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Enhancement of heat transfer coefficient using Diamond shaped roughness on the absorber plate of solar air heater

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Abstract

Solar collector (Air heater) has low thermal efficiency due to the low thermal conductivity between Air and absorber plate. So there is a need to enhance the thermal conductivity between air and absorber plate. This leads higher temperature to absorber plate and hence maximum thermal losses occurs to environment .It can be made possible by creating artificial roughness on absorber plate .There are number of parameters which can enhance the thermal conductivity such as relative roughness pitch (P/e), Reynolds no. (Re) and angle of attack (α).Experimental investigation has been carried out to study heat transfer enhancement by using diamond shape as a rib on absorber plate. Electric heater is used to heat the plate in indoor experiment. Absorber plate heated with the Electric heater while three walls insulated with thermocol sheet. The relative roughness pitch (p/e) varies between 10-25 mm, the relative roughness height (e/D_h) = 0.023, Rib height (e) = 1mm, Duct aspect ratio (W/H) = 8, rate of air flow corresponds to Reynolds no.(Re) ranging from 3000-14000.Finally comparison of the heat transfer from both smooth and roughened plate under the same condition of air flow is made.

Keywords: Solar air heater; Diamond shape rib; Heat transfer enhancement

1. Introduction

Flat plate solar air heaters are used to heat air. Thermal conductivity of air can be enhanced by developing roughened surface on the absorber plate. Many investigators created artificial roughness in various forms to enhance the heat transfer coefficient. Creation of artificial roughness helps to produce turbulence near the wall to break the viscous sub layer which enhances the heat transfer between the absorber plate and air of solar heater. [4, 6, 8] created Artificial roughened absorber plates of solar air heater as transverse wedge shaped rib, 90 degree broken transverse ribs, W-shaped ribs and circular wire rib roughness respectively. [8, 9] used fine wires as a rib to create artificial roughness on surface of different shapes to enhance heat transfer coefficient. This also increased the frictional losses which thereafter resulted in consumption of more power to run the blower.

This paper aims to minimize friction losses .This friction losses create at a region very near to the duct surface. This region is known as a laminar sub layer. Laminar sub-layer breaks by the ribs. This rib creates local wall turbulence which develops flow separation also reattachment developed between consecutive ribs, Reduction of the thermal resistance and enhancement of the heat transfer caused by reattachment between consecutive ribs. The roughness on the metal sheet can be created by many methods, such as machining, forming, welding, sand blasting, casting and fixing ribs of small diameter wires.[1] developed a friction similarity law and a heat momentum transfer analogy for flow in rough tubes. [10] founds that Artificial roughness on the underside of the absorber plate enhance the heat transfer capability of a solar air heater. [2] 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 as roughness. It has been observed that increase in the relative roughness height results in a decrease of the rate of heat transfer enhancement and increase in the rate of friction factor. Rate of heat transfer and friction factor decreases if the relative roughness pitch increases. The Result of maximum enhancement in Nusselt number and friction factor occurs as 2.38 and 4.25 times than that of smooth duct, respectively. [3] Works on the effect of relative roughness height (e/D_h), angle of attack (α) and Reynolds number (Re) on heat transfer and friction factor in rectangular duct with circular wire as ribs on the absorber plate. Heat transfer coefficient can be improved by a factor up to 1.8 by using roughened plate in the duct and the friction factor can be increased by 2.7 times of smooth plate. The maximum heat transfer coefficient as well as friction factor can be found maximum at an angle of attack of 60 and 70, respectively,[9] developed geometry as artificially roughened by an expanded metal matrix for a rectangular duct on the heat transfer coefficient and friction factor in a large aspect ratio rectangular duct and observed 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.

2. Indoor experimental program

2.1 Experimental apparatus

An indoor experimental set up consist of long duct which is divided in to four parts. These are 177 mm (2.5 √WH) inlet section, 1500 mm (33.75 D_h) test section, mixing section and exit section 354mm (5 √WH) along with 87mm baffles spacing. A blower operates on three phase 240 Volts, control valve, orifice plate and other devices such as milivoltmeter, micro manometer to measure pressure head for Reynolds no. (ASHRE 1977) and inclined manometer for pressure measurement. A roughened absorber plate of length 1500 mm long placed on the top of the test section with several numbers of thermocouples as shown in fig-1.

Figure 1. Actual photograph of experimental setup

Exit section is provided after this section i.e. 345 mm in length. The objective behind providing the exit section was to reach the end effect in the last section. To determine uniform temperature at outlet, three baffles at 87 mm distance are placed to mix the hot air coming from the duct. An orifice plate is placed between the blower and exit section of the duct. A control valve is also provided beside the orifice plate to control the rate of air flow that varies the Reynolds number. The setup is covered with 25 mm thick thermocol sheet from inlet section of the duct to orifice plate to avoid heat losses .On one side of the G.I.1mm thick sheet diamond shaped roughness is created by pasting diamond shape and on the other side over which the thermocouples is fixed is painted by a black paint. The calibrated copper- constantan 0.3 mm (24 SWG) thermocouples were used to measure the temperature of air and the heated plate at different locations. The location of thermocouples on the heated wall is shown in Fig-2.

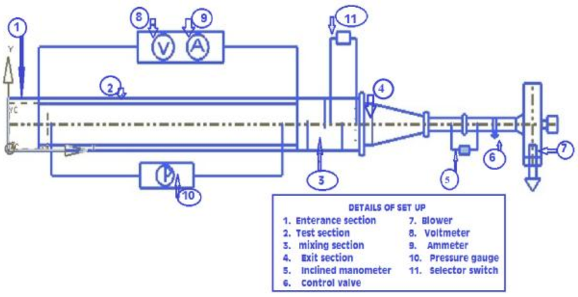


Fig. 2. Schematic line-diagram of experimental setup

This G.I. roughened sheet placed on duct at height of 25 mm from the duct base .Flat plate heater of size 1500 mm x 200 mm is placed on the G.I. plate to heat up the roughened plate. Voltmeter and Ammeter are also connected through wire with the heater. Calibrated copper–constantan thermocouples are affixed on the black side of the sheet for measuring temperature through a digital micro-voltmeter. The mass flow rate is measured with the help of inclined manometer across the orifice plate and can be varied with the help of control valve. [12], [13] reported the optimum value of p/e for the rectangular ducts to be10. [12], [13], [14] have reported an optimum rib angle of 45° to 60°. The length of the circular GI pipe provided was based on pipe diameter d1, which is a minimum of 10 d1 on the upstream side and 5 d1 on the downstream side of the orifice plate. In the present experimental set-up, we used 1000 mm (13 d1) pipe length on the upstream side and 700 mm (9 d1) on the downstream side.

2.2 Absorber plates

Fig -3 and fig-4 shows the geometry of roughness plate. Present investigation is taken up with the objective of experimentation on diamond shape ribs as an artificial roughness. This roughness is created with the help of diamond shape rib of mild steel with rib height (e) =1mm and pitch (p) =10mm to 25mm.

