RESEARCH ARTICLE OPEN ACCESS

Effect of Artificial Roughness on Solar Air Heater: An Experimental Investigation

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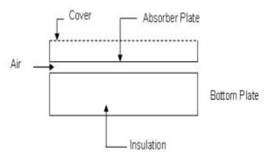
ABSTRACT:
It is well known, that, the heat transfer coefficient between the absorber plate and working fluid of solar air heater is low. It is attributed to the formation of a very thin boundary layer at the absorber plate surface commonly known as viscous sub-layer The heat transfer coefficient of a solar air heater duct can be increased by providing artificial roughness on the heated wall (i.e. the absorber plate) The use of artificial roughness on the underside of the absorber plate disturbs the viscous sub-layer of the flowing medium. It is well known that in a turbulent flow a sub-layer exists in the flow in addition to the turbulent core. The purpose of the artificial roughness is to make the flow turbulent adjacent to the wall in the sub-layer region. Experiments were performed to collect heat transfer and friction data for forced convection flow of air in solar air heater rectangular duct with one broad wall roughened by discrete v –groove & v- shape ribs. The range of parameters used in this experiment has been decided on the basis of practical considerations of the system and operating conditions. The range of Reynolds number of 3000-14000, Relative Roughness Height (eh/D) of height 0.030 to 0.035, Rib angle of attack 60^0, heat flux 720 W/m^2 and pitch of relative roughness pitch 10 the Result has been compared with smooth duct under similar flow and boundary condition It is found from the investigation that on increasing the roughness of a roughened plate the friction factor andheat transfer performance of solar air heater increase and the rate of increase of heat transfer performance of solar air heater get reduced as the roughness of plate increases.
Keywords:-Solar Air Heater, Duct, Absorber Plate, ArtificialRoughness, Reynolds Number,

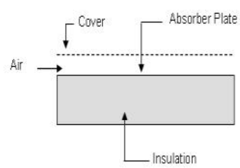
I. Introduction

Solar air heaters, because of their inherent simplicity, are cheap and most widely used as collection device. The thermal efficiency of solar air heaters has been found to be generally poor because of their inherently low heat transfer capability between the absorber plate and air flowing in the duct. In order to make the solar air heaters economically viable, their thermal efficiency needs to be improved by enhancing the heat transfer coefficient. In order to attain higher heat transfer coefficient, the laminar sub-layer formed in the vicinity of the absorber plate is broken and the flow at the heat-transferring surface is made turbulent by introducing artificial roughness on the surface. Various investigators have studied different types of roughness geometries and their arrangements.

A conventional solar air heater generally consists of an absorber plate with a parallel plate below forming a passage of high aspect ratio through which the air to be heated flows. As in the case of the liquid flat-plate collector, a transparent cover system is provided above the absorber plate, while a sheet metal container filled with insulation is 'provided on the bottom and sides. The arrangement is sketched in fig. 1.1 Two other arrangement, which are not so common are also shown in fig 1.1 In the arrangement shown in fig 1.1, the air flows between the cover and absorber plate; as well as through the passage below the absorber plate.

However, the value of the heat transfer coefficient between the absorber plate and air is low and this result in lower efficiency. For this reason, the surfaces are sometimes roughened or longitudinal fins are provided in the airflow 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 in friction losses and hence greater power requirements for pumping air through the duct. In order to keep the friction losses at a low level, the turbulence must be created only in the region very close to the duct surface, i.e. in laminar sub layer.





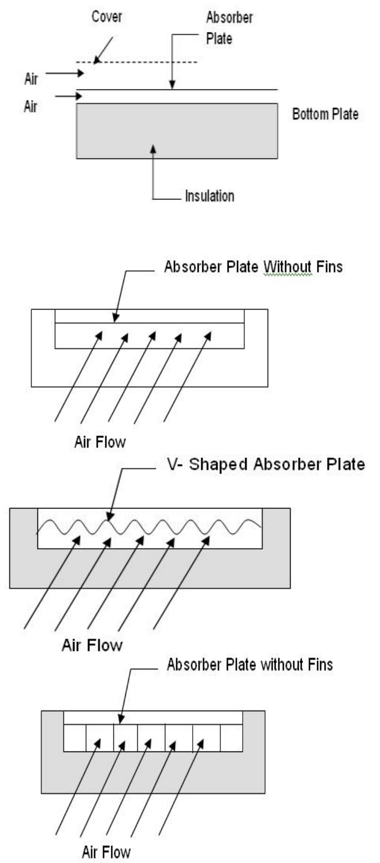
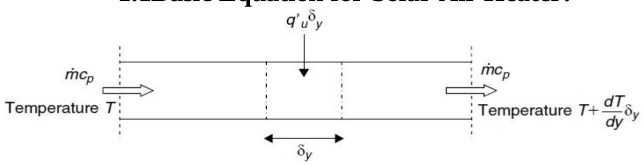


Fig-1Various flow arrangements in solar air heater

1.1Basic Equation for Solar Air Heater:



The useful heat gain of the air is calculated as:

$$Q_u = m'C_p(Tf_o - Tf_i)$$

Where,
m' is mass flow rate of air through the test duct (kg/sec)
Cp is specific heat of air
Tf_o is fluid temperature at exit of test duct
Tf_i is fluid temperature at inlet of test duct
The heat transfer coefficient for the test section is:

$$h = Q_u/A \cdot (T_{pm} - T_{fm})$$

where,
T_{pm} is the average value of the heater surface temperatures,
$T_{fm}$is the average air temperature in the duct = ($Tfi + Tfo$)/2

The Nusselt number:

$$Nu = h \cdot D_h/K_{air}$$

D_h is hydraulic mean diameter of test duct
h is convective heat transfer coefficient
K_{air} is thermal conductivity of air

The friction factor was determined from the measured values of pressure drop across the test length:

$$f = (\Delta P)D_h/(2\rho_{air}LV_{air}^2)$$

ΔP is pressure drop in the test duct
ρ is density of air
L is test duct length
V_{air} is average velocity of air
Thermal Performance (overall enhancement ratio)
$(Nu_r/Nu_s)/(f_r/f_s)^{1/3}$

1.2 Mean Air & Plate Temperature

Tile mean air temperature or average flow temperature flow 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 6 points located on the absorber plate.

1.3 Pressure Drop Calculation

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

$\Delta P_o = \Delta h \times 9.81 \times 1$

Where, ΔP_o = Pressure diff.

ρ_m = Density of the fluid (kerosene) i.e. 0.8×10^3

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

1.4 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

1.5 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

1.6 Reynolds Number

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

$R_e = VD / \nu$

Where,
ν = Kinematics viscosity of air at t_{fav} in m^2/sec
$D_h = 4WH / 2 (W+H) = 0.04444$

1.7 Heat Transfer Coefficient

Heat transfer rate, Qa 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:

$h = Qa / A_p (t_{pav} - t_{fav})$

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

1.8 Nusselt Number

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

Nusselt No. (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.

1.9 Thermo hydraulic performance

Heat transfer and friction characteristic of the roughened duct shows that enhancement in heat transfer is , in general , accompanied with friction power penalty due to a corresponding increase in the friction faceted. Therefore it is essential to determine the geometry that will result in maximum enhancement in heat transfer with minimum friction penalty. In order to achieve this object of simultaneous consideration of thermal as well hydraulic performance, i.e. thermo hydraulic performance,

$\square_{hp} = (N_u / N_{us}) / (f_r/f_s)^{1/3}$

A value of this parameter higher then unity ensure the fruitfulness of using an enhancement device and can be used to compare the performance of a number of arrangement to decide the best among these. The value of this parameter for the roughness geometries are investigated.

