EXPERIMENTAL ANALYSIS FOR DETERMINING THERMAL ENHANCEMENT AND FLUID FLOW BEHAVIOR OF ROUGHENED SOLAR AIR HEATER

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ABSTRACT-*The coefficient of heat transfer of a solar air heater is low because of minimum utilization of solar energy by the absorber plate used in solar air heater. It is attributed to the formation of a very thin boundary layer at the absorber plate surface commonly known as viscous sub-layer. The coefficient of heat transfer of a solar air heater duct can be increased by providing artificial roughness on the heated wall (i.e. the absorber plate).Providing artificial roughness makes the flow turbulent and disturbs the viscous sub layer of the flowing medium. Comparison is made by carrying out experiment using the different parameters. The result predicts a significant enhancement of heat transfer in comparison to that of for a smooth surface with different 'P' and various range of Reynolds number.*

Keyword:- heat transfer , efficiency, absorber plate, laminar sub layer.

INTRODUCTION

Solar Air Heater is a type of solar thermal system where air is heated in a collector and either transferred directly to the interior space or to a storage medium. The value of heat transfer coefficient between the absorber plates and air is low and this result in lower efficiency. For this reason, the surfaces are sometimes roughened or longitudinally fins are provided in the air flow passage. A roughness element has been used to improve the heat transfer coefficient by creating turbulence in the flow. However, it would also result in increase friction losses and hence greater power requirement for pumping air through the duct. In order to keep the friction losses at the low level, the turbulence must be created only in the region very close to the duct surface i.e. in laminar sub layer.

A roughness type which can be employed for such purpose can be described by the dimensionless geometrical parameters namely the relative roughness height, the relative roughness pitch (P/e: the ratio of the distance of two successive roughness elements to the height of the roughness), the angle of attack (α: the angle of the roughness rib with respect to the direction of flow) and the shape of the roughness elements.

The focus is on use of V shaped ribs on absorber plate used in solar air heater. The result for absorber plate which gives the heat performance in respect of heat transfer coefficient has to be identified.

