A Review Paper on Experimental Investigation on Heat Transfer and Fluid Flow Phenomena of Arc Shaped RIB with Gaps on The Plate

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Abstract- A simple solar air heater has a low value of heat transfer coefficient. It is because of low interaction between absorber plate and the flowing air. In the turbulent flow near the absorber plate surface, a laminar sub layer forms, that is less efficient for the heat transfer and hence act as an insulating medium. So due to this reason we create artificial roughness on the underside of the absorber plate to break the laminar sub layer. Artificial roughness disturbs the laminar sub layer and makes it turbulent, which results in increase in the heat transfer rate. Thermo-hydraulic performance of solar air heater duct can be improved through enhancing the heat transfer. Hence arc shaped artificial roughness is an effective technique to enhance the value of heat transfer of fluid flow

Keyword - Solar Air Heater, Thermal Conductivity, Heat Transfer, Artificial Roughness, Thermo-hydraulic performance.

1. INTRODUCTION

Solar energy is considered as the light and Radiant Heat which is coming from the sun and impacts Earth's atmosphere, climate and supports life. Solar energy is the huge source of energy which is freely available in plenty and it does not cause any effect on environment. Sun based advancements are extensively delineate as passive solar or active solar filling up on the method they catch, change over and publicize solar energy.

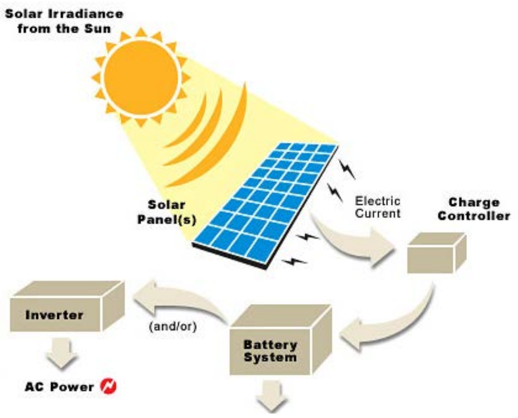


Fig 01Solar energy

Active sunlight based system uses sun's radiation which further converts to electricity. Passive light system based on heat coming from the sun used to produce electricity.

Fig .1Solar energy now been used in many applications such as in the industrial and domestic applications. Different techniques used to utilise solar energy are solar cooking, solar lighting, solar cooling, space Technology etc. Fossil fuels are important part of solar energy which is stored in the form of organic matter. Most importantly fossil fuels affects environment badly and causes global warming acid rain smog pollution etc. as we know fossil fuels are decreasing day by day rapidly. That's why we have to select the alternate source of energy which can replace fossil fuels such as solar energy wind energy geothermal energy etc.

As the development in the field of solar energy technology is rapid and limited fossil fuels, solar energy become the most affordable and cheapest source of energy. In solar energy the initial cost of installation is very high but running and maintenance cost is very low.

2. LITERATURE SURVEY

Yadav et al experimentally investigated the effect on heat transfer coefficient and friction factor characteristics due to presence of artificially roughened Circular rotation arrangement. The experiment encompassed Reynolds number 3600 to 18,100 relative roughness pitch (p/e) of range 12 to 24, angle of attack(α) of range 45^0 to 75^0,and aspect ratio(w/H) as 8,Relative roughness height(e/Dh) of range 0.015 to .03. The maximum heat transfer coefficient and friction factor increased by 2.89 and 2.93 times respectively as compared with smooth duct.

D. Guptaet al investigated the performance of solar air heater using the oblique wire shaped artificial roughness. They found that the effective efficiency of solar air heater is obtained at low air flow rate. And as the height of roughness increases the optimum value of air flow rate get decreased. They also resulted that the higher effective efficiency obtain as the insolation increases for Reynolds number above 10,000 and with decrease in insolation for Reynolds number below 10,000.

Aharwal et alperformed an experiment to investigate the heat transfer and friction characteristics of rectangular duct. The experiment encompassed relative roughness pitch (p/e) of 10,, angle of attack(α) of 60^0,Relative roughness height(e/D_h) of range 0.0377.They found that the Nusselt number and friction factor increase by 2.59 times and 2.87 times respectively as compared with smooth duct.