Figure 3. Figure of smooth and rough plates at different pitch

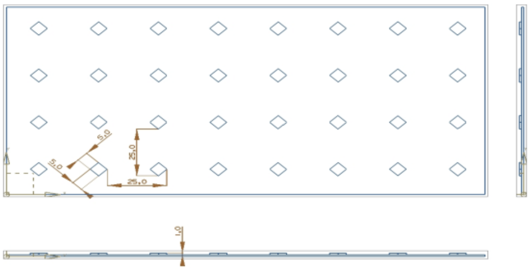


Fig.4. Pitch (p) =25 mm, Rib Height (e) =1mm

2.3 Experimental procedure

All components of the experimental setup and the instruments are well connected and placed for proper operation. The joints of the setup are checked for air leakage with soap bubble technique by switching the blower. Micro manometer is connected for pressure measurement which is used to calculate friction factor and inclined U-tube manometer is used to measure pressure difference across orifice plate .Blower is switched on and the flow control valve is adjusted to give a predetermined rate of airflow to the test section. The quasi-steady state condition is assumed to occur if the temperature at a point does not change for about 10-12 minutes. Every reading takes approximately 40-50 minutes to reach at quasi-steady state condition. Air flow rate is changed with the help of turning valve and then the system was allowed to attain a steady state before the data were recorded. The temperatures of air entering the duct, leaving the duct and plate reading at different thermocouples point in the form of millivolt-meter reading were recorded. All millivolt-meter readings are converted to temperatures. All readings have been noted under steady state condition which was assumed to have been obtained when the plate and air outlet temperature did not deviated over a 15 min. period. After the steady state has reached, the heater assembly voltage and current, the plate temperatures, the inlet and exit air temperatures and the pressure drop across the duct and across the orifice plate have been recorded. For each rib configuration 07 runs have been conducted at air-flow rates corresponding to the flow Reynolds numbers between 3000 and 14000.

The following parameters were measured during the experiments

1. Pressure drop across the orifice plate by inclined U- tube manometer
2. Pressure drop across the duct by using micro manometer.
3. Inlet air temperature of collectors by using digital milivoltmeter and thermocouples.
4. Outlet air temperature of collectors
5. Temperature of plate

Table -1 Experimental condition

Parameter	*Values*
Reynolds Number (Re)	3000 – 14000
Roughness height (e)	1mm
Relative roughness height (e/D_h)	0.023
Relative roughness pitch (p/e)	10 mm-25mm
Heat Flux (l)	900 W/m^2
Plate Material	G.I. Sheet
Thickness of Plate	1 mm
Channel Aspect ratio (W/H)	8
Test Length	1500 mm
Hydraulic Diameter	44.44 mm

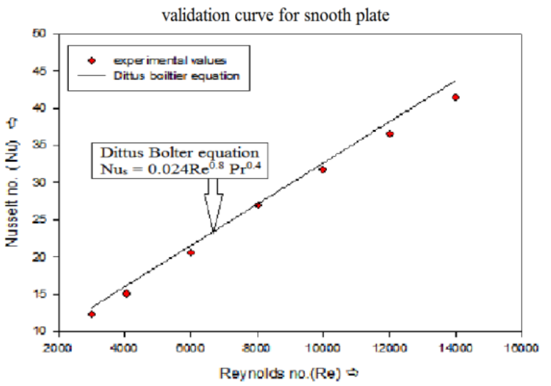
2.4 Validation test

The value of Nusselt number as obtained from experimental data is compared with the value of Dittus Boelter equation. The Dittus Boelter equation is given by:

$$Nu_s = 0.024Re^{0.8}\ Pr^{0.4}$$

S. No.	**Reynolds no. (Re)**	**Inlet temperature of air (t_i)°C**	**Average outlet temperature (t_{oav}) °C**	**Average plate temperature (t_{pav}) °C**	**Average air temperature (t_{fav}) °C**	**Heat transfer Q (Watt)**	**Convective heat transfer coffecient (h) W/m^2-K**	**Nusselt no. (Nu)**	**Thermal efficiency η_{th}**
1	3018	25.89	38.84	66.25	32.37	82.22	8.09	12.93	30.5
2	4070	25.89	36.05	61.18	30.97	86.96	9.60	15.34	32.2
3	6037	25.89	33.25	53.56	29.57	93.51	12.99	20.77	34.6
4	8038	25.89	31.68	48.23	28.79	97.88	16.78	26.82	36.3
5	10012	26.15	31.12	46.45	28.63	104.80	19.60	31.34	38.8
6	12040	26.15	30.46	44.17	28.30	109.32	22.97	36.72	40.5
7	14012	26.15	29.95	42.27	28.05	112.25	26.32	42.08	41.6

