Experimental Investigation of Heat Transfer and Friction with Double Arc Reverse Shaped Roughness on Absorber Plate of Solar Air Heater

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Abstract - In this paper, experimental investigation has been carried out to study the heat transfer coefficient and friction factor characteristic of duct by using Double Arc reverse shaped roughness rib on the absorber plate (1.5 X 0.2 m) has been compared with the result of smooth plate. Experiment has been performing on outdoor set-up. In this experimental investigation, one smooth plate and one absorber plates having roughness of circular wire of 1.2 mm diameter fixed in Double arc reverse shaped roughness were tested in duct section. Result find in 5 test runs of Reynolds number range from 3,000 to 11,000. According to the roughness geometry, turbulence occurs in fluid flow and in this way heat transfer is increased. Thermal efficiencies of this absorber plate were 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 plate, the heat transfer coefficient in such ducts can be improved by factor 28% To 34% as compared to that of smooth plate. Similarly the friction factor has been found to increase of 1.5 to 2.7. The Thermal efficiency has been increased by around 22% as compare to smooth plate.

Keywords:- Double arc reverse shaped roughness, friction factor, heat transfer coefficient, solar air heater, Pressure drop.

I. INTRODUCTION

Energy action in the universe is an expression of energy in one form or the other. The living standard can be directly related to per capita energy consumption. Presently major portion of our energy demand is met by crude oil which supplies nearly 39%, natural gas about 20% and coal about 33%. The present energy sources as exhaustible and are depleting fast. The present energy consumption is about 0.3 to 0.5 Q/year (1Q = 1010 KJ) whereas availability in the form conventional energy resources such as coal, oil and natural gas is 35Q. Conventional energy sources are not sufficient to meet the energy demands for very long. A systematic study of various forms of energy and energy transformation, involving human experience and observation is called energy science. The applied part of energy science useful to human society, nation and individual is called energy technology. A conventional solar air heater consists of an absorber parallel plate blow forming a passage of high aspect ratio through which the air to be heated flows as shown in figure 1.1 and figure 1.2. Like a liquid flat plate collector, a solar air heater is simple in design and requires little maintenance.

In addition, since the fluid does not freeze, the solar air heater has the advantage of not requiring any special attention at temperature below 0° C, corrosion and leakage problems are also less severe. However, the value of the heat transfer coefficient between the absorber plate and air is low and these results roughened or longitudinal fins are provided in the flow passage. A roughness element has been used to improve the heat transfer coefficient by creating turbulence in the flow. However, it would also result increase friction losses and hence greater power required for pumping air through the duct. In order to keep the friction losses at a low level, the turbulence must be created only in region very close to duct surface, i.e. in laminar sub layer. Solar air heater is a device to produce hot air for drying application by using freely available sun light, without using conventional fuels like electricity, diesel, LPG, firewood, coal, etc., but it could be coupled with an existing conventional drying systems like tray driers, tunnel Driers, Conveyor Drier, FBD drier and bin drier operated by conventional fuels to save fuel consumption.