METHODOLOGY

Experimental Set Up

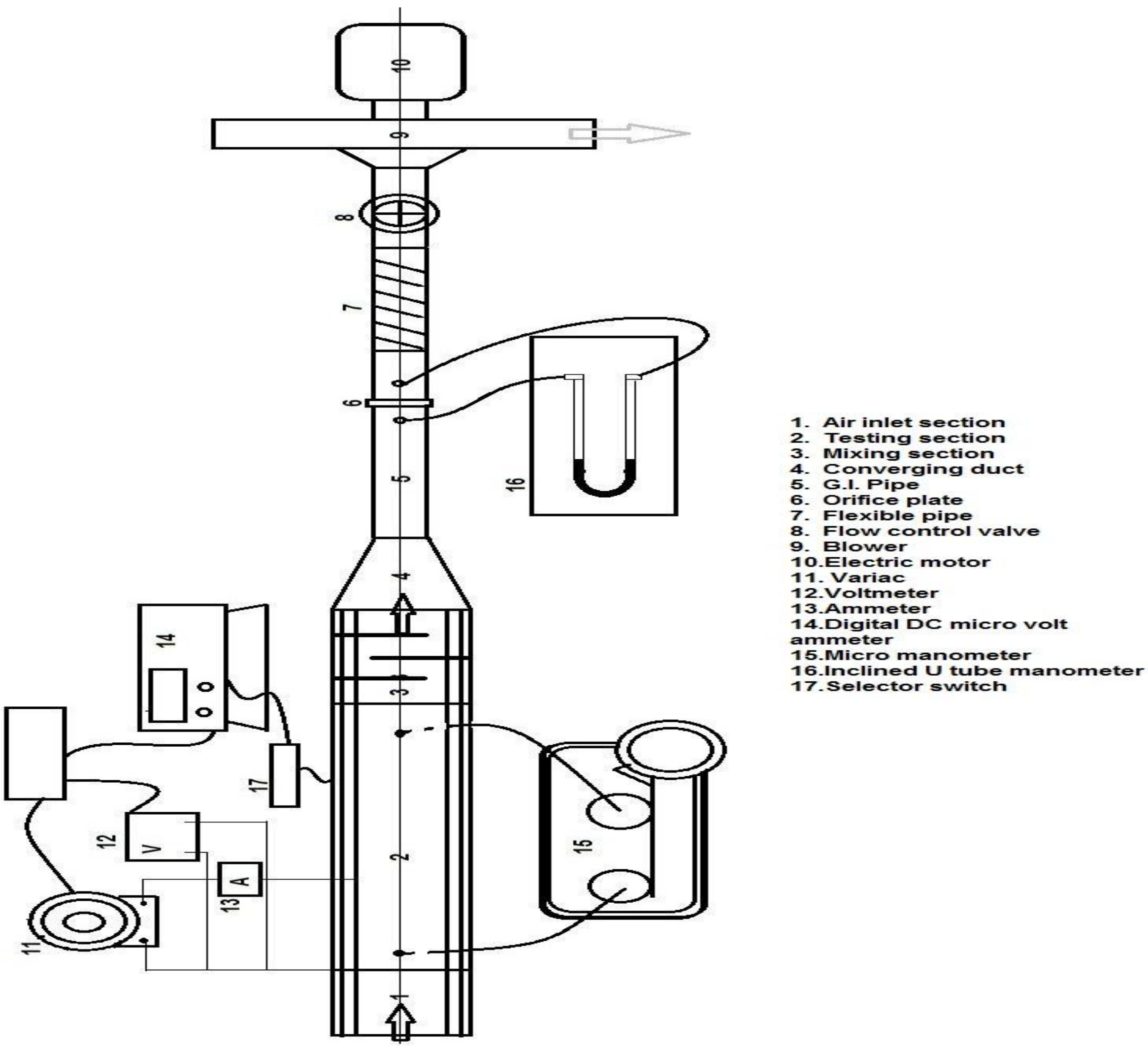


Figure1: Schematic Diagram of Experimental Set up Showing Top View

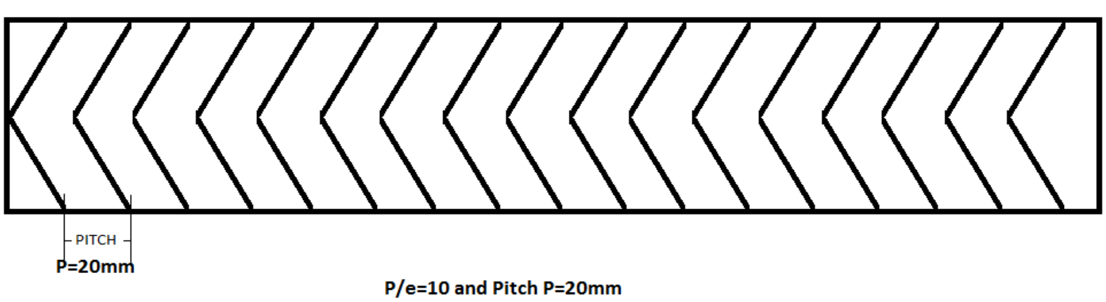


Figure2: Absorber plate having P/e=10, Pitch(P)=20mm

Formulae

1. **Mean bulk air temperature (T_f)**

The mean air temperature or average flow temperature T_f is the simple arithmetic mean values at the entrance and exit of the test section. Thus,

$$T_f = (T_i + T_{oav})/2$$

where,

T_f = mean bulk air temperature (K),
T_i = air inlet temperature (K) and
T_{oav} = air outlet temperature (K).

2. The mean plate temperature (T_{pav}) is the average of the reading of Eight points located on the absorber plate.

3. **Pressure Drop Calculation**

Pressure drop measurement on the orifice plate was calculated by following equation,

$$\Delta P_0 = \Delta h \times 9.81 \times p_m \times 1/5$$

where,

ΔP_0 = pressure drop across the orifice plate (N/m^2),
Δh = difference of liquid head in U-tube manometer in (m),
p_m= density of fluid (kg/m^3)

4. **Mass Flow Measurement**

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

where,

m = mass flow rate of air (kg/s),
C_d= coefficient of discharge of orifice meter, i.e.0.62
A_o= throat area of orifice meter (m^2),
ΔP_0 = pressure drop across orifice meter (N/m^2) and
β = the ratio of orifice diameter to the pipe diameter, (d_o/d_p)i.e. (26.5/53=0.5)
p= density of air in kg/m^3

5. **Velocity Measurement**

$$V = [m / (\rho WH)]$$

Where,

m = mass flow rate of air (kg/s),
ρ = Density of air in (kg/m^3),
H= height of the duct in m (0.025)
W = Width of the duct in m (0.2)

6. **Reynolds Number**

$$Re = [VD_h /\nu]$$

where,

Re = Reynolds Number,
V = Velocity of air (m/s),
D_h= hydraulic diameter, $D_h = 4WH/2\ (W + H)$
ν = kinematic viscosity (m^2 /s)

7. **Heat gained by air**

$$Q_a = [\ m \times C_p \times (T_{oav} - T_i)]$$

where,

Q_a = heat transfer rate to air (W),
m = mass flow rate of air (kg/s),
C_p= Specific heat of air at constant pressure (KJ/kgK)
T_i = air inlet temperature (K) and
T_{oav} = air outlet temperature (K).

8. **Convective Heat transfer coefficient**

$$h = Q_a/[A_p \times (T_{pav} - T_f)]$$

where,

Q_a = heat transfer rate to air (W),
h = Convective heat transfer coefficient (W/m^2 K),
T_{pav}= average plate temperature (K) and
T_f= mean air temperature (K),
A_p = surface area of absorber plate (m^2).

9. **Nusselt Number**

$$Nu = hD_h / k$$

Where,

Nu = Nusselt Number
h = heat convective transfer coefficient (W/m^2),
k= Thermal conductivity of air (W/m-K),
D_h= hydraulic diameter (m).

10 **Properties of Air**

(a) Specific heat C_p= 1006.74 J/kg K

(b) Kinematic viscosity(ν) in Ns/m^2 = 1.621768×10^{-5}

(c) Thermal conductivity = 0.0268 W/mK

(d) Characteristic gas constant (R) = 287.045 J/kg K

(e) Density of air (ρ) in kg/m^3 = 1.153070533

PROCEDURE

Observation is made for Δh = 0.009m , 0.054m , 0.108m , 0.163m, 0.231m , 0.326m in case of smooth plate , plate 1(p/e=10) and plate 2(p/e=12) and temperature of air at inlet , temperature of air at outlet, temperature of plate at 8 points are noted. Calculation carried out using the formulae.

RESULTS AND DISCUSION

The data is collected experimentally by using different components and instruments for different roughness pitch of absorbing plate. The flow of air changes by using control valve and the data of roughened plate is compared with the smooth plate. The data obtained through experiment has been used to determine the nusselt number, heat transfer coefficient.

Table 1: Experimental Parameters

PARAMETERS	VALUE
Roughness height (e)	**2mm**
Reynolds number (Re)	**4000-14000**
Hydraulic diameter(Dh)	**0.04444**
Relative roughness pitch (P/e)	**10, 12**
angle of attack (α)	**60^o**
Material of absorbing plate	**GI sheet**
Aspect ratio of Duct (W/H)	**8**
Testing length	**1500mm**
Heat flux (l)	900 W/m^2K

Table 2:Reynolds No., Heat transfer coeff.,Nusselt No.,for Plate 1 (p/e=10)

S.No.	Reynolds Number	Heat transfer Coeff.	Nusselt Number
1	3947	10	17
2	5658	19	32
3	7976	28	46
4	9795	32	53
5	11660	34	57
6	13936	39	64

Graph 1: Nusselt No. Vs Reynolds No. (Plate 1, p/e=10)

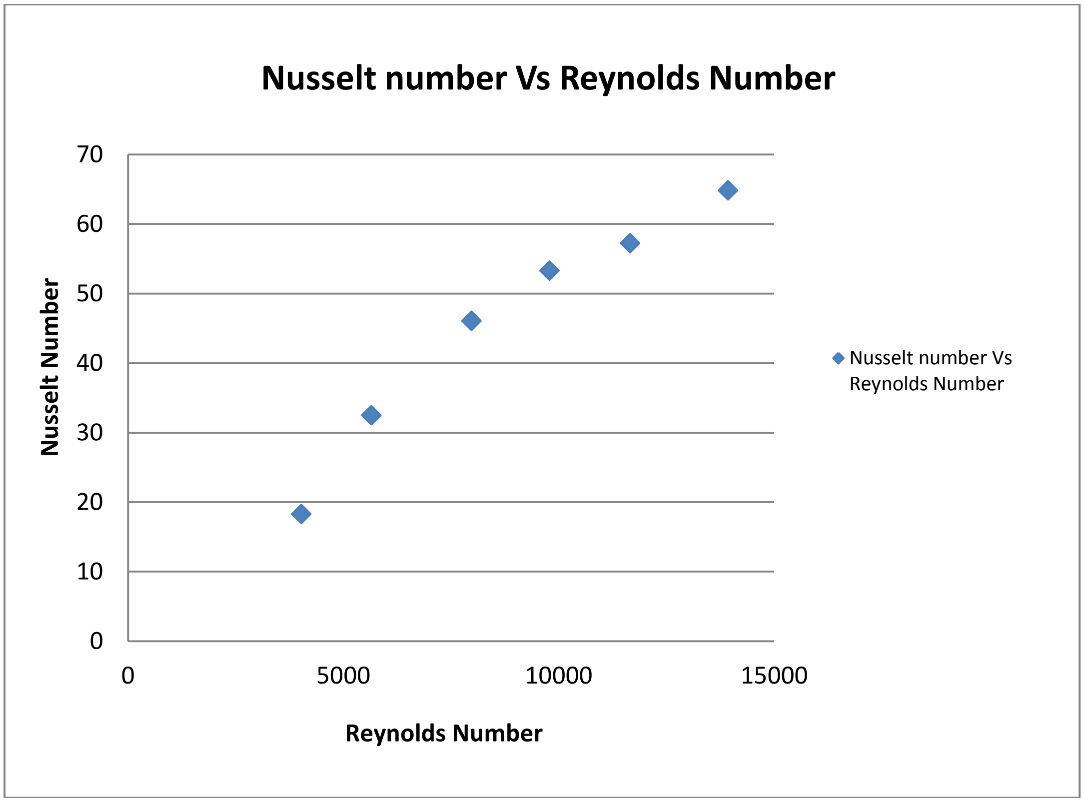


Table 3: Reynolds No., Heat transfer coeff., Nusselt No., for Plate 2 (p/e=12)

S.No.	Reynolds Number	Heat transfer Coeff.	Nusselt Number
1	4059	11	19
2	5702	19	32
3	8093	26	44
4	9948	31	51
5	11914	33	55
6	14192	38	63

Graph 2: Nusselt No. Vs Reynolds No. (Plate 2, p/e=12)