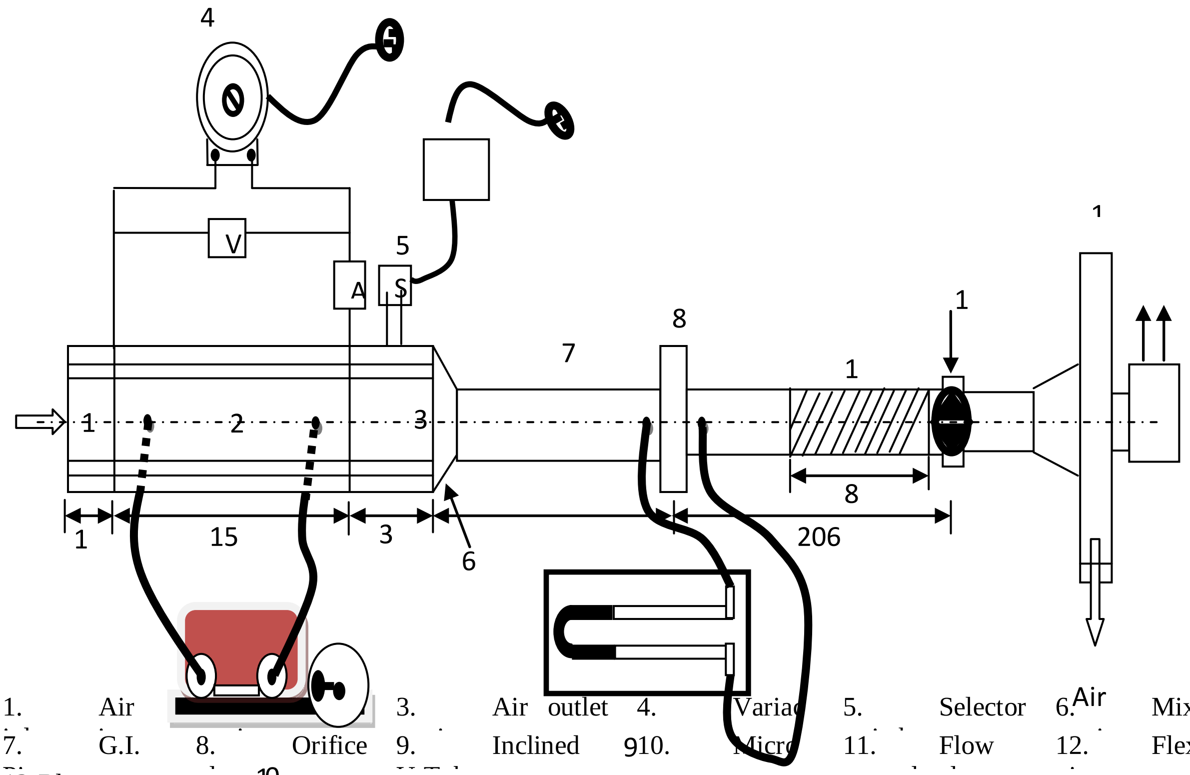
II. OBJECTIVES OF PRESENT INVESTIGATION

Forced convection heat transfer in smooth and roughened ducts has been investigated by several investigators, and a large amount of useful information is in tile literature. The use of artificial roughness on a surface is an effective technique to enhance heat transfer to fluid flowing in the duct.

The application of artificial roughness in the form of fine wires and staggered inclined ribs of different shapes has been recommended to enhance the heat transfer coefficient by several investigators. Roughness elements have been used to improve the heat transfer coefficient by creating turbulence in the flow. However, it would also result in an increase in friction losses and hence greater power requirements for pumping air through the duct. In order to keep the friction losses at a low level, the turbulence must be created only in the region very close to the duct surface, i.e. in the laminar sub layer. A number of investigations have been carried out on the heat transfer characteristics of channels or pipes with roughness elements on the surface.

Our objective is to investigate the effect discrete v & v-groove shaped roughness on the absorber plate of solar air heater, on the heat transfer coefficient and friction factor and to compare it with smooth absorber plate to know the actual increase in performance of flat plate solar air collector by using this particular artificial roughness on absorber plate. The steps to be followed in present experimental investigation are:

- 1) Setting of experimental setup for solar air heater.
- 2) Preparation of artificially roughened plate.
- 3) Data collection for roughened and smooth absorber plate.
- 4) Heat Transfer analysis.



1. Air 3. Air outlet 4. Variac 5. Selector 6.Air Mixing
7. G.I. 8. Orifice 9. Inclined 910. Micro 11. Flow 12. Flexible
13. Blower 10

Fig 2.Schematic diagram showing top view of experimental Setup

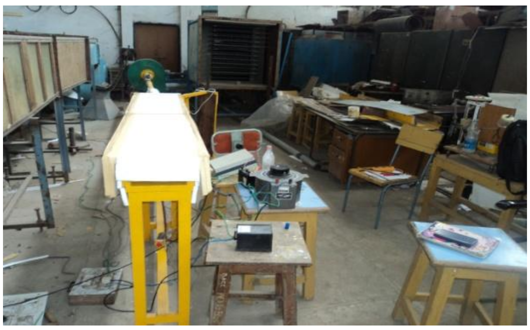


Fig 3.Experimental setup

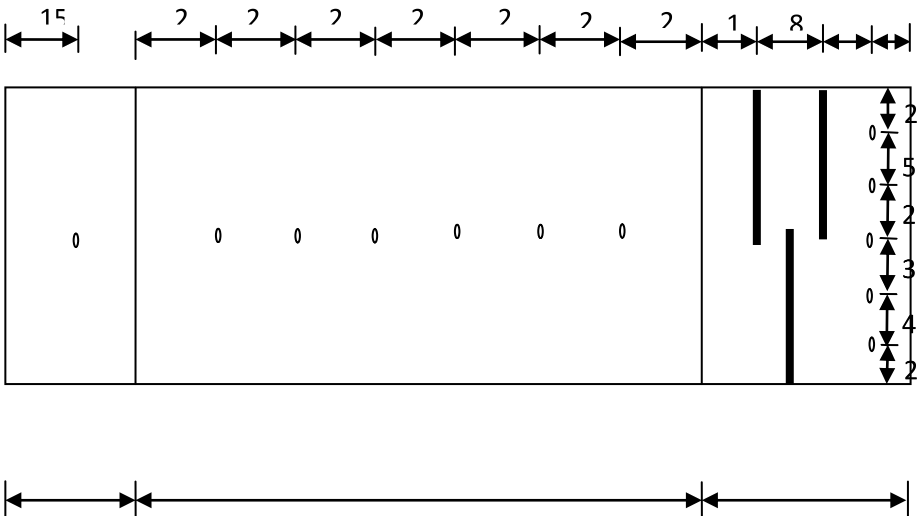


Fig 4.Location of thermocouple in the inlet section, on the absorber plate and in the outlet section

III. EXPERIMENTAL SET-UP

The experimental schematic diagram set-up including the test section consists of an entry section, a test section, an exit section, a flow meter and a centrifugal blower. The duct is of size 2042mm x 200 mmX20mm (dimension of inner cross-section) and is constructed from wooden panels of 25 mm thickness. The test section is of length 1500mm (33.75 D_h). The entry and exit lengths were 192 mm (7.2 D_h) and 350 mm (12 D_h), respectively.

A short entrance length (L/D_h=7.2) was chosen because for a roughened duct the thermally fully developed flow is established in a short length 2-3 hydraulic diameter [24]. For the turbulent flow regime, ASHRAE standard 93-77 [24] recommends entry and exit length of $5\sqrt{W_H}$ and $2.5\sqrt{W_H}$, respectively.

In the exit section after 116 mm, three equally spaced baffles are provided in a 87 mm length for the purpose of mixing the hot air coming out of solar air duct to obtain a uniform temperature of air (bulk mean temperature) at the outlet.

An electric heater having a size of 1500 mm x 216 mm was fabricated by combining series and parallel loops of heating. Mica sheet of 1 mm is placed between the electric heater and absorber plate. This mica sheet acts as an insulator between the electric heater and absorber plate (GI plate). The heat flux may be varied from 0 to 4500 W/m^2 by a variac across it.

The outside of the entire set-up, from the inlet to the orifice plate, is insulated with 25 mm thick polystyrene foam having a thermal conductivity of 0.037 W/m- K. The heated plate is a 1 mm thick GI plate with integral rib-roughness formed on its rear side and this forms the top broad wall of the duct, while the bottom wall is formed by 1 mm aluminium plate and 25 mm wood with insulation below it. The top sides of the entry and exit sections of the duct are covered with smooth faced 8 mm thick plywood.

The mass flow rate of air is measured by means of a calibrated orifice meter connected with an inclined manometer, and the flow is controlled by the control valves provided in the lines. The orifice plate has been designed for the flow measurement in the pipe of inner diameter of 53 mm, as per the recommendation of Preobrazhensky [25]. The orifice plate is fitted between the flanges, so aligned that it remains concentric with the pipe.

OBSERVATION TABLE

S. No.	**Reynolds no. (Re)**	**Inlet temperature of air (t_i) ^{O}C**	**Average outlet temperature (t_{oav}) ^{O}C**	**Average air temperature (t_{fav}) ^{O}C**	**Average plate temperature (t_{pav}) ^{O}C**	**Heat transfer Q (Watt)**	**Convective heat transfer coffecient (h) W/m^2-^{o}K**	**Nusselt no. (Nu)**	**Friction Factor (f)**	**Thermo hydraulic performance**
1	5387	34.00	46.00	40.00	72.28	136.80	14.12	22.57	0.032	0.5
2	7604	33.50	44.00	38.75	70.48	169.00	17.75	28.37	0.0275	0.69
3	9315	33.00	42.00	37.50	66.00	178.00	20.81	33.23	0.025	0.853
4	10788	33.00	41.00	37.00	63.00	182.00	23.33	37.26	0.023	1.0
5	12051	32.00	39.50	36.00	61.20	191.30	25.26	40.37	0.022	1.1
6	13211	31.50	39.00	35.2	60.00	209.00	27.80	44.40	0.021	1.07

IV. EXPERIMENTAL RESULTS

Tile following results have been obtained from the experiment

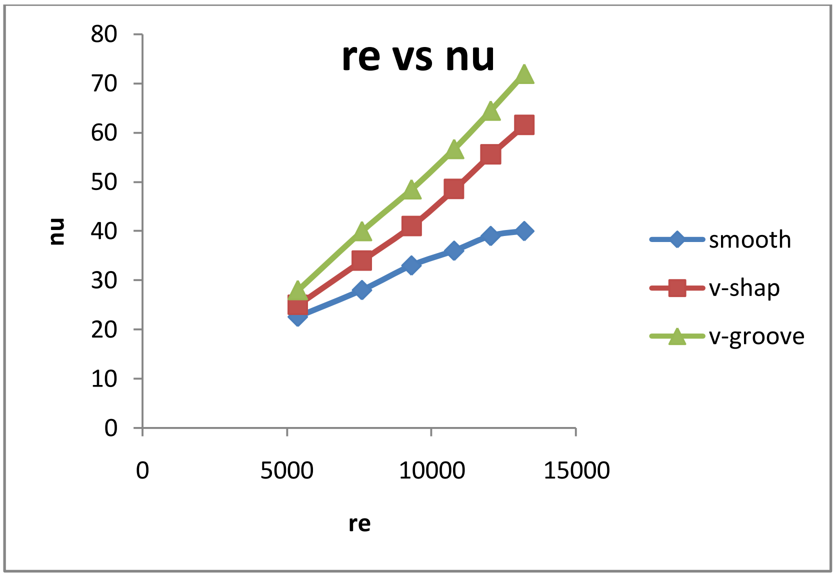


Fig-Reynolds numbers vsNusselt number

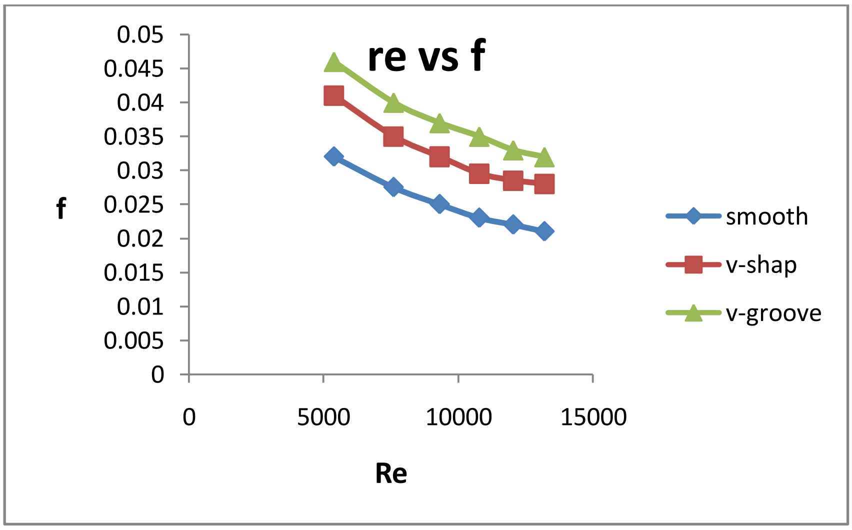


Fig-Reynolds numbers vs friction factor

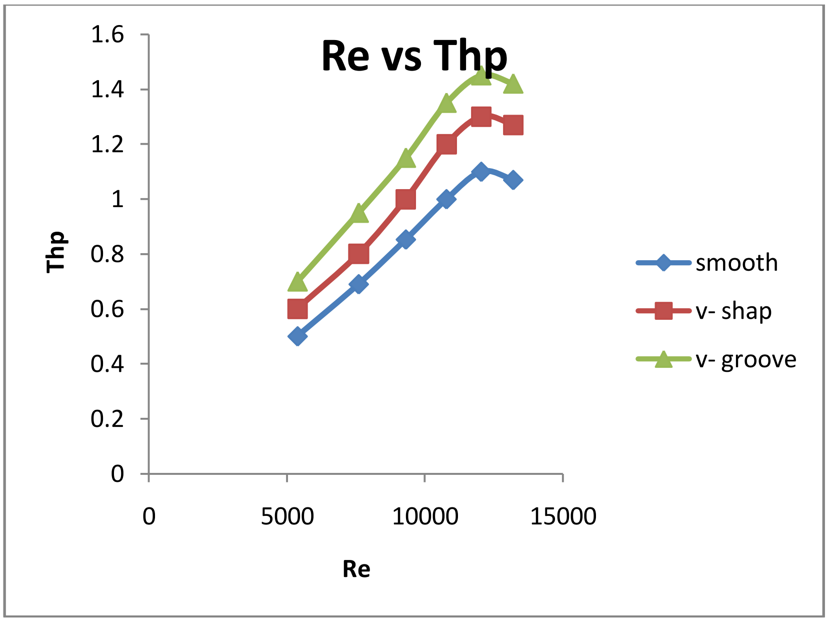


Fig-Reynolds numbers vsThermo hydraulic performance

V. Result and Discussions

The effect of various flow and roughness parameters on heat transfer characteristics for flow of air in rectangular ducts of different relative roughness height in the present investigation are discussed below. Results have also been compared with those of smooth ducts under similar flow and geometrical conditions to see the enhancement in heat transfer coefficient.

Figure 1 shows the values of Nusselt Number increases with increases in Reynolds Numbers because 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 turn increase the Nusselt Number.

Figure 2 shows the plots of experimental values of the friction factor as the function of Reynolds number for smooth plate and rough surface. It is clear that Value of friction factor drop proportionally as the Reynolds number increases due to the suppression of viscous sub-layer with increase in Reynolds number.

Figure 3 shows as Reynold No. increases Thermo hydraulic performance
also increases and it is max. for v groove plate and minimum for smooth plate.

VI. CONCLUSION

The present work was undertaken of with the objectives of extensive investigation into v shaped ribs as artificial roughness on the broad wall of solar air heater Results have been compared with those of a smooth duct under similar flow condition to determine heat transfer and friction factor

The following conclusion has been draw from this investigation

1) In the entire range of Reynolds number, it is found that the Nusselt Number increases, attains a maximum value for v groove roughened plate and increases with increasing roughness geometry.