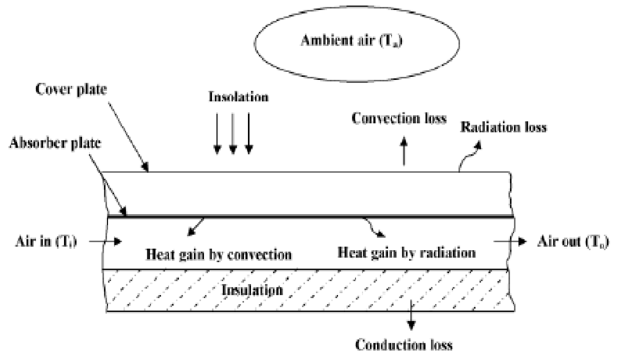

Fig. 1.1 Air Flow through the Duct

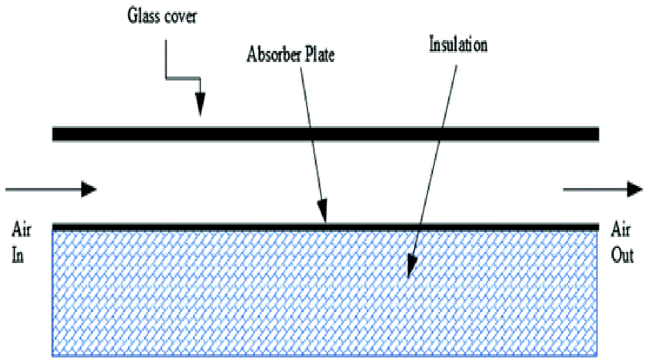

Fig. 1.2 Air Flow between Glass Cover and Absorber Plate

II. LITERATURE REVIEW

A. **Austin Whillier [44]** suggested the use of crimped galvanized steel plates with close spaced crimps so as to produce series of random ridges with sharp edge in order to ensure high level of turbulence in the boundary layer of the air stream to increase the heat transfer coefficient between the air and the absorber plate.

B. **Gupta and Garg [43]** carried out performance studies on four types of solar air heaters, two of corrugated and the other two of wire mesh type. The efficiency was compared for four types of air heaters for the same amount of pumping power. The rating parameters like efficiency factor and heat loss coefficient have reported for winter conditions.

C. **Prasad and Mullick [7]** experimentally investigated the effect of a small diameter wire attached to the underside of the absorbent plate as an artificial roughness.

D. **Webb et al. [40]** developed friction and heat transfer correlations, for turbulent flow in tubes having repeated rib-roughness, based on law of the wall similarity and application of the heat-momentum transfer analogy to flow over a rough surface, respectively.

E. **Han et al. [8]** developed the correlation for friction factor and heat transfer, in order to define roughness geometry which gives the best heat transfer performance for given flow friction.

F. **Saini and Saini [39]** carried out experimental investigation for fully developed turbulent flow in a rectangular duct having expanded metal mesh as artificial roughness, and developed correlations for Nusselt number and friction factor in terms of geometry of expanded metal mesh.

G. **Karwa et al. [37]** carried out experimental investigations, to develop the correlation of heat transfer and friction, for flow of air in rectangular ducts with integral and repeated chamfered rib roughness on one board uniformly heated wall, and remaining walls insulated. They observed that the Stanton number and friction factor take their maximum values at the chamfer angle of 15°.

H. **Verma and Prasad [38]** developed the heat transfer and friction factor correlation for roughness elements consisting of small diameter wires, and evaluated the thermo-hydraulic performance.

I. **Jaurker et al. [9]** developed the correlations for Nusselt number and friction factor, for rib-grooved artificial roughness on one broad heated wall. They carried out the thermo-hydraulic performance analysis of air duct (solar air heater), based on efficiency index, and concluded that rib grooved arrangement is better than rib only. Similar investigations for heat transfer and fluid flow characteristics have been carried out by **Gupta et al. [10]** for transverse wire roughness; **Momin et al. [11]** for V shaped ribs; **Bhagoria et al.[12]** for wedge shaped rib; **Sahu and Bhagoria [13]** for broken transverse ribs; and **Layek et al. [14]** for chamfer red rib-groove roughness.

III. EXPERIMENTAL INVESTIGATION

A. Experimental Set-Up

The experimental set-up is an open flow loop that consists of a test duct with entrance and exit section, a blower, control valve, orifice plate and various devices for measurement of temperature & fluid head. A schematic diagram of outdoor experimental set-up including test section is shown in figure 2.1 and photograph of the same is given in fig. 2.2. The flow system consists of an entry section, a test section, an exit section, a flow meter and a centrifugal blower. The set-up consists of two identical wooden ducts. Each duct is of size 2030 mm x 200 mm x 25 mm (dimensions of inner cross section) and is constructed from wooden panels of 32 mm thickness. The test section is of length 1500 mm. One of the duct carries the roughened absorber plate at the top, while the other duct carries a smooth absorber plates. The sun facing side of absorber plates are smooth and coated with blackboard paint. An unheated entrance duct length of 177 mