EXPERIMENTAL ANALYSIS OF SOLAR AIR HEATER DUCT ARTIFICIALLY ROUGHNED ABSORBER PLATE WITH CONTINUOUS DISCRETE RIBS

Deepak Kumar Dubey[1], Prof. Vinay Kumar Yadav[2]
Mtech.Scholar[1],HOD[2]
Department of Mechanical Engineering[1,2]
Aisect University. Bhopal, India

Abstract — In a analytical methodology a trails have been conducted on the heat transfer and friction factor for a solar air heater duct with continuously discrete rib of circular cross-sectional ribs without symmetric gaps. In the experimental investigation, it was assured that the height is 2mm, the pitch (p) of the ribs is 20, the relative roughness pitch (p/e) is 10mm. This investigation was carried out on a Reynolds Number (Re) ranging within 4000-18000. When the observations were compared between smooth absorber plates.
The smooth ribs couldn't transfer the desired heat and so it was not practically prepared.
The rough ribs having symmetrical gaps were efficient enough to transfer heat but it is not economical and has a huge complexity in design, whereas the continuous discrete ribs without symmetric gaps overcome this problem.
The result of continuous discrete ribs was concluded by comparing Nusselt number and Friction factor with circular ribs having symmetric gaps.

Keywords— Solar air heater, Nusselt number, Friction Factor, Continuous discrete ribs

I. INTRODUCTION

Solar air heaters form the major component of solar energy utilization system which absorbs the incoming solar radiation, converting it into thermal energy at the absorbing surface, and transferring the energy to a fluid flowing through the collector. Solar air heaters because of their inherent simplicity are cheap and most widely used collection devices. These have found several applications including space heating and crop drying. The efficiency of flat plate solar air heater has been found to be low because of low convective heat transfer coefficient between absorber plate and the flowing air which increases the absorber plate temperature, leading to higher heat losses to the environment resulting in low thermal efficiency of such collectors. Several methods, including the use of fins, artificial roughness and packed beds in the ducts, have been proposed for the enhancement of thermal performance. Use of artificial roughness in the form of repeated ribs has been found to be a convenient method. Ribs of various shapes and orientations have been employed and the performance of such systems has been investigated. The use of artificial roughness in solar air heaters owes its origin to several investigations carried out in connection with the enhancement of heat transfer in nuclear reactors and turbine blades.
The thermal conductivity of air is low but we cannot use other gases instead of air for collecting the solar energy, because air is freely and easily available as compare to other gases. However heat transfer in air increases by increasing the time of contact between air and solar duct, so this require the reduction in flow velocity of air in the solar duct. Thus, reduction in the flow velocity of air is achieved by providing artificial roughness in the solar duct.

This roughness reduces the velocity of air by providing obstacles in the flow; it means that more friction is applied in the duct for the flow of air. Due to this roughness, the time of contact of flowing air inside the duct increases which results in the increase in heat transfer.

II. ABOUT THE SOLAR DUCT

Solar air heating is a solar thermal technology in which the energy from the sun is absorbed by air and is used to heat the spaces in building or process heat application such as drying crops (that is tea, corn, coffee) and other drying applications.

In solar duct the flow of air inside the duct is due to blower or fan installed on one end of the duct which sucks the air from the other end of the duct. In the duct, solar collectors are provided which collects the solar energy and transfers it to air as the air becomes dry.

Solar Duct is based on the highly efficient and award-winning Solar Wall system. The technology has been specifically engineered for roof settings and for applications in which a traditional wall mounted system is not feasible. Like the original Solar Wall technology, Solar Duct is a solar heating system that heats ventilation air before it enters the air handling units. The patented system uses an all-metal collector panel (transpired solar collector) and is suitable for commercial, industrial, and institutional facilities. Perforations in the panels allow the heat that normally collects on a dark surface to be uniformly drawn through the Solar Duct panel and then ducted into the conventional HVAC system. The Solar Duct system is optimized to meet site conditions in terms of orientation towards the sun and proximity to rooftop air handling units. The modular arrays are sized according to the energy requirements of the building.

III. THERMO COUPLE & VENTURI METER

A thermoelectric device used to measure temperatures accurately, especially one consisting of two dissimilar metals joined so that a potential difference generated between the points of contact is a measure of the temperature difference between the points.
In other words

1. (General Physics) a device for measuring temperature consisting of a pair of wires of different metals or semiconductors joined at both ends. One junction is at the temperature to be measured, the second at a fixed temperature. The electromotive force generated depends upon the temperature difference
2. (General Physics) a similar device with only one junction between two dissimilar metals or semiconductors
3. A thermoelectric device used to make accurate measurements of temperatures, especially high temperatures. It usually consists of a circuit having two wires of different metals welded together. When one of the

metals is heated, and the other left cold, the difference in temperature causes an electric current to flow through the circuit. Because the amount of electromotive force generated depends on the temperature difference between the two metals, a measurement of the force can be used to calculate the temperature of the heated metal. Thermocouples are also used in the generation of electricity and in refrigeration devices.

Figure 3.1 Thermo couple

VENTURI METER

Venturi meters are flow measurement instruments which use a converging section of pipe to give an increase in the flow velocity and a corresponding pressure drop from which the flow rate can be deduced.

Venturi meter tube –

Venturi tubes are used in processes where permanent pressure loss is not tolerable and where maximum accuracy is needed in case of highly viscous.

Figure 3.2 Venturi meter tube

Flow Rate

A venture meter can be used to measure the volumetric flow rate, Q

$$Q = v_1 A_1 = v_2 A_2$$

$$p_1 - p_2 = \frac{\rho}{2}(v_2^2 - v_1^2)$$

A venturi can also be used to mix a liquid with a gas. If a pump forces the liquid through a tube connected to a system consisting of a venturi to increase the liquid speed (the diameter decreases), a short piece of tube with a small hole in it, and last a venturi that decreases speed (so the pipe gets wider again), the gas will be sucked in through the small hole because of changes in pressure

IV. FEATURES & ADVANTAGES OF SOLAR DUCT

- Heats ventilation air using the highest performing and lowest cost solar collector on the market
- Collector efficiency up to 80%
- Easy to install modular rooftop units
- Optimized to meet site conditions
- Internally ballasted or fastened system which is quick to assemble and simple to integrate into existing air intake system
- Individual units are 6’ by 4’ and each produces 1000 watts of thermal energy
- Typical string length is 48 feet long (8 units) with no limit to array size, and will deliver up to 2000 CFM of heated ventilation air and 8kW of heating
- Substantial CO2 displacement

V. LITERATURE REVIEW

S.C. Lau, R.T Kukreja And R.D Mcmillin [1], examined the turbulent Heat transfer and friction characteristics is fully developed in the flow of air in Continuous discrete channel in which two opposite Walls were roughed with aligned array of Ribs, the angle of attack of the V-shaped Ribs array are 40°, 60°, 90°, 120°, 135°; and it is found that 60° as the angle of attack with p/e=10 has highest Heat transfer in Air.

J.C. Han and Y.M Zhang [2] studied the effect of broken rib orientation on the local heat transfers distribution. The pressure drop in a continuous discrete channel by using two opposite inline ribbed wall was investigated for Reynolds number 15000 to 90000 and the result obtained was so that 60° broken ribs with P/e=10 provides more heat transfer.

Tabish Alam, R.P.Saini, J.S.Saini [3] studied the effect of non-circular perforation holes in terms of V-shaped blockages attached to a heated wall of rectangular duct of solar air heater. Five different holes ranging from circular to continuous discrete and rectangular in the circularity range of 0.6-1.0 has been used with varying relative pitch of 4-12. The relative blockage height was 0.4-1.0, and open area ratio was 5-25%, and angle of attack was 30°-75° with Reynolds number of flow was varied in between 2000-20000. It was also found that the Nusselt number value is in the ratio of 1.13 when circular perforation holes were replaced by rectangular holes of circularity of 0.69.

Rajendra Karwa [4] experimentally investigated that the heat transfer and friction factor in a rectangular duct with rectangular cross-section rib on one broad wall in a transverse inclined V-continues and V-discrete pattern. The duct has width to the height ratio of 7.19-7.75, P/e=10 and roughness height 0.050 and angle of attack 60°, Reynolds number 2800-15000. The rough wall of the duct was uniformly heated while the three walls were insulated and it was founded that V-down discrete arrangement gives best heat transfer performance.

Dhananjay Gupta, S.C. Solanki, J.S. Saini [5] investigated the Thermo-hydraulic performance of solar air heaters with roughened absorber plates. They also found the optimum design used for roughened solar air heaters for varying relative roughness height; and also for a relative roughness pitch which is 10 and the angle of attack of roughness elements is 60°; for the Reynolds no. of 13000 to 19000.