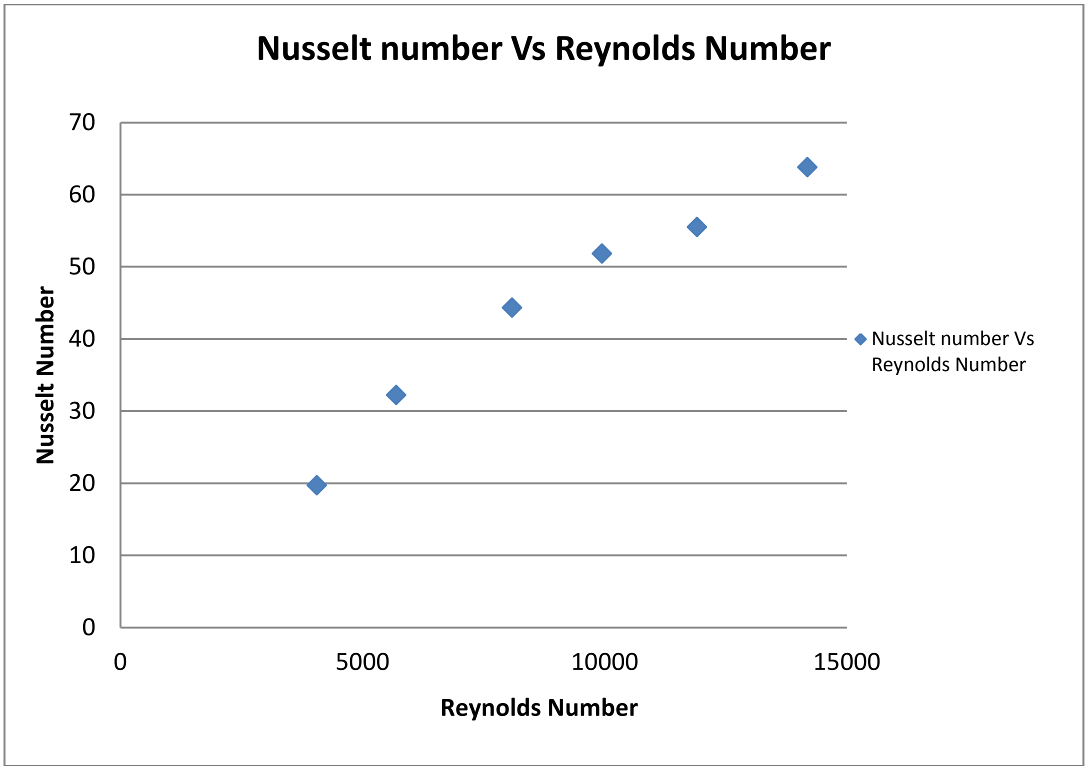
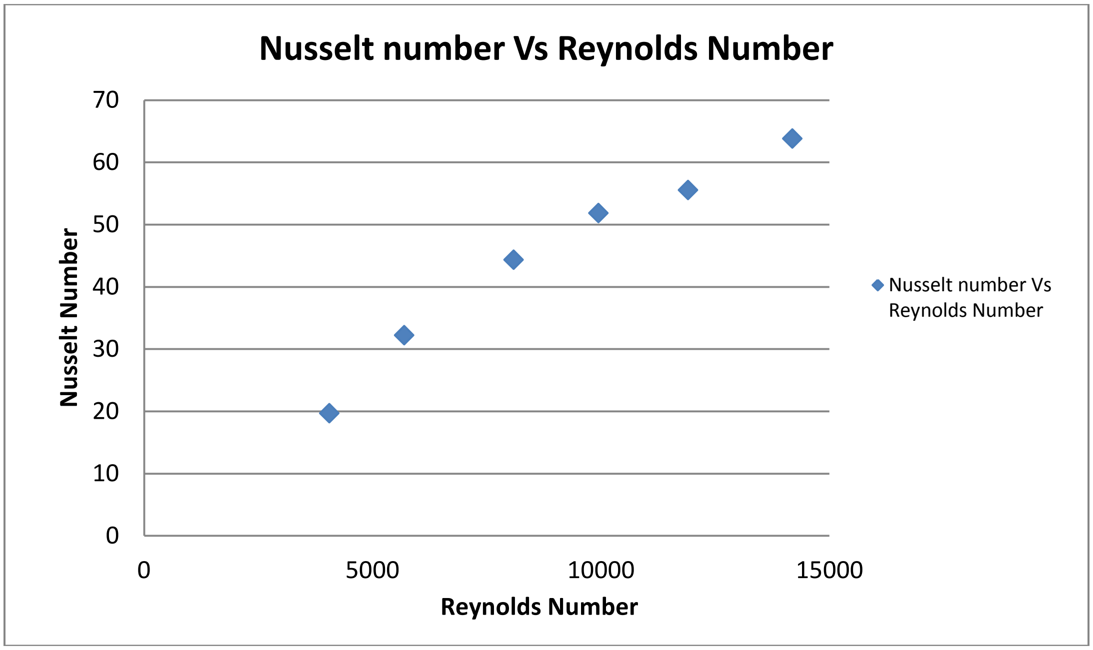


