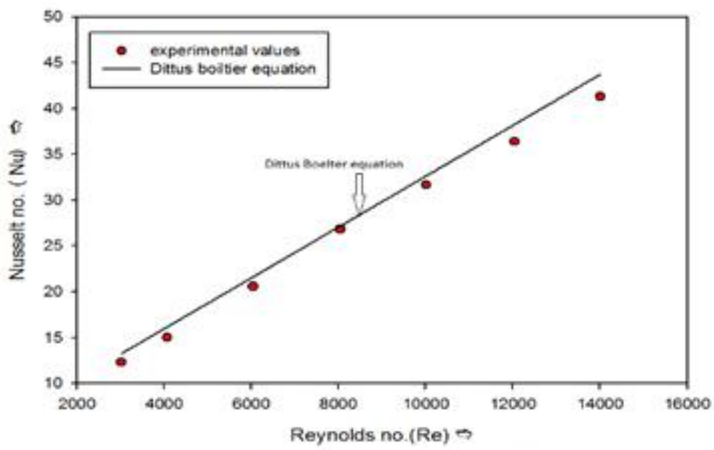
1. Inlet temperature of air at entrance of duct.
2. Outlet temperature of air at exit section of the duct.
3. Intermediate temperature of the collector.
4. Pressure drop across orifice plate measure with the help of inclined manometer.
5. Solar insulation

2.4 Validation test

Validation curve with smooth plate is taken by S.S.Pawar et al. [28] and the value of Nussselt no. and friction factor is obtained from experimental data .These data is compared with value of DittusBoelter and modified Blasius equation respectively.

DittusBoelter equation $Nu_s = 0.024Re^{0.8}Pr^{0.4}$ (1)

Modified Blasius equation $fs = 0.085Re^{-0.25}$ (2)



Velidation curve for smooth plate

Fig.5 Graph no.-1

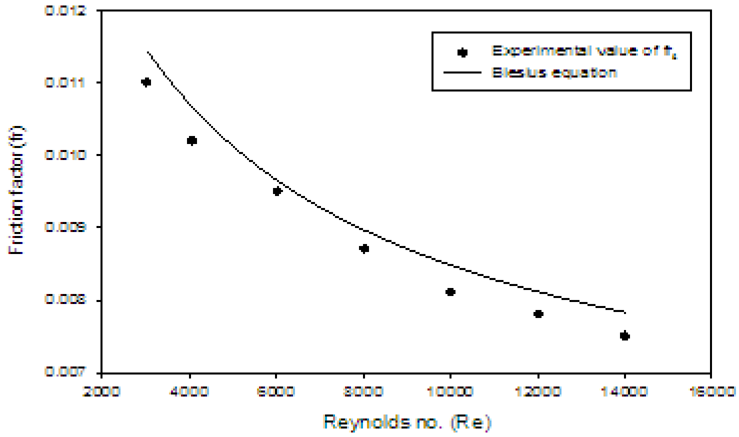


Fig.6 Graph no.-2

2.5 Variation of temperature along test duct

Setup consist four section i.e. entering section, heating section, mixing section and exit section .Air enters at atmosphere temperature in the entering section, heat added in heating section ,air mixed in mixing sections temperature reduced .Finally uniform temperature received after mixing section. Temperatures of all the four sections are noted down and are plotted as shown in graph in fig-4.

III. DATA REDUCTION

3.1 Data analysis

Table – 1 shows the experimental parameter and table 2 -3 shows the experimental data for smooth and roughened plate.

3.2 Mean Air & Plate Temperature

The mean air temperature is the simple arithmetic mean of the measure values at the inlet and exit of the test section. Thus

$$T_{fav} = (t_i + t_{oav}) /2$$

The mean plate temperature, t_{pav} is the weighted average of the reading of all points located on the absorber plate.

3.3 Pressure Drop Calculation

Pressure drop measurement across the orifice plate by using the following relationship:

$$\Delta P_o = \Delta h \times 9.81 \times \Delta_m \times 1/5$$

Where

ΔP_o = Pressure difference

$\Delta \rho_m$ = Density of the fluid (Mercury) i.e. $13.6\text{x}10^3$

Δh = Difference of liquid head in U-tube manometer, m

3.4 Mass Flow Measurement

Mass flow rate of air has been determined from pressure drop measurement across the orifice plate by using the following relationship:

$$m = C_d \times A_0 \times [2\, \rho\Delta P_0 / (1 - \beta^4)]^{0.5}$$

Where

m = Mass flow rate, kg / sec.

C_d = Coefficient of discharge of orifice i.e. 0.62

A_0 = Area of orifice plate, m^2

ρ= Density of air in Kg/m^3

β= Ratio of dia. (d_o / d_p) i.e. 26.5/53 = 0.5

3.5 Velocity Measurement:

$$V = m / \rho WH$$

Where,

m = Mass flow rate, kg / sec

ρ = Density of air in Kg/m^3

H = Height of the duct in m

W = Width of the duct, m

3.6 Reynolds Number

The Reynolds number for flow of air in the duct is calculated from:

$$R_e = VD / v$$

Where,

ν= Kinematics viscosity of air at t_{fav} in m^2/sec

D_h = 4WH / 2 (W+H) =0.04444 m

3.7 Heat Transfer Coefficient

Heat transfer rate, Q_a to the air is given by:

$$Q_a = m\, c_p\, (t_o – t_i)$$

The heat transfer coefficient for the heated test section has been calculated from:

$$h = Q_a / A_p\, (t_{pav} - t_{fav})$$

A_p is the heat transfer area assumed to be the corresponding smooth plate area.

3.8 Nusselt Number

The Heat Transfer Coefficient has been used to determine the Nusselt number defined as;

$$Nu = h\,D_h / K$$

Where k is the thermal conductivity of the air at the mean air temperature and D_h is the hydraulic diameter based on entire wetted parameter.

3.9 Thermal Efficiency

The Thermal efficiency for test section is calculated from:

$$\eta = Q_a / A_p I$$

Where, I = Heat Flux i.e. 900 W/m^2

IV. RESULT AND DISCUSSION

Heat transfer coefficient and friction factor compared roughened plate with smooth plate under similar fluid flow condition. Roughness creates by pasting regular diamond shaped rib to see the enhancement in heat transfer coefficient. Fig.3 shows the roughened plate of different pitches. Figure-4 shows the geometry of roughened plate. These graph shows as Nusselt number increases with increases in Reynolds numbers. Comparison of two rib height results takes place in this experiment. These are 1mm and 1.25mm.finally compared for higher heat transfer coefficient.

The Nusselt number found maximum at the pitch value of 15mm with Rib height 1mm.Also indicate that heat transfer coefficient is maximum at 15 mm pitch roughened plate.

It is nothing but the ratio of conductive resistance to convective resistance of heat flow and as Reynolds number increases thickness of boundary layer decreases and hence convective resistance decreases which in term increases the Nusselt number.

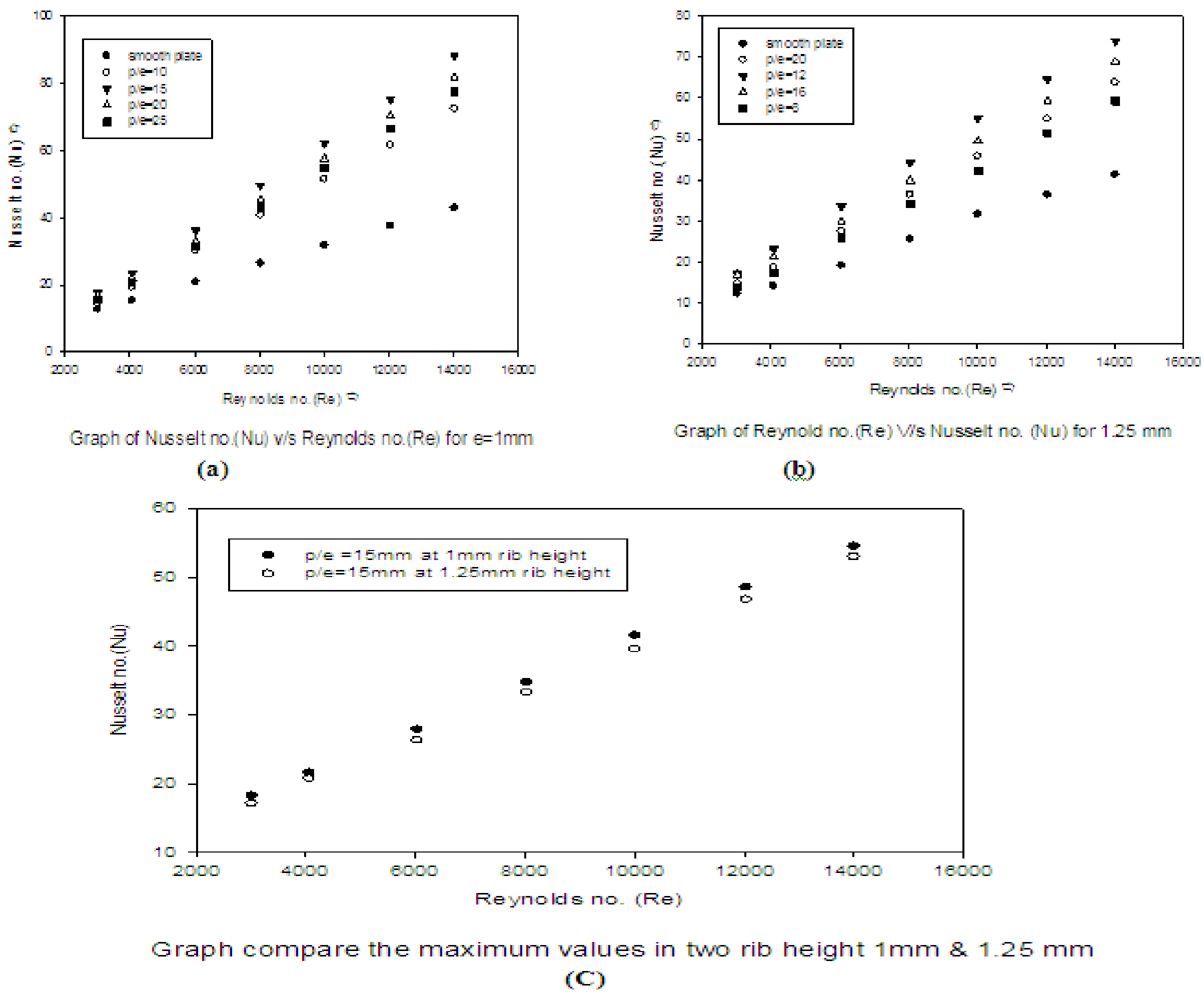
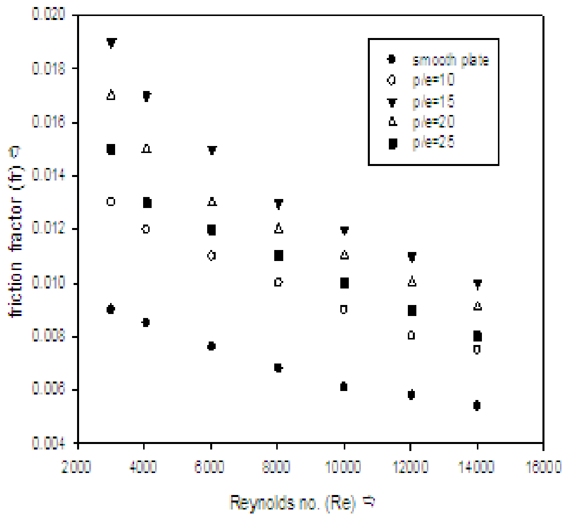
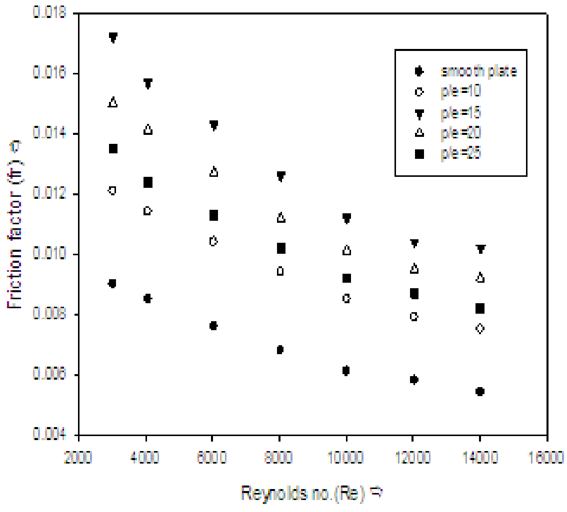


Fig. 7 (a) Nu vs Re for e=1mm (b) Nu vs Refor e=1.25mm (c)Nu vs Refor e=1mm&for e=1.25mm (c)



Graph of Renould no.(Re) with friction factor (fr) of 1mm rib height.

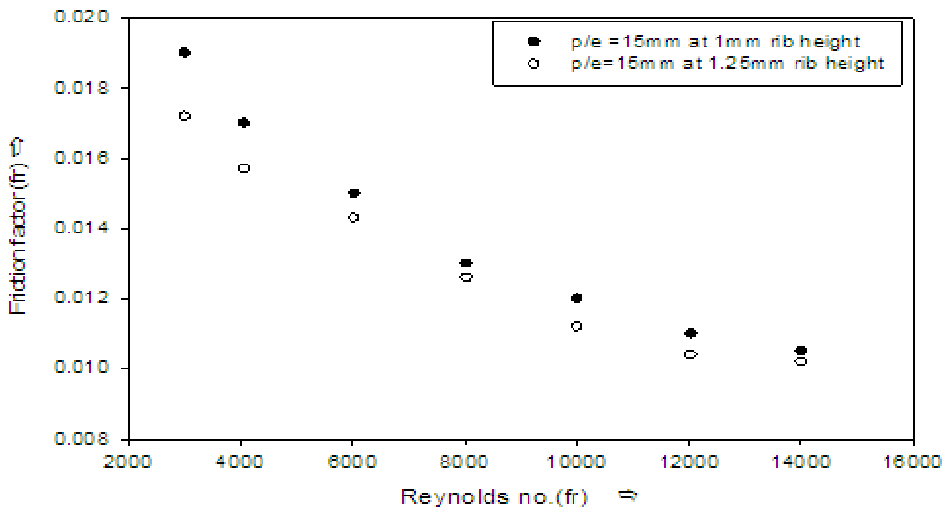
(a)



Graph for friction factor (fr) versus Reynolds no. (Re) of 1.25mm rib height at 0 deg.angle of attack

(b)

Graph for Comperation of result for 1mm and 2mm rib height



(c)

Fig. 7 (a) fvs Re for e=1mm (b) fvs Re for e=1.25mm (c) fvs Re for e=1mm & for e=1.25mm

V. CONCLUSIONS

The major conclusions of this article are as follows:

1. Presence of diamond shaped rib on the absorber plate is an effective technique to enhance the rate of heat transfer as compared to the smooth solar air heaters.
2. The Nusselt no. (Nu) and friction factor (f_r) are strongly dependent on the relative roughness pitch (P/e) and relative roughness height (e/D_h) of diamond shaped rib together with the flow Reynolds number.
3. It has been found that Nu increases with the increase in Re.
4. Maximum value of Nusselt no. (Nu) has been found to be 84.72 at a Reynolds no. (Re) of 14012.
5. It has been found that friction factor (f_r) decreases with the increase in Reynolds no. (Re)
6. Maximum value of friction factor (f_r) has been found to be 0.0194 at a Reynolds no. (Re) is 3010.

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