R Karwa, S.C solanki, J.S Saini [6] had an experimental investigation of heat transfer and friction factor for the flow of air in a rectangular duct with recurring chamfered ribs with roughness on one broad wall. The aspect ratios and the rectangular duct used are 4.8, 6.1, 7.8, 9.66, and 12. Only roughened wall was heated and other three walls were insulated; and the boundary condition corresponds closely to those which were found in solar air heaters. The range of parameters studied in Reynolds numbers is from 3000 to 20000. The relative roughness height is from 0.0141 to 0.0328 and the relative roughness pitch are 4.5, 5.8, 7, & 8.5 and Rib chamfer angles are 0°, 5°, 10° ,15° and 18°. In roughness the Reynolds number corresponding to these parameters range from 5 to 60 & was found to have high heat transfer and also the highest friction factor formed 15° chamfered ribs. The heat transfer function is increased with the increase in aspect ratio from 4.65 to 9.65.

P.R. Chandra, C.R Alexandra, J.C.Han [7] conducted an experimental study of surface heat transfer and friction characteristics of a fully developed turbulent air flown over a continuous discrete channel with transverse ribs by one, two, three, and four walls. The tests were performed by Reynolds numbers which are ranging from 10000 to 80000. The pitch of rib height ratio, P=e, was kept at 8 and rib height of channel hydraulic with the diameter ratio, e=DH which was kept at 0.0625. L/DH was 20. The heat transfer enhanced the increase in the Number of ribbed channel walls from 2.16 from one ribbed walls case to 2.57 to four ribbed walls case (Re = 30000). The channel with two opposite ribbed walls shows an increase in heat transfer from the one ribbed walls case by 6%. The three ribbed walls case shows an increase over two ribbed walls case by 5%. In the four ribbed walls case it shows an increase over the three ribbed walls case by 7%. The Friction factor ratio increases with increase in Reynolds number, for this experiment. For Re=30000, the experiments reveals a 214% increase in Case B as compared to Case A, a 72% increase in Case C as compared to Case B, and a 35% increase in Case D as compared to Case C, and also 30% increase of Case E as compared to Case D. Heat transfer performance decreases with increase in Reynolds number and with each additional ribbed wall.

J.L. Bhagoria, J.S.Saini, S.C. Solanki, [8] performed an experiment to collect heat transfer and friction data for forced convection flow of air in a solar air heater with rectangular duct on one broad wall roughened by the wedge shaped transverse integral ribs. This experiment considered the Reynolds number ranging from 3000 to 18000 with wedge height 0.015 to 0.033. The relative roughness pitch are 60.17, 1.0264, P/e=12.12, and Rib wedge angles were 8°, 10°, 12°, and 15°were compared with the smooth duct. In the presence of ribs, yields Nusselt number is up to 2.4 times, while the friction factor rises up to 5.3 times the range of Parameters is then investigated. Therefore, maximum heat transfer occurs in a relative roughness pitch was about 7.57 while the friction factor keeps on decreasing as the relative roughness pitch increases. As maximum enhancement of heat transfer occurs at wedge angle of about 10; while, on either side of the wedge angle, Nusselt number decreases but the friction factor increases and the wedge angle also increases. This Statistical correlations for Nusselt number and friction factor has been developed as the functions of rib spacing used in, rib height, rib wedge angle, and Reynolds number. These correlations have been found to predict the values within the error limits of 12% and 15%.

Abdul-Malik Ebrahim Momin, J.S. Saini, S.C. Solanki [9] performed an experimental investigation which creates an effect of geometrical parameters on V-shaped ribs and on heat transfer and fluid flow characteristics with rectangular duct of solar air heater with absorber plate having V-shaped ribs on which underside have been mentioned. The range of parameters used in this study has been decided on the basis of practical considerations for system and operating conditions. The investigation covered the Reynolds number (Re) range of2500–18000 with relative roughness height of (e/DH) of 0.02–0.034 and angle of attack of flow 30°–90° for a fixed relative pitch of 10. Due to the maximum enhancement of Nusselt number and friction factor, as a result it provides artificial roughness which has been founded with respect to 2.30 and 2.83times that of smooth duct for an angle of attack with 60°. The thermo-hydraulic performance parameters increases the angle of attack of flow and relative roughness height and this maxima occurs with an angle of attack of 60°. It was found that for the relative roughness height of 0.034 and for the angle of attack of 60°, the V-shaped ribs ensures the values of Nusselt number by 1.14and 2.30 times over inclined ribs with smooth plate in case of Reynolds number of 17034.

M.M. Sahu, J.L. Bhagoria [10] conducted an experimental investigation; it has been observed to verify and study the heat transfer coefficient by using 90°, and the broken transverse ribs is on absorber plate used for solar air heater. The roughened wall was been heated while the remaining three walls were been insulated. The provided roughened wall have roughness pitch (P), ranging from 10–30 mm, height of this rib was 1.5 mm and duct aspect ratio provided was 8. The air flow rate corresponds to Reynolds number in between 3000–12000. In the entire range of Reynolds number it was found that the Nusselt number increases, and attains a maximum roughness for pitch of 20 mm and decreases with an increase in roughness pitch. The maximum enhancement of the heat transfer coefficient occurs at pitch of about 20 mm while on both side of this pitch the Nusselt number decreases. The experimental values of the thermal efficiency of a three roughened absorber plates which were tested has been compared with the smooth plates used in the experiment. A plate of roughness 20 mm had the highest efficiency of 83.5%. Roughened absorber plates also increased the heat transfer coefficient 1.25–1.4 times as compared to the smooth rectangular duct under similar operating conditions at a higher Reynolds number.

Alok Chaube, P.K. Sahoo, S.C. Solanki [11] performed an experiment of the computational analysis of heat transfer augmentation and flow characteristics of artificial Roughness in the form of ribs on a heated wall of a rectangular duct for Reynolds number range 3000–20000, which is relevant in a solar air heater. It has been carried out and the analysis demonstrates that the 2D analysis model itself yields which were closer to the experimental ones and was compared with 3D models. The 3D models require much higher memory and computational time as compared to 2D. Secondary flow in transverse rib has no effect in the heat transfer. Thus, it is sufficient to provide a simple 2D model which is more economical with the memory and computational time requirement. The highest heat transfer was been achieved with chamfered ribs and the best performance and index was found with rectangular rib of size 3.5 mm.

Rajendra Karwa, B.K. Maheshwari, Nitin Karwa[12] performed an experimental study on the heat transfer and friction in ducts with baffles attached to only one of the broad walls. The duct has W/H ratio of 7.77, the baffle pitch to height ratio was 29, the baffle of height to duct height ratio was 0.495; The Reynolds number ranges from 2850 to 11500. This baffled wall of the duct was considerably heated and the remaining three walls were insulated. This boundary condition corresponds closely to those which were found in the solar air heaters. Over the range of this study, the Nusselt number of the solid baffles is 73.7–82.7% and for the perforated baffles ranges from 60.6–62.9% to 45.0–49.7% is higher than that of the smooth duct. The friction factor in the solid baffles is found to be 9.6–11.1 times more than the smooth duct, which decreases significantly for the perforated baffles as it increases in open area ratio.

A.R. Jaurker, J.S. Saini, B.K. Gandhi [13] conducted an experimental checking used in the heat transfer and friction characteristics of artificial grooved roughness on one of broad heated wall of a large aspect ratio duct; shows that Nusselt number can be further enhanced beyond that of ribbed duct, keeping the friction factor enhancement low. The experimental investigation ensures the Reynolds number ranging from 3000 to 21000, the relative roughness height 0.0181–0.0363, relative roughness pitch 4.5–10.0, and with groove position to pitch ratio 0.3–0.7. The effects of importance of these parameters on the heat transfer coefficient and the friction factor have been discussed and these results were compared with the results of ribbed and smooth duct under the same flow conditions. The presence of rib grooved artificial roughness yields the Nusselt number up to 2.7 times and also the friction factor rises up to 3.6 times within the range of parameters investigated. The maximum heat transfer occurred of a relative roughness pitch is about 6.0.

VI. ANALYTICAL CALCULATION

The experimental data obtained by test runs have been reduced to obtain the average plate temperature (Tp), average fluid (air) temperature (Tf) in the test section, mass flow rate (m) and Reynolds number (Re). This reduced data was used to determine the heat transfer coefficient (h) between absorber plate and air flow through the duct, also the Nusselt number (Nu) and friction factor (f) were calculated by using this data.

The average or mean plate temperature have been calculated by taking the arithmetic mean of the plate temperatures taken at 12 different positions and is given by

$$T_p = \frac{T_6 + T_7 + T_8 + T_9 + T_{10} + T_{11} + T_{12} + T_{13} + T_{14} + T_{15} + T_{16} + T_{17}}{12}$$

Similarly the average fluid (air) temperature underside the absorber plate has been taken as the average of the inlet and outlet temperature of the test section and is expressed as

$$T_f = \frac{T_o + T_i}{2}$$

Where T_i is the average air inlet temperature which have been calculated by recording the temperature of air at four different locations at the inlet of test section and T_o is the outlet temperature of the air from the test section measured at four different locations.
Heat gained by the air was calculated by based on the flow rate of air as

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

Then the heat transfer coefficient was calculated from heat gained Q_u as

$$h = \frac{Q_u}{A_c(T_p - T_f)}$$

The heat transfer coefficient (h) was then used to obtain the Nusselt number as follows

$$h = \frac{hD_h}{k}$$

The friction factor was determined from the measured values of pressure drop $(\Delta p)_d$ across the test section as given below,

$$f = \frac{2D_h(\Delta p)_d}{4\rho L V^2}$$

Uncertainty Analysis

During the conduct of experimentation maximum possible care and precautions are taken; but errors do creep into the measurement of various parameters. This may occur due to the least count of the instruments used in the experimentation. Therefore, it is desired to estimate the possible inaccuracy (error) in measurement during any experiment. The degree of inaccuracy or the total measurement error is the difference between the measured quantity and its true value.
If a parameter is calculated using certain measured quantities as,

$$y = f(x_1, x_2, x_3, \dots\dots\dots.x_n)$$

Then uncertainty in measurement of "y" is given by

$$\frac{\delta y}{y} = \left[\left(\frac{\delta_y}{\partial x_1}\delta x_1\right)^2 + \left(\frac{\delta_y}{\partial x_2}\delta x_2\right)^2 + \left(\frac{\delta_y}{\partial x_3}\delta x_3\right)^2 \dots\dots\dots\dots + \left(\frac{\delta_y}{\partial x_n}\delta x_n\right)^2\right]^{0.5}$$

Where, δx_1, δx_2, δx_3, …… δx_n are the possible errors in measurements of $x_1, x_2, x_3, \dots\dots x_n$.
δ_y is known as absolute uncertainty
$\frac{\delta_y}{y}$ is known as relative uncertainty.

In the present work Reynolds number (Re), Nusselt number (Nu) and friction factor are the main parameters which are related to the heat transfer and performance of the solar air heater and hence the maximum possible uncertainty in the calculated values of these parameters is of interest and was calculated by using the above mentioned procedure and calculation of uncertainty for the these parameter are given in following subsections

Heat transfer Coefficient (h)

The heat transfer coefficient $h = \frac{Q_u}{A_p.(T_p - T_f)}$ or $h = \frac{Q_u}{A_p.\Delta T}$

The uncertainty in heat transfer coefficient (h) be calculated as

$$\frac{\delta h}{h} = s\left[\left(\frac{\delta Q_u}{Q_u}\right)^2 + \left(\frac{\delta A_p}{A_p}\right)^2 + \left(\frac{\delta(\Delta T)}{\Delta T}\right)^2\right]^{0.5}$$

Nusselt number (Nu)

$$Nu = \frac{hD_h}{k}$$

$$\frac{\delta Nu}{Nu} = \left[\left(\frac{\delta h}{h}\right)^2 + \left(\frac{\delta D_h}{D_h}\right)^2 + \left(\frac{\delta k}{k}\right)^2\right]^{0.5}$$

Reynolds Number (Re)

$$Re = \frac{\rho V D_h}{\mu}$$

$$\frac{\delta Re}{Re} = \left[\left(\frac{\delta V}{hV}\right)^2 + \left(\frac{\delta\rho}{\rho}\right)^2 + \left(\frac{\delta D_h}{D_h}\right)^2 + \left(\frac{\delta\mu}{\mu}\right)^2\right]^{0.5}$$

Friction factor (f)

$$\boldsymbol{f} = \frac{2(\Delta P)tD_h}{4\rho L V^2}$$

$$\frac{\delta f}{f} = \left[\left(\frac{\delta V}{V}\right)^2 + \left(\frac{\delta\rho}{\rho}\right)^2 + \left(\frac{\delta D_h}{D_h}\right)^2 + \left(\frac{\delta L}{L}\right)^2 + \left(\frac{\delta(\Delta P)d}{(\Delta P)d}\right)^2\right]^{0.5}$$

The uncertainty in all the measurements were calculated in the similar way and maximum uncertainties in Reynolds number (Re), Nusselt number (Nu) and friction factor (f)