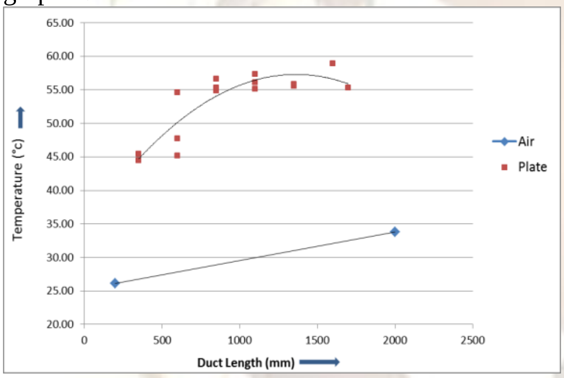
Table -2 Experimental data for smooth absorber plate



Graph -1

2.5 *Variation of temperature along test duct*

Experimental duct have four sections viz. air entering section, test section (heating section), mixing section and exit section. Air enters at room temperature in the entering section and absorbs heat in the test section (heating section).Complete air of the duct mixes in the mixing section and finally exit from the exit section. Temperatures of all the four sections are noted down and are plotted as shown in graph 2.



Graph-2

Graph-2 shows the typical variations of plate and air temperatures along the length of roughened test duct.

3. 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 measured values of air temperatures 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 all the temperatures reading at all points located on the absorber plate.

3.3 Pressure Drop Calculation

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

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

Where

ΔP_o = Pressure difference

$\Delta\rho_m$ = Density of the fluid (Mercury) i.e. 13.6×10^3

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:

S. No.	Reynolds no. (Re)	Inlet temperature of air (t_i)°C	Average outlet temperature (t_{oav}) °C	Average plate temperature (t_{pav}) °C	Average air temperature (t_{fav}) °C	Heat transfer Q (Watt)	Convective heat transfer coffecient (h) W/m^2-K	Nusselt no. (Nu)	Thermal efficiency η_{th}
Roughness height (e) =1mm, relative roughness height (e/D_h) =0.0225, relative roughness pitch (P/e) =10 and Heat flux I=900 w/m^2									
1	3018	25.89	44.42	35.16	82.5	117.69	8.29	13.64	43.6
2	4070	25.89	42.39	34.14	79.96	141.31	10.28	16.92	52.3
3	6037	25.89	39.09	32.49	73.36	167.67	13.68	22.51	62.1
4	8038	25.89	36.55	31.22	65.49	180.31	17.54	28.87	66.8
5	10012	26.15	35.03	30.59	60.42	187.15	20.92	34.42	69.3
6	12040	26.15	34.27	30.21	58.13	205.78	24.56	40.43	76.2
7	14012	26.15	33.25	29.7	55.08	209.54	27.52	45.29	77.6
Roughness height (e) =1mm, relative roughness height (e/D_h) =0.0225, relative roughness pitch (P/e) =15 and Heat flux I=900 w/m^2									
1	3018	25.89	46.96	76.66	36.43	133.82	11.09	18.25	49.6
2	4070	25.89	43.23	72.35	34.56	148.48	13.10	21.56	55.0
3	6037	25.89	38.58	63.97	32.24	161.22	16.94	27.88	59.7
4	8038	25.89	37.57	62.95	31.73	197.49	21.08	34.70	73.1
5	10012	26.15	36.05	58.64	31.10	208.54	25.24	41.54	77.2
6	12040	26.15	34.90	55.59	30.53	221.85	29.50	48.56	82.2
7	14012	26.15	33.76	52.55	29.95	224.51	33.12	54.52	83.2
Roughness height (e) =1mm, relative roughness height (e/D_h) =0.0225, relative roughness pitch (P/e) =20 and Heat flux I=900 w/m^2									
1	3018	25.89	46.20	36.05	78.44	128.98	10.14	16.69	47.8
2	4070	25.89	43.79	34.84	77.17	153.26	12.07	19.86	56.8
3	6037	25.89	40.11	33.00	71.33	180.57	15.70	25.85	66.9
4	8038	25.89	37.06	31.48	63.21	188.90	19.84	32.66	70.0
5	10012	26.15	35.67	30.91	59.15	200.52	23.67	38.96	74.3
6	12040	26.15	34.65	30.40	56.35	215.42	27.67	45.54	79.8
7	14012	26.15	33.63	29.89	53.05	220.76	31.77	52.29	81.8
Roughness height (e) =1mm, relative roughness height (e/D_h) =0.0225, relative roughness pitch (P/e) =25 and Heat flux I=900 w/m^2									
1	3019	25.89	45.57	80.47	35.73	124.95	9.31	15.32	46.3
2	4070	25.89	43.03	78.18	34.46	146.74	11.19	18.41	54.3
3	6037	25.89	39.55	72.09	32.72	173.48	14.69	24.17	64.3
4	8038	25.89	36.68	64.48	31.29	182.46	18.32	30.16	67.6
5	10012	26.15	35.28	59.65	30.72	192.5	22.17	36.5	71.3
6	12040	26.15	34.47	57.37	30.31	210.92	25.98	42.76	78.1
7	14012	26.15	33.51	54.58	29.83	217.02	29.23	48.11	80.4