Sachin Baraskar et.al Artificial roughness in the form of repeated ribs is generally used for enhancement of heat transfer from heated surface to the working fluid. This paper presents the experimental investigation of heat transfer and friction factor characteristics of a rectangular duct roughened with repeated v-shape ribs with and without gap on one broad wall arranged at an inclination of 60° with respect to the flow direction. A rectangular duct of aspect ratio of (W/H) of 8, relative roughness pitch (p/e) of 10, relative roughness height (e/Dh) of 0.030, and angle of attack 600. The heat transfer and friction characteristics of this roughened duct have been compared with those of the smooth duct under similar flow condition. The effect of gap in v-shaped rib has been investigation for the range of flow Reynolds numbers from 5000 to 14000. The maximum enhancement in Nusselt number and friction factor is observed to be 2.57 and 2.85 time of that of the smooth duct.

Singh s. Chander et alinvestigated on discrete v-down rib roughened absorber plate to examine the thermos-hydraulic performance and efficiency of solar air heater. They found that the effectively increased in efficiency by 91% as compared to smooth plate solar air heater.

R.P. Saini et alperformed an experiment to study the effect of Dimple shape rib roughness on the performance of solar air heater. The experiment encompassed Reynolds number ranges from 2000 to 12000 relative roughness pitch (p/e) of range 8 to 12,Relative roughness height (e/Dh) vary from 0.018 to .037.They found that the maximum muscle number achieved at relative roughness height (e/D_h) of 0.037 and relative pitch of 10. While the minimum value of friction factor is achieved at relative roughness height of 0.0289 and relative pitch of 10.

Anil Kumar et al [19] experimental investigated the effect of geometrical parameters of multi v-shaped ribs with gap on heat transfer and fluid flow characteristics. the experiment encompassed Reynolds number from 2000 to 20,000 relative roughness pitch (p/e) of 10, angle of attack(α) of 60^0 ,and aspect ratio(w/H) as 6, relative gap distance of 0.24 to 0.80 Relative roughness height(e/Dh) is 0.043,aspect ratio(w/H) vary from 0 to 10. They resulted that the Nusselt number and friction factor is increased by 6.32 times and 6.12 times respectively as compared with smooth duct.

Patil et al [20,21,22] studied the effect of discrete v-rib with combined staggered rib pieces on heat transfer and friction factor characteristics on the rectangular duct. the experiment encompassed Reynolds number from 3000 to 17,000 relative roughness pitch (p/e) of 10,staggered rib position of 0.2 to 0.8, angle of attack(α) of 60^0,and aspect ratio(w/H) as 6, relative gap distance of 0.2 to 0.8, relative staggered rib size of 1 to 2.5 . They resulted that the maximum value of Nusselt number and friction factor corresponds to relative gap position of 0.6, relative staggered rib size of 2.5 and staggered rib position of 0.6.

3.EXPERIMENTALSETUP DETAILS

The experimental setup involves a blower, control valve, a Test duct having inlet and outlet section for air flow and various temperature and pressure measuring devices. The experimental setup is shown in figure below. Blower is used to suck the atmospheric air through the rectangular duct which is having artificial roughness. Artificial roughness is created by fixing Arc-shaped ribs having attack angle of 30^0 and varying value of rib gap(g) from 1,2, 3,4,5 roughness on the bottom of the top plate. The mass flow rate of air passes through the duct controlled by control valve on the line. The rectangular duct is made up of wood having dimensions of 2042 mm in length and cross sectional area of 200 mm x 20 mm. it has a test section of length 1500 mm, entrance section of length 192 mm and an exit section of length 330 mm. Some of the other data are as follows-

1.AirInlet Section 2.Test Section 3.Air Outlet4.Varice 5.Selector Switch 6. Mixing Section 7. G.I Pipe 8. Orifice Plate 9.Inclined U-Tube 10. Micro Manometer 11 Flow Control Valve 12.Flexible Pipe 13.Blower

According to ASHRE standard 93-77(1950) the entry length is $5(H)^{3/2}$, and exit length is 2.5(W.H)½ [25]. The experimental setup is designed according to this data. At the exit section three baffle plates are placed with an equal distance of 75mm Length. For proper mixing of air baffles are employed. At the exit of rectangular duct a plenum is installed to connect the circular pipe with rectangular duct. A HR plate having dimension 1500 mm x 200 mm x 1.6 mm is used as absorber plate which is having artificial roughness of arc shape at the test section of duct. The copper wires are stick on the underside of the absorber plate work as artificial roughness. The wire diameter and the angle of attack are remains fixed while the number of rib gaps varies to obtain very shape of roughness. To heat the plate uniform heat flux is supplied by electric heater assembly. The electric heater size is 1500 mm x 200 mm was assembled by combining loops of parallel and series of heat resistance wire installed on asbestos sheet of 5 mm.