VII. EXPERIMENTAL SET-UP

An experimental set-up to study the effects of Continuous discrete ribs with symmetrical gaps on heat transfer and fluid flow characteristics in a rectangular duct has been designed and fabricated as per the ASHRAE standard recommendation. The schematic diagram of the experimental set-up and test plate is shown in Fig.3.1. The rectangular duct is 2395 mm long with a flow cross section area of 300 mm×25 mm, fabricated from ply board. The duct has an inlet, test and exit section of 740 mm, 1100 mm and 555 mm length respectively; equivalent to 16 D, 24 D and 12 D respectively. The duct is insulated with 50 mm thick polystyrene insulation of 0.037W/mK thermal conductivity. An aluminum test plate of dimension 300 mm×1100 mm provided with a roughness is placed on the top of the test section to form a roughened wall of the duct. A uniform heat flux of 1000 W/m^2 was supplied to the roughened plate by an electric heater. Air was circulated in the duct by a 2 HP centrifugal blower. The mass flow rate of air was calculated by measuring the pressure drop across the orifice meter employed in the circulation pipeline and the pressure drop across the test section was measured with the aid of digital micro-manometer, Having least count of 0.01 Pa. The temperature of heated plate at 21 locations was measured under the steady state condition by calibrated J-type copper constantan thermocouples connected to digital temperature display via selector switch. The exit temperature of the air was recorded at three different locations in the Transverse direction to get the mean air temperature at the outlet and; inlet air temperature was measured by placing a thermocouple at the inlet section.

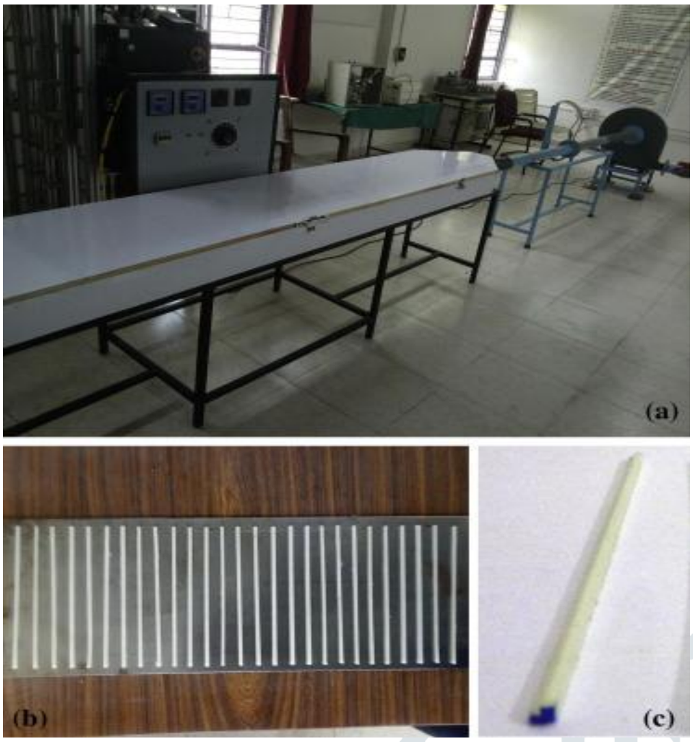

Fig. 7.1 Schematic of the test setup and test plate

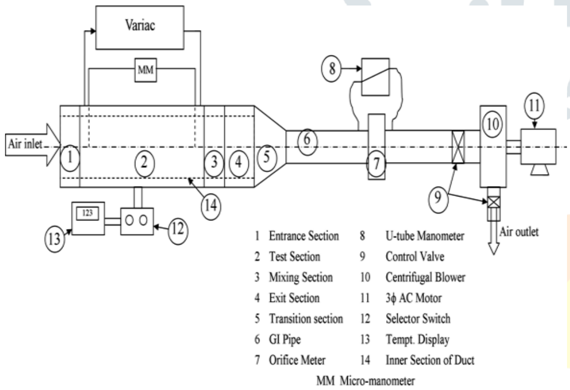

Figure 7.2: Schematic diagram of the experimental set-up

VIII. RESULT AND DISCUSSION

8.1 Result Obtained From Experimentally Solar Duct with Roughened Absorber Plate with Smooth Plate

8.1.1 Table and Graph obtained from Experimental nusslet number of Smooth plate

Table 8.1 Nusslet number of Smooth plate

Reynolds Number	Nusslet Number
4000	16.13307
8000	27.39476
12000	37.59050
16000	46.86068
18000	53.47757

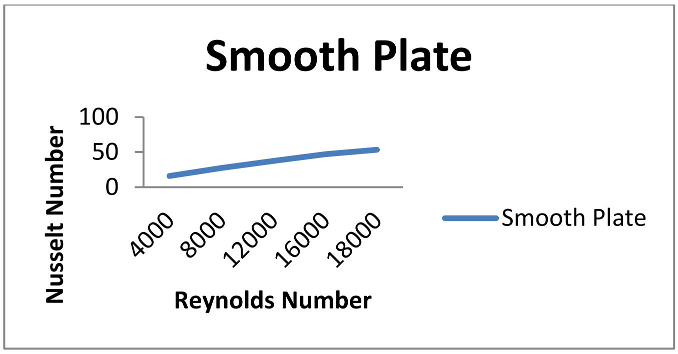

Figure 8.1 Graph of Nusselt Number versus Reynold Number

8.1.2 Table and Graph obtained from Experimental Friction factor of Smooth plate

Table 8.2 Friction factor of Smooth plate

Reynolds Number	Smooth plate
4000	0.010134
8000	0.008738
12000	0.007767
16000	0.007243
18000	0.007112