Table -3 Experimental data for roughened absorber ate

$R_e = VD / \nu$

Where,

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

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

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.

3.9 Thermal Efficiency

The Thermal efficiency for test section is calculated from:

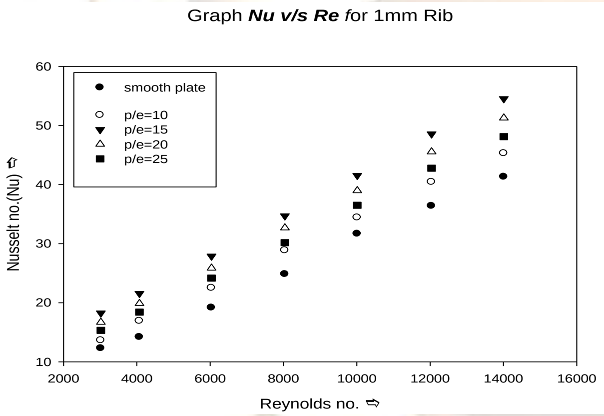
Thermal efficiency (η) = Q_a / A_p I

Where,

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

4. Results and discussion

A comparison is made of Heat transfer coefficient for smooth plate and roughened plate under similar fluid flow condition. Roughness was created to observe the enhancement in heat transfer coefficient. Fig.3 shows the geometry and arrangement of roughened plate. Experimental data for roughened absorber plate Table-3 shows that corresponding to Roughness height (e) =1mm, relative roughness height (e/D_h) =0.0225, relative roughness pitch (P/e) =15 and Heat flux I=900 w/m^2, Nusselt number is found to be increasing with increase in Reynolds number. The Nusselt number attains a maximum value of 54.52 at Reynolds number 14012 and the corresponding thermal efficiency is found to be 81.8 percent. A comparison of Nusselt v/s Reynold number for different p/e values is shown in Graph-3. Also GRAPH indicates that heat transfer coefficient is maximum for 15 mm pitch roughened plate.



Graph -3

5. Conclusion

The present work is undertaken with the objective of experimental investigation into diamond shape roughness created artificially on the absorber plate of solar air heater. Results obtained have been compared with those of smooth duct under similar flow conditions to determine the enhancement in heat transfer coefficient.

The following conclusion have been drawn from this investigation

1. In the entire range of Reynolds number, it is found that the Nusselt number attains a maximum for roughness pitch of 15 mm.
2. The value of Nusselt number increases sharply at low Reynolds number and become constant or increases very slightly in comparison to low Reynolds Number.
3. The maximum enhancement of heat transfer coefficient occurs at pitch of 15 mm while on either side of this pitch the Nusselt number decreases.
4. It has also been observed that at low Reynolds number (about 3000) a smooth plate and an artificial roughened plate have very less variation of heat transfer .
5. The experimental values of the thermal efficiency of the roughened absorber plate tested have been compared with that of the smooth plate and it is found that a plate having roughness Pitch 15 mm gives the highest efficiency of 83.2%.i.e 2.01 times that of the smooth plate.

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