thermal insulating material such as polystyrene is used on the backside of electric heater to reduce the thermal energy losses. The gap between absorber plate and electric heater is taken as 20 cm. The energy input to the heater is controlled by the variac such that we can obtain desired amount of heat flux. The centrifugal blower, which is driven by three phase 5HP, 2880 rpm blower used to suck the atmospheric air through the rectangular duct. The ambient air sucked by the blower using pipeline through the rectangular duct delivered to the atmospheric air. Air flow rate is control by control valve at the desired level. The mass flow rate of air is controlled by orifice metre through the rectangular duct.

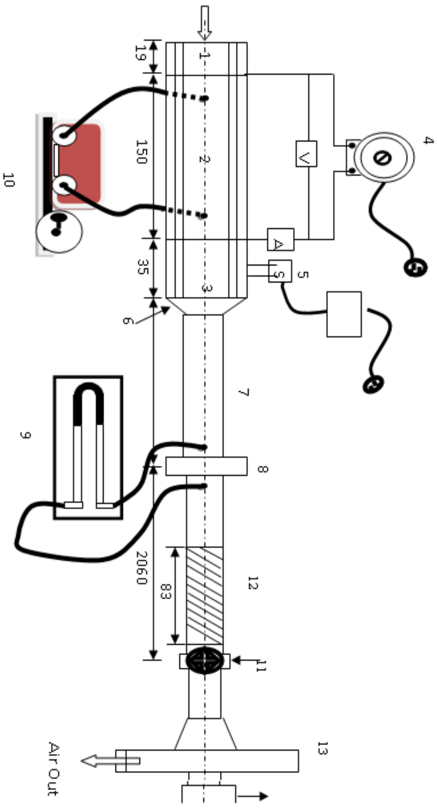


Fig 02Schematic Diagram Showing Top View of Experimental Setup

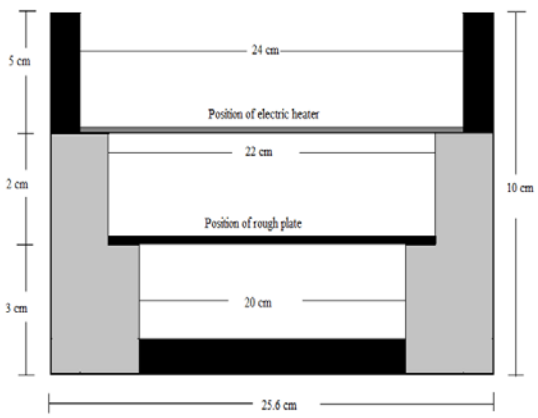


Fig03Schematic diagram of experimental setup

HEATER-Since we have indoor type explain experimental set up. It is become important to remain on our goal to provide the similar atmosphere to sun primary based radiation on the absorbing plate by an electric heater dimension of 1650 mm x 216 mm which was created by applying heating wire in series and parallel loop type arrangement on 5 mm asbestos sheet acting as an insulating sheet to prevent heat transfer losses 1 mm thick sheet of mica is placed in between the absorbing plate and heater which act as a casing. Where is used to vary heat flux from 0 to 1000 W/m^2. The top portion of the channel is completely secured with 1.2 mm thick ply board which insulate the whole get along.

EXPERIMENTAL DATAS

In this thesis the data is collected experimentally using various instruments for different roughness gap of absorber plate. The flow rate of air changed by control valve and the data collected for rough plate is compared with smooth plate. Different air flow characteristics such as Nusselt number, heat transfer coefficient, friction factor and thermal efficiency is determined by using experimental data.

EXPERIMENTAL PARAMETERS

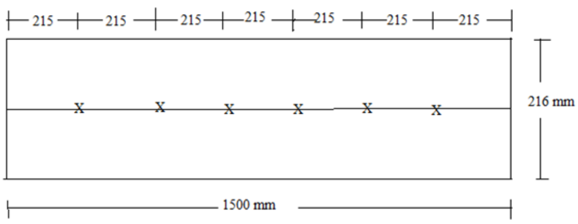
PARAMETERS	VALUE
Roughness height (e)	2 mm
Relative roughness height (e/D_h)	0.0450
Relative gap width (g/e)	4
Number of gaps (N_g)	1,2,3,4,5
Reynolds number (Re)	2000-12000
Angle of attack of flow (α)	30^0
Equivalent diameter of air passage or hydraulic diameter (D_h)=4WH/[2(W+H)]	0.044 m
Relative roughness pitch (P/e)	10
Material of the absorbing plate	HR sheet
Aspect ratio of duct (W/H)	8
Testing length	1500mm
Heat flux (I)	1000

EXPERIMENTAL PROCEDURE

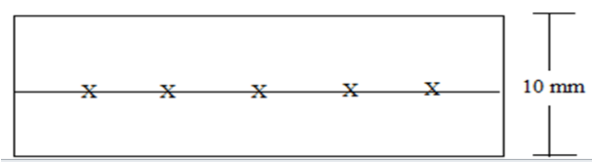
For correct operation the components and instruments are joined suitably with the experimental setup. The blower is then started. In GI pipe all the leakage proofing done on the joints to remove the errors. Flow control valve helps in controlling the mass flow rate of air for the testing section. It is important to control the air flow through the control valve to adjust the level of U tube pressure gauge and the micrometre. Experiment is performed under the Quasi-steady state condition to collect the data related to the heat transfer Coefficient and friction factor.

After every adjustment in mass flow rate of air with the help of control valve, the data was recorded only when a stable state is achieved. as we change the operating condition at any point the system takes approximately 45 to 50 minutes to attend the quasi-steady state .To achieve accuracy in result the temperature of air stream flowing through the air duct were kept more than 100c on the grounds that the accuracy in determining the heat transfer coefficient is influenced by the temperature estimation. The temperature distinction has been kept more than 200c between the air temperature and absorber plate.

The inlet of air flowing through the channel is kept limited between 300to 400c while testing in ambient condition. The outer temperature may vary between 500c to 800c. The plate temperature has taken just to achieve the steady state condition, if the plate temperature and outlet temperature of air didn't change quite about 15 minutes. As the quasi-static state condition achieved, we measure the plate temperature, outlet and inlet temperature of air with the help of thermocouple while voltage and current of heater is measured by using voltmeter and digital ammeter respectively, the pressure drop over the operating plate And pressure drop over the pipe is measured with the help of U-Tube pressure manometer. Air flows with six various rates so the Reynolds number is carried for each of roughness height. The range of Reynolds number is taken from 2000 to 12000. By using data taker different parameters are measured in the experiment such as plate temperature, outlet temperature and inlet air temperature. Pressure drop over the pipe measured by pressure gauge and pressure drop over the whole plate is measured by U tube manometer.



X shows position of thermocouple wire on the absorber plate



P/e= 10, Ng=1g/e=4,α= 30

P/e= 10, Ng=2 g/e=4,α= 30

Smooth plate

4.SAMPLE CALCULATIONS

Sample calculation for D_h= 44.44mm.

Reynolds number- 2000-14000 and relative roughness $e/D_{h=}$ 0.045

1. Mean bulk air temperature (T_{tav}):-

Simple arithmetic mean of measured inlet ant exit value of air under testing section

$(T_{tav}) = (T_i + T_o)/2$

Where

Ti= Inlet temperature of air in ^{0}c

To= Outlet temperature of air in ^{0}c

2. Mean plate temperature (T_{pav}):-

Thermocouples wire are arranged at equidistance on the entire plate therefore it is the average reading of six points located at the distance of 187mm on the plate

$$(T_{pav}) = (T_{p7} + T_{p8+} T_{p9+} T_{p10+} T_{p11+} T_{p12})/6$$

3. Pressure drop across the orifice plate (ΔP_a):-

$$(\Delta P_a) = (\rho \times g \times \Delta h)$$

Where,

Δh = Difference of mercury level in U-tube manometer

ρ = Density of water

4. Mass flow rate measurement (m):-

$$m = C_d A_o[2\rho(\Delta Po)/1-\beta^4]^{0.5}$$

Where,

$\beta = d_2/d_1$

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

A_o= Area of orifice plate, m^3

ρ = Density of air, kg/m^3

5. Velocity of air (V):-

V = m/(ρWH)

Where,

m = Mass flow rate of air, kg/s

H = Height of rectangular duct, m

W = Width of the rectangular duct, m

6. Reynolds number (R_e):

$R_e = VD_h/\nu$

Where,

V = Velocity of air, m/s

Dh = Hydraulic diameter = 4WH/2(W+H)