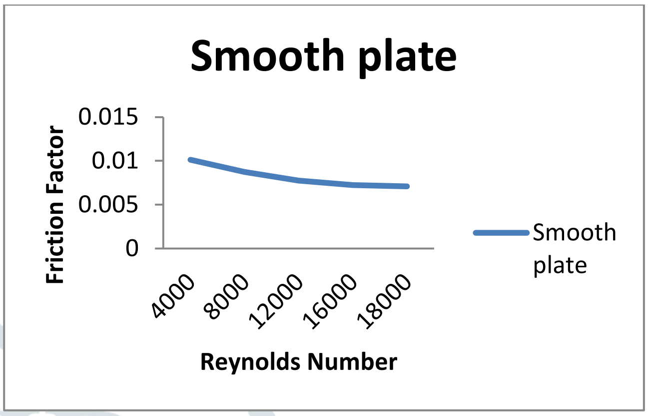

Figure 8.2 Graph of Friction factor versus Reynolds Number of smooth plate

8.2 Result Obtained From Experimentally Solar Duct with Roughened Absorber Plate with Continuous Discrete Ribs

8.2.1 Nusselt Number of Continuous Discrete Ribs of Experimental

Table 8.3 Nusselt Number of Continuous Discrete Ribs of Experimental

Reynolds Number	Nusselt Number
4000	22.86523
8000	33.17355
12000	43.8112
16000	55.4399
18000	70.1615

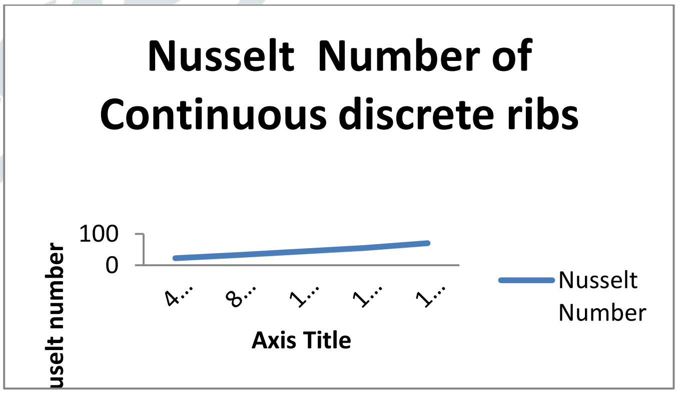

Figure 8.3 Graph of Nusselt Number versus Reynolds Number of continuous discrete ribs

8.2.2 Table and Graph from Experimental friction factor for continuous discrete ribs

Table 8.4 Friction Factor for continuous discrete ribs

Reynolds Number	Continuous discrete ribs
4000	0.0502668
8000	0.0458720
12000	0.0368920
16000	0.0298350
18000	0.0342350

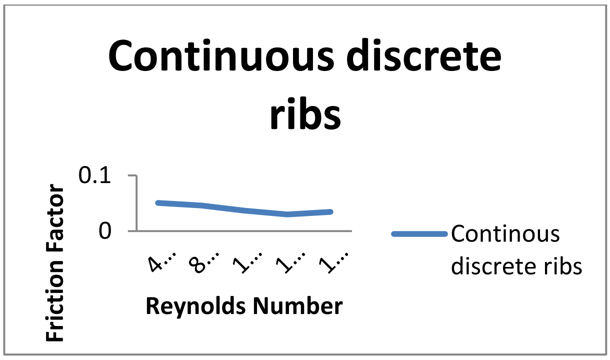

Figure 8.4 Graph of Friction Factor versus Reynolds Number of continuous discrete ribs

Table 8.5 Comparision for Nusselt Number of smooth plate and continuous discrete ribs with different Reynolds Number

Nusselt Number		
Reynolds Number	Continuous discrete	Smooth plate
4000	22.86523	16.13307
8000	33.17355	27.39476
12000	43.8112	37.59050
16000	55.4399	46.86068
18000	70.1615	53.47757

