

## **EXERGY ANALYSIS OF FLAT PLATE COLLECTOR BY USING ARTIFICIAL ROUGHNESS**

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### **ABSTRACT**

This paper presents an experimental exergy analysis for a flat plate solar air heater (SAH) with shot peened artificial roughness and without obstacles (smooth plate). For increasing the available heat-transfer area may be achieved if air is flowing over roughened absorbing plate, instead of smooth plate, leading to improved collector efficiency. The measured parameters were the inlet and outlet temperatures, the absorbing plate temperatures, the ambient temperature.

Further, the measurements were performed at different values of mass flow rate of air. The results show that the largest irreversibility is occurring at the flat plate (without obstacles) collector in which collector efficiency is smallest.

**Keywords:** Solar air heater; Experimental Setup; Exergy and Thermal efficiency analysis; Single flow; Shot Peened Roughness;

### **1. INTRODUCTION**

The heat transfer between the absorber surface (heat transfer surface) of solar air heater and flowing air can be improved by either increasing the heat transfer surface area using extended and corrugated surfaces without enhancing heat transfer coefficient or by increasing heat transfer coefficient using the turbulence promoters in the form of artificial roughness on absorber surface. The artificial roughness on absorber surface may be created, either by roughening the surface randomly with sand blasting or by use of regular geometric roughness. It is well known that in a turbulent flow a laminar/viscous sub-layer exists in addition to the turbulent core. The artificial roughness on heat transfer surface breaks up the laminar boundary layer and turbulent flow and makes the flow turbulent adjacent to the wall. The use of artificial roughness or turbulence promoters on the heat transfer surface in the form of ribs or grooves is one of the well known effective passive techniques of heat transfer enhancement used in pipes or ducts. An experimental investigation has been carried out to study the forced convection heat transfer and friction factor characteristics in solar air heater ducts using roughness by shot peening applied on upper broad wall of rectangular ducts.

Experiments were performed to collect the data for the calculation of forced convective heat transfer, efficiency and friction factor of air in simulated solar air heater rectangular ducts and exergy is a suitable quantity for the optimization of solar air heaters having different design and roughness elements. Second law of thermodynamic analysis combined with standard design procedure in thermal system gives details of system operation. Also exergy analysis takes into account quality of heat transfer. Exergy analysis reveals thermodynamic faults including exergy loss by absorber plate temperature level is minimized and reasonably optimizes design of absorber and flow ducts are getable. In this work an experimental investigation on heat transfer coefficient and friction factor characteristic of duct by using roughness by shot peening on the absorber plate (1.5 x 0.3 m) has been compared with the result of smooth plate. It has been found that by shot peening roughness duct performs better results as compared to smooth duct. Experiment has been performing on Indoor set-up. In this experimental investigation, one smooth plate and one absorber plates having shot peened roughness were tested in duct section. Result find in 5 test runs of Reynolds number range from 2635, 3616, 4295, 6073, 8590. According to the roughness geometry, turbulence occurs in fluid flow and in this way heat transfer is increased.

Thermal efficiency of rough absorber plate was compared to smooth plate. It was seen that heat transfer and pressure loss increased depending on shape of roughness. As a result of artificial roughness on the wall, the heat transfer coefficient in such ducts can be improved by a factor 1.5 to 2.0 as compared to that of smooth walls. Similarly the friction factor has been found to increase of 1.5 to 2.7. The thermal efficiency has been increased by around 40% as compared to smooth plate. And the exergy loss of the system decrease depending on the increase of collector efficiency. In present investigation dimensionless exergy loss decreases with Reynolds numbers.

## 2. RANGE OF PRESENT INVESTIGATION

Type of Roughness	Shot Peening Method
Roughness area coverage	100% Rough
Shots used	3mm
Aspect ratio, W/H	10
Test length, L	1500 mm
Reynolds number, Re	2635, 3616, 4295, 6073, 8590
Solar flux, I	600,700,800,900 W/m <sup>2</sup>
Hydraulic diameter, D <sub>h</sub>	0.0545 m
Plate material	Aluminum Plate
Thickness of Plate	3 mm

### EXPERIMENTAL INVESTIGATION:

An experimental investigation has been planned to generate data on heat transfer coefficient and friction factor that can be utilized to increase the heat transfer and friction factor it is proposed to collect data on heat transfer coefficient and friction factor as a function of roughness parameter (relative roughness height) aspect ratio of duct and Reynolds number of flow experimental data have also been collected on smooth duct under similar geometrical and flow condition in order to have a direct comparison of the performance of the roughened duct with that of a conventional smooth rectangular duct flow with respect to heat transfer and fluid flow characteristics. The Detail of Experimental set-up, procedure, experimental results and the method used to develop suitable correlations are described in the following section.

## 3. EXPERIMENTAL SET-UP

The Experimental schematic Diagram set-up including the test section is shown in the fig. The flow system consists of an entry section, a Test Section, An Exit section, a flow meter and A Centrifugal Blower. The duct is of the size 2043 mm x 200mm x 25mm (dimension of the inner cross section) and it is constructed from wooden panel of 25mm thickness. In the exit section of 200 mm three equally spaced baffles are provided in a 100 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 1650mm X 150mm was fabricated by combining series and parallel loops of heating wire on 5mm asbestos sheet. Mica sheet of 1mm is placed between the electric heater and absorber plate. This mica sheet acts as an insulator between the electric heater and absorber plate (G.I. Sheet) the heat flux may vary from 0 to 1000 W/m<sup>2</sup> by a variance across it. The back of the heater is covered by a 50 mm glass wool layer and a 12 mm thick plate of wood to insulate the top of the heater assembly. The side of the entire set-up, from the inlet to the orifice plate is insulated with 25mm thick polystyrene foam having a thermal conductivity of 0.037 W/m k. The heated plate is a 1mm thick G.I. Sheet with integral rib-roughness formed on its rear side and this form the top broad wall of the duct, while the bottom wall is formed by 25 mm wood with insulation below it. The mass flow rate of the air is measured by means of a calibrated orifice meter connected with an inclined manometer, and the flow is controlled by the control valve provided in the lines. The orifice plate has been designed for the flow measurement in the pipe of inner Diameter of 53mm, the orifice plate is fitted between the flanges so aligned that it remains concentric with the pipe.



**OBSERVATION TABLE (SMOOTH PLATE)**

S/N	Heat Flux (w/m <sup>2</sup> ) (I)	U-tube Manometer (dh)mm	Micrometer no. of division	Inlet Temp. in (T <sub>i</sub> °C)	Outlet Temp. in (T <sub>o</sub> °C)	Average Plate Temp. (T <sub>pav</sub> °C)	Average Fluid Temp. (T <sub>fav</sub> °C)
1	600	10	8	34.5	44.66	71.1	39.58
2	600	20	15	34.5	43.62	68.22	39.06
3	600	30	21	34.5	42.65	65.34	38.57
4	600	60	40	34.5	41.32	60.54	37.91
5	600	120	64	34.5	39.34	56.22	36.92

**EXPERIMENTAL RESULTS**

S/N	Reynolds No.	Convective heat transfer (w/m <sup>2</sup> /°C)	Nusselt No.	Thermal eff (%)	Exergy (E <sub>x</sub> ) (Watt)	Dimensionless Exergy loss (E <sub>d</sub> )	Friction factor (F <sub>s</sub> )
1	2635	6.41	12.143	32.1	41.14	0.43	0.095
2	3616	8.5	16.20	36.54	39.2	0.37	0.07
3	4295	10.8	21.1	42.5	36.5	0.33	0.06
4	6073	12.8	26.2	48.51	35.4	0.28	0.034
5	8590	16.47	32.3	50.12	32.5	0.24	0.015

**OBSERVATION TABLE (SMOOTH PLATE)**

S/N	Heat Flux(w/m <sup>2</sup> ) (I)	U-tube Manometer (dh)mm	Micrometer no. of division	Inlet Temp. in (T <sub>i</sub> °C)	Outlet Temp. in (T <sub>o</sub> °C)	Average Plate Temp. (T <sub>pav</sub> °C)	Average Fluid Temp. (T <sub>fav</sub> °C)
1	900	10	8	39.5	57.62	72.00	48.56
2	900	20	15	39.5	54.8	71.00	47.15
3	900	30	21	39.5	53.20	69.5	46.35
4	900	60	40	39.5	50.54	67.3	45.02
5	900	120	64	39.5	47.59	65.5	43.54

**EXPERIMENTAL RESULTS**

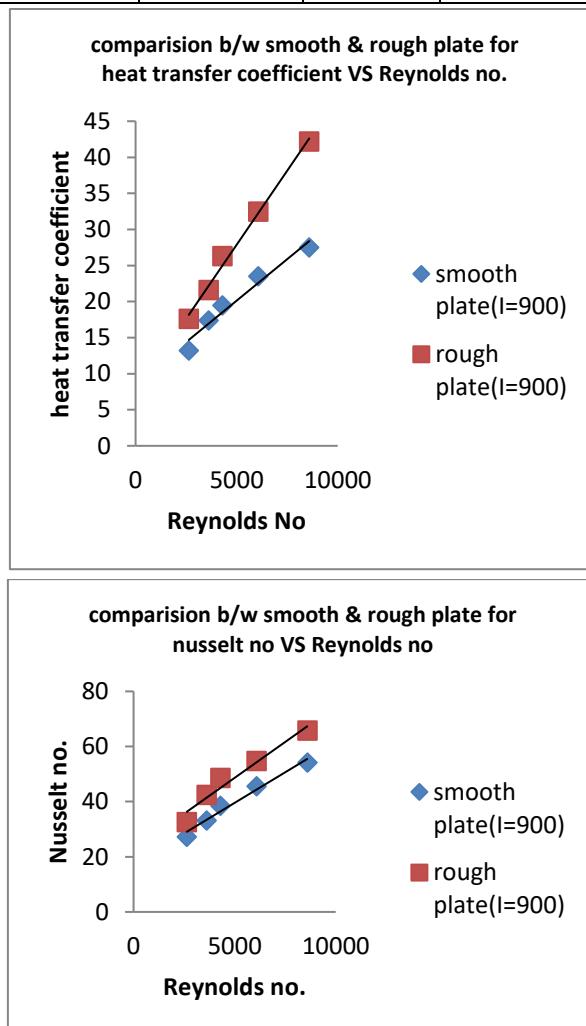
S/N	Reynolds No.	Convective heat transfer (w/m <sup>2</sup> /°C)	Nusselt No.	Thermal eff (%)	Exergy (E <sub>x</sub> ) (Watt)	Dimensionless Exergy loss (E <sub>d</sub> )	Friction factor (F <sub>s</sub> )
1	2635	13.2	27.25	38.8	55.69	0.31	0.095
2	3616	17.4	33.2	45.8	55.59	0.30	0.07
3	4295	19.5	38.5	51.2	52.62	0.27	0.06
4	6073	23.5	45.6	56.32	53.24	0.24	0.034
5	8590	27.52	54.2	60.5	51.62	0.20	0.015

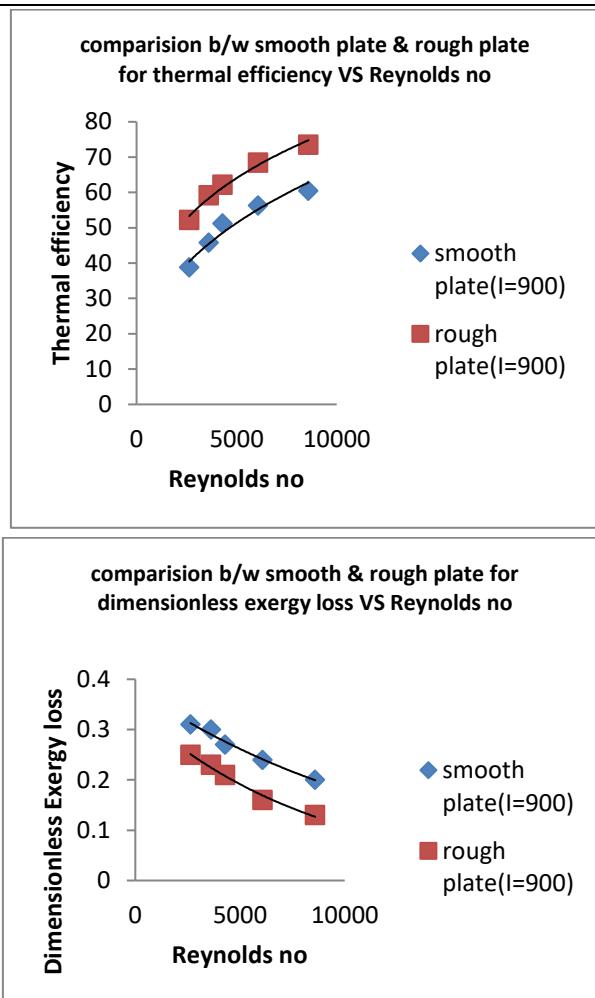
**OBSERVATION TABLE (ROUGH PLATE)**

S/N	Heat Flux( $w/m^2$ ) (I)	U-tube Manometer (dh)mm	Micrometer no. of division	Inlet Temp. in ( $T_i^0C$ )	Outlet Temp. in ( $T_o^0C$ )	Average Plate Temp. ( $T_{pav}^0C$ )	Average Fluid Temp. ( $T_{fav}^0C$ )
1	900	10	8	40.2	62.7	72.2	51.45
2	900	20	15	40.2	59.5	71.3	49.85
3	900	30	21	40.2	57.52	70.2	48.86
4	900	60	40	40.2	53.53	70.2	48.86
5	900	120	64	40.2	50.28	59.4	45.24

**EXPERIMENTAL RESULTS**

S/N	Reynolds No.	Convective heat transfer ( $w/m^2/0C$ )	Nusselt No.	Thermal eff. (%)	Exergy ( $E_x$ ) (Watt)	Dimensionless Exergy loss ( $E_d$ )	Friction factor ( $F_s$ )
1	2635	17.6	32.6	52.23	59.43	0.25	0.095
2	3616	21.6	42.5	59.22	60.51	0.23	0.07
3	4295	26.3	48.6	62.21	59.7	0.21	0.06
4	6073	32.5	54.8	68.51	55.17	0.16	0.034
5	8590	42.2	65.82	73.50	48.28	0.13	0.015





#### MEAN AIR & PLATE TEMPERATURE

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

#### PRESSURE DROP CALCULATION

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

#### MASS FLOW MEASUREMENT

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

#### VELOCITY MEASUREMENT

$$V = m / \rho W H$$

#### REYNOLDS NUMBER

$$Re = VD / \nu,$$

#### HYDRAULIC DIAMETER

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

#### HEAT TRANSFER COEFFICIENT

$$Q_a = m C_p (T_o - T_i)$$

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

#### NUSSELT NUMBER

$$Nu = hD_h / k$$

#### FRICTION FACTOR

$$f = (2 \times P_{micro} \times \rho \times D_h) / (4 \times \rho \times L \times V^2)$$

#### THERMAL EFFICIENCY

$$\eta = Q_a / I \cdot A_p$$

## EXERGY

$$E_x = mC_p \Delta T - mC_p T_a \ln (T_o/T_i) -$$

$$mRT_a \ln (P_o/P_i) + I.A (1 - T_a/T_p)$$

## DIMENSIONLESS EXERGY LOSS

$$E_D = E_x / Q_a$$

## 4. RESULT AND DISCUSSION

The effect of various flow and roughness parameters on thermal efficiency and exergy analysis for the flow of air in rectangular ducts of shot peening artificial roughness (100%) of the present investigation on indoor set-up are being discussed below. Results have been compared with those of smooth duct under similar flow and see the enhancement in heat transfer coefficient, friction factor and thermal efficiency. The aim of our analysis is to increase collector efficiency and decrease exergy losses with using passive method. The result has been compared with the Reynolds number vs. heat transfer coefficient, Nusselt number, friction factor, thermal efficiency and dimensionless exergy loss. Increasing the mass flow rate there occurs increase efficiency. However, the outlet temperature of air increases significantly changes with geometry of the absorber plate. As known the incident solar radiation is one of the most important parameter in solar collector efficiency. As seen from the result, the collector efficiency increased with increasing mass flow rate of the fluid. The radiation is maximum collector efficiency is also maximum. The dimensionless exergy loss of the system decreases depending on the increase of collector efficiency. There is reverse relationship between dimensionless exergy loss and heat transfer, as well as pressure loss. The most important parameters in order to decrease the exergy loss are the collector efficiency, temperature difference ( $T_o - T_i$ ) of the air and pressure loss. Graph shows the variation of thermal efficiency with respect to Reynolds number. As found by experiment that the thermal efficiency increases rapidly in lower Reynolds number region and in the region of high Reynolds number, rate of increase of thermal efficiency is lower. Thermal efficiency is higher (73.5%) at higher Reynolds No (8590) as per my investigation for rough plate and at same Reynolds No the efficiency of smooth is 60.5%. When the Reynolds number is low the roughness elements lie within the thermal boundary layer and velocity of air not sufficient to break the thermal boundary layer with retard flow rate. But with increase the mass flow rate so.

## 5. CONCLUSION

The conclusions can be drawn from the experimental study are:

1. Friction factor decreases with increases in Reynolds number.
2. Thermal efficiency increase with increase in Reynolds number.
3. Dimensionless exergy decreases with increases Reynolds number.
4. While comparing experimental value of Nusselt number with theoretical value, it is found that the experimental values are within a range of predicted value.
5. The solar air heater with roughened absorbers performs better as compare to smooth heater when intensity of solar radiations is high.
6. Thermal efficiency of roughened duct is increases as compare to smooth duct at same Reynolds number.
7. When the surface roughness is increased, the heat transfer and pressure loss increases.
8. The experimental value of the thermal efficiency of the shot peening roughness (100%) absorber plate tested have been compared with that smooth plate, gives high efficiency 73.5 % at high Reynolds number (8590).
9. The dimensionless exergy loss of the collector decreases depending on the increase of the collector efficiency.
10. Thermal efficiency decreases rapidly when the ambient temperature increases.

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