ν = Kinematics viscosity of air at tfav in m2/sec

7. Heat transfer rate (Q_a):-

$$Q_a = mC_p (T_o - T_i)$$

Where,

m = Mass flow rate, kg/s

Cp = Specific heat of air at constant pressure (kJ/kg K)

Ti = Inlet temperature of air in o C

To = Outlet temperature of air in ^{o}C

8. Heat Transfer Coefficient (h):-

$h = Q_a / A_p (T_{pav} - T_{fav})$

Where,

A_p= Area of absorber plate (m^2)

9. Nusslet Number (Nu_r):-

$Nu_r = hD_h/ k$

Where,

h = heat transfer coefficient

D_h = Hydraulic diameter, m

k = Thermal conductivity (W/mK)

10. Thermal Efficiency (η_{th}):-

$\eta_{th}= Q_a/A_p I$

Where,

Q_a = Heat transfer rate

A_p = Area of absorber plate (m^2)

I = Heat flux i.e. 950 W/m^2

11. Friction Factor (f):-

$f = 2(\Delta P_d)D_h/ 4\rho L_f V^2$

Where,

ΔP_d= Pressure drop, N/m^2

L_f = Length of test-section for pressure drop measurement, mm

L_f = 1360mm

12. Thermal Hydraulic Performance (THP):-

$Nu_s= 0.023 \times (Re)^{0.8} \times (Pr)^{0.4} \times (2R_{av} / D_e)^{-0.2}$Where,

$2R_{av} / D_e$ = (1.156+H/W-1)/(H/W)

$f_s= 0.085 \times (Re)^{-0.25}$

$THP=(Nu_r/Nu_s)/(f_t/f_s)^{0.33}$

Experimental observation table for smooth plate Manometer Reading						
h (mm)	15	27	54	108	168	231
Micrometer Reading						
ΔP (Pa)	1.1	1.5	2.5	3.3	4.5	6.1
Inlet Temperature (^{0}c)						
T1	32.5	34.1	35.5	37.2	38.3	38.8
Outlet Temperature (^{0}c)						
T2	37.61	40.4	41.11	41.16	41.4	41.27
T3	37.78	40.65	41.55	41.64	42.05	41.87
T4	37.84	40.76	41.73	41.83	42.24	42.05
T5	36.03	39.48	40.71	40.86	41.25	41.14
T6	37.03	39.48	40.56	40.72	41.07	41.03
Plate Temperature (^{0}c)						
T7	58.1	58.3	58.1	57.2	57.1	56.28
T8	57.8	57.6	57.5	55.38	55.32	53.2
T9	56.8	56.3	56.1	55.78	55.2	54.8
T10	56.2	56.1	56	54.3	54.8	53.8
T11	55.3	55.1	54.9	52.7	51.8	50.3
T12	55.2	54.9	54.7	51.54	50.60	50.40

Experimental observation table for N_g= 1 Manometer Reading						
h (mm)	15	27	54	108	168	231

Micrometer Reading						
ΔP (Pa)	2.7	4.5	8.9	16.4	24.6	33.2
Inlet Temperature (0c)						
T1	30.15	31.37	33.1	35.1	36.55	37.85
Outlet Temperature (0c)						
T2	37.61	40.4	41.11	41.16	41.4	41.27
T3	37.78	40.65	41.55	41.64	42.05	41.87
T4	37.84	40.76	41.73	41.83	42.24	42.05
T5	36.03	39.48	40.71	40.86	41.25	41.14
T6	37.03	39.48	40.56	40.72	41.07	41.03
Plate Temperature (0c)						
T7	47.92	49.96	49.75	48.89	49.23	48.85
T8	50.01	52.48	51.97	50.57	50.48	47.72
T9	54.89	58.08	57.51	55.78	52.96	51.78
T10	53.03	55.69	54.86	52.96	52.42	50.34
T11	52.07	54.60	53.79	51.78	51.05	48.77
T12	51.29	54.07	53.61	51.54	50.60	47.3