2) On increasing the roughness on the plate the friction factor also increase.
3) The value of the friction factor reduces sharply at low Reynolds Number and then decrease very slightly in comparison to low Reynolds Number.
   The experimental values of the heat transfer of the v groove Roughness absorber plate has been compared with smooth plate. The plate having Roughness geometry v groove, gives the maximum heat transfer

CALCULATIONS FOR ROUGHENED DUCT

Sample calculations for l = 800 w/m^2, D_h = 0.04444 m, Re = 13211, Rough Plate No 1,
e/D_h = 0.0225

1. Average plate temperature:

T_{pav} = $(Tp_1 + Tp_2 + Tp_3 + p_4 + Tp_5 + Tp_6) / 6$
= (54.76 + 55.74 + 58.9 + 60.35 + 55.49 + 55) / 6
= 56.71^{o}C

2. Average Outlet Air Temperature:

Similarly the average air temperature is determined as :

T_{oav} = $(T_{01} + T_{02} + T_{03} + T_{04}) / 4$
= (44.475 + 42.531 + 42.045 + 45.206) / 4
= 43.536^{0}C.

3. Pressure difference:

ΔP_0 = Δh x 9.81 x 0.8
= 36 x 9.81 x13.76
= 2830 N/m^2

4. Mass flow rate:

m = C_d x A_0 x $[2 \rho \Delta P_0 / (1 - \beta^4)]^{0.5}$
= 0.62 x π / 4 (26.5 / 1000)^2 x [2 x 1.1 x 2830/ (1-0.5^4)]$^{0.5}$
= 0.0278kg/s

5. Velocity of Air:

V = m / ρWH
= 0.0278 / 1.1 x 0.2 x 0.025
= 5.054 m/s

6. Equivalent Diameter:

D_h = 4 x Area of Cross section / Perimeter
= 4 x .02 x 0.025 /2 (0.2 + 0.025)
= 0.04444m

7. Reynolds Number:

Re = V D / ν
= 5.054 x 0.04444 / 17 x 10^{-6}
= 13211

8. Heat gained by Air:

Qa = m C_p $(T_o – T_i)$
= 0.0278 x 1006 x (42 – 30.)
= 335.6 Watts

9. Convective Heat Transfer Coefficient:

h = $Q_a / A_p (T_{pav} - T_{fav})$
= 335.6 / 0.3 (66.– 36)
= 37.28 W / m^2 - ^{0}C

10. Nusselt Number:

Nu = h d / K
= 37.28 x 0.04444 / 0.0278
= 59.60

11. Thermo hydraulic performance

Thp = $(N_u / N_{us}) / (f_r/f_s)^{1/3}$
= 1.2

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