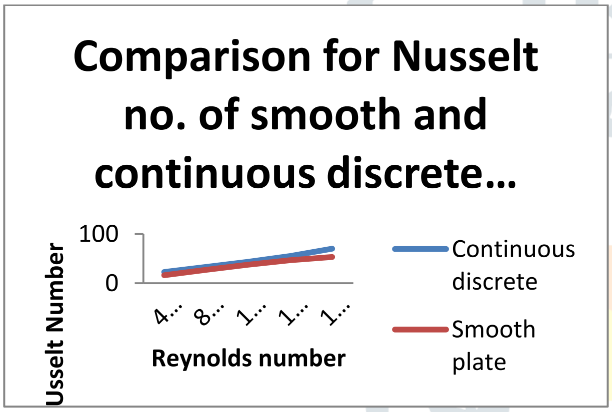

Figure 8.5 Comparison Graph of Nusselt Number versus Reynolds Number of smooth plate and continuous discrete ribs

Table 8.6 Friction Factor of smooth plate and continuous discrete ribs with different Reynolds Number

Frition Factor		
Reynolds Number	Continuous discrete ribs	Smooth plate
4000	0.0502668	0.010134
8000	0.0458720	0.008738
12000	0.0368920	0.007767
16000	0.0298350	0.007243
18000	0.0342350	0.007112

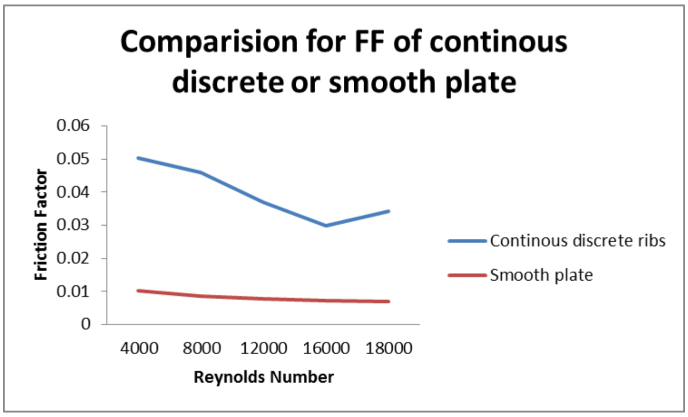

Figure 8.6 Comparison Graph of Friction factor versus Reynolds Number of smooth plate and continuous discrete ribs

Table 8.7 Thermo-Hydraulic performance comparison table for continuous discrete ribs and smooth plate

Reynolds Number	Continuous discrete plate	Smooth
4000	1.824939	1.705939
8000	2.32616	1.897882
12000	3.159923	2.393248
16000	2.604093	2.10788
18000	2.421878	2.012641

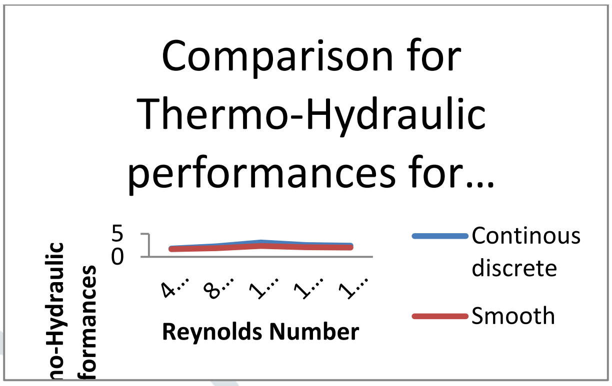

Figure 8.7 Graph of Thermo-Hydraulic performances for continuous discrete ribs and smooth plate at different Reynolds number

IX CONCLUSION

- Average deviation of result obtained from EXP for smooth & Continuous discrete ribs plate for Nu number & Friction factor lies within the range, average Nu Number is deviate 3.76% for smooth plate and Average Friction factor is deviate 3.91% for smooth plate.
- Average deviation of results obtained for continuous discrete ribs from EXP in Nu Number is deviated by 17.01 % i.e., Nu Number increases for Continuous discrete ribs plate at each Reynolds number taken for experimentally.
- Average deviation of result obtained for continuous discrete ribs from EXP in Friction factor is deviated by 20.15% i.e., Friction factor increases for Continuous discrete ribs plate at each Reynolds number taken for experimentally.
- Thermo-Hydraulic performance increases at Reynolds number 4000 for continuous discrete ribs by 6.7%; and for Reynolds number 8000, 12000, 16000, 18000 it increases by 18.63%, 24.12%, and 19.35%, 16. 97% respectively.
- This experiment clearly indicates that continuous discrete ribs roughness increases the turbulence in the air; and at the contact, the heat transferring area of air is increased which results in the increase in Nu Number and Friction factor.

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