Experimental observation table for N_g= 2 Manometer Reading						
h (mm)	15	27	54	108	168	231
Micrometer Reading						
ΔP (Pa)	2.4	4.2	8.1	15.1	22.4	29.6
Inlet Temperature (^{0}c)						
T1	36.9	38.3	39.9	40.3	41.1	42.5
Outlet Temperature (^{0}c)						
T2	44.1	47.2	47.6	46.8	46.3	45.8
T3	44.3	47.5	48.2	47.6	47.4	46.9
T4	44.4	47.6	48.5	48	47.8	47.4
T5	43.6	46.9	47.7	47.2	47	46.8
T6	42.8	45.9	46.7	46.2	45.8	45.7
Plate Temperature (^{0}c)						
T7	53.4	55.4	54.8	53.6	53.4	52.8
T8	55.8	58.5	57.6	55.8	55	52.1
T9	55.5	58.4	57.3	55.3	54.2	52.1
T10	58.1	61.2	60.1	57.8	56.7	54.5
T11	57.6	60.7	59.5	57	55.6	54.2
T12	61.8	65.6	64.8	62.3	60.6	58.5

5. RESULTS

1.Friction factor of smooth plate decreases with increase in Reynolds number.

2. Nusselt number of smooth plate increases with increase in Reynolds number.

3. Values of Nusselt number for roughened plate having number of gap (Ng= 1,2,) increases with increase in Reynolds number

4. Values of friction factor for roughened plate having number of gap (Ng= 1,2) decreases with increase in Reynolds number.

5. The maximum value of friction factor, nusselt number is and THP is 2.55 times, 2.33 times and 1.25 times respectively for number of gap (Ng)= 3.

6. CONCLUSION

1. With increase in Reynolds number the Nusselt number increases and friction factor decreases. Friction factor and Nusselt number values of rough plate are higher in comparison with smooth plate for similar conditions.

2. Maximum enhancement in Nusselt number occur at result table for number of gap (Ng)= 3 which is shown in figure-6.18 among all (Ng=1,2,3,4,5 and smooth plate).

3. Maximum enhancement in THP occur at result table for number of gap (Ng)= 3 which is shown in figure-6.20 among all (Ng=1,2, and smooth plate).

REFERENCES

[1] S. Singh, S. Chander, J.S. Saini Heat transfer and friction factor correlations of solar air heater ducts artificially roughened with discrete V-down ribs.

[2] Sachin Baraskar, K.R.Aharwal, A.Lanjewar / International Journal of Engineering Research and Applications (IJERA) ISSN: 2248-9622 www.ijera.com Vol. 2, Issue 6, pp.1024-1031

[3] Saini RP, Verma J. Heat transfer and friction factor correlations for a duct having dimple-shaped artificial roughness for solar air heaters. Energy 2008;133:1277–87.

[4] V.S. Hans, R.P. Saini, J.S. Saini Heat transfer and friction factor correlations for a solar air heater duct roughened artificially with multiple V-ribs

[5] A. Kumar, Saini R P, J.S. Saini Development of correlations for Nusselt number and friction factor for solar air heater with roughened duct having multi v-shaped with gap rib as artificial roughness Renew Energy, 58 (2013), pp. 151-163

[6] Patil A K, Saini J S, Kumar K. Nusselt number and friction factor correlations for solar air heater duct with broken V-

down ribs combined with staggered rib roughness. J Renew SustainEnergy2012;4:033122.

[7] Patil A K, Saini J S, Kumar K. Heat transfer and friction characteristics of solar air heater duct roughened by broken V-shape ribs combined with staggered rib pieces. J Renew Sustain Energy 2011;4(1):013115–7.

[8] Patil A K, Saini J S and Kumar K. Effect of gap position in broken V-rib roughness combined with staggered rib on thermos-hydraulic performance of solar air heater. Green1 (4): 329–8.

[9] Layek A, Saini J S, Solanki S C. Heat transfer coefficient and friction characteristics of rectangular solar air heater duct usingrib-grooved artificial roughness. Int J Heat Mass Transf 2007; 50:4845–54.

[10] Jaurker AR, Saini J S, Gandhi B K. Heat transfer and friction characteristics of rectangular solar air heater duct using rib-grooved artificial roughness. Sol Energy 2006;80:895–907.

[11] ASHRAE standard, 93-77, "Methods of Testing to determine the thermal performance of solar collectors", 1950.

[12] Patil A K, Saini J S and Kumar K. Effect of gap position in broken V-rib roughness combined with staggered rib on thermos-hydraulic performance of solar air heater. Green1 (4): 329–8.

[13] Layek A, Saini J S, Solanki S C. Heat transfer coefficient and friction characteristics of rectangular solar air heater duct usingrib-grooved artificial roughness. Int J Heat Mass Transf 2007; 50:4845–54.