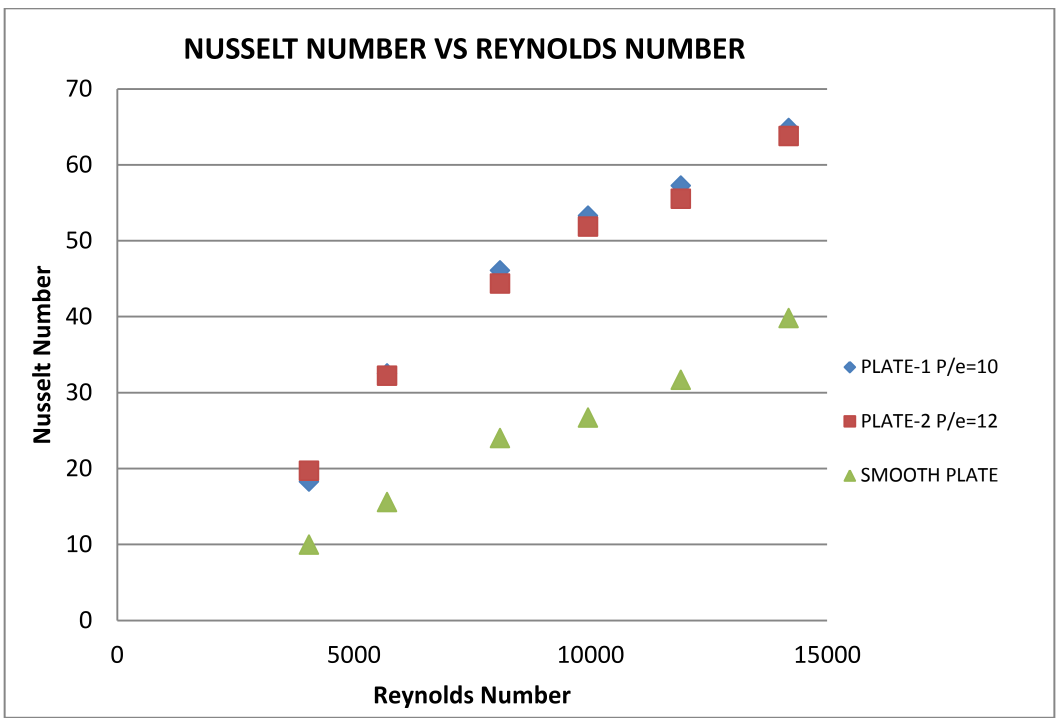
Table 4:Reynolds No., Heat transfer coeff., Nusselt No., for Smooth Plate

S.No.	Reynolds Number	Heat transfer Coeff.	Nusselt Number
1	4001	6	9
2	5994	9	15
3	7996	14	23
4	9986	16	26
5	11900	19	31
6	13995	24	39

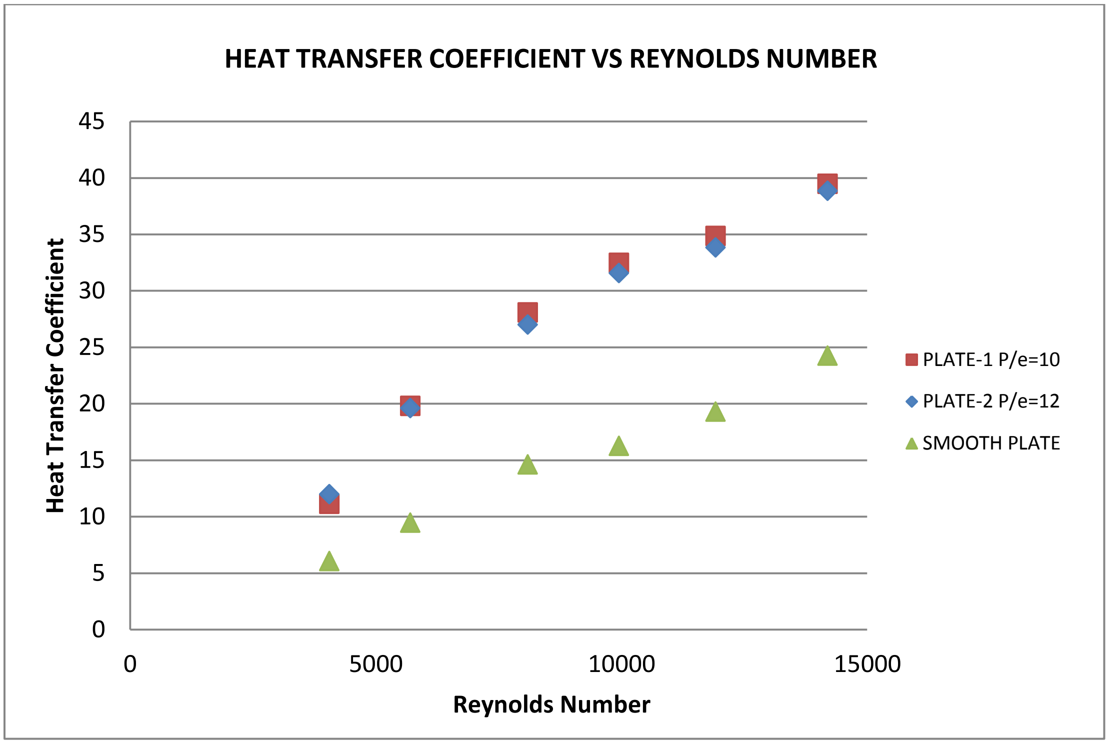
Graph 3: Nusselt No. Vs Reynolds No. (Smooth Plate)



Graph 4: Comparison b/w Nusselt No. Vs Reynolds No.



Graph 5: Comparison b/w Heat transfer Coeff. Vs Reynolds No.



CONCLUSION

In the view of present analysis the following conclusions are drawn:

1) Convective heat transfer coefficient and Nusselt number increases with the increase in Reynolds number for constant V-shaped and for same mass flow rate as shown in experiment. It shows the different values for different roughness pitch.

2) The predicted values shownusselts number for plate 1 comparable with experimental results.

3) Experimental work shows maximum enhancement in Nusselt No. of 64 at Reynolds No. 13936 with usage of roughness pitch P=20. This is more than nusselt No.of 63 at Reynolds No. of 14192 with usage of roughness pitch P=24. The value obtained in case of smooth plate is 39 for Nusselt No. at Reynolds No. 13995. Thus, we observe percentage increase in Nusselts No. being more with usage of roughness pitch=20 when compared to smooth plate.

4) Further, experimental workshowsmaximum enhancement in heat transfer coefficient to 39 at Reynolds No. of 13936 with usage of roughness pitch, P=20. This more than heat transfer coefficient of 38 at Reynolds No. of 14192 with usage of roughness pitch P=24. The value obtained in case of smooth plate is 24 for heat transfer coefficient at Reynolds No. of 13995. Thus, we observe percentage increase in heat transfer coefficient being more with usage of roughness pitch=20 when compared to smooth plate. Hence, it is recommended to use V shaped ribs with roughness pitch of P=20 (roughness height, e=2mm).

REFERENCES

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

[2] M.M. Sahu, J.L. Bhagoria, Augmentation of heat transfer coefficient by using 90° broken transverse ribs on absorber plate of solar air heater, Renewable Energy (30) (2005) 2057–2073.

[3] R.P. Saini, JitendraVerma. Heat transfer and friction factor correlations for a duct having dimple-shape artificial roughness for solar air heaters. Energy 33 (2008) 1277– 1287.

[4] Gupta Dhananjay, Solanki SC, Saini JS. Heat and fluid flow in rectangular solar Air heater ducts having transverse rib roughness on absorber plate. Solar Energy 1993; 51:31–7.

[5] J.C. Han, J.S. Park, C.K. Lei.Heat transfer enhancement in channels with turbulence promoters. J. Eng. Gas Turb.Power 107 (1985) 628–635.

[6] Y.M. Zhang, W.Z. Gu, J.C. Han, Heat transfer and friction in Rectangular channel with ribbed or ribbed-grooved walls. ASME/J.Heat Transfer 116 (1994) 58–65.

[7] Liou TM, Hwang J.J. Effect of ridge shapes on turbulent heat transfer and Friction in a rectangular channel. International Journal of Heat and Mass Transfer 1993; 36:931–40.

[8] AlokChaube, P.K. Sahoo, S.C. Solanki. Analysis of heat transfer augmentation and flow characteristics due to rib roughness over absorber plate of a solar air heater, Renewable Energy 31 (2006) 317–331.

[9] R. Kamali, A.R. Binesh. The importance of rib shape effects on the local heat transfer and flow friction characteristics of square ducts with ribbed internal Surfaces. International Communications in Heat and Mass Transfer 35 (2008) 1032–1040.

[10] Han, J.C., Chandra, P.R., Lau, S.C., 1988. Local heat/mass transfer distributions around sharp 180 deg turns in two-pass smooth and rib roughened channels. J. Heat Transfer 110 (February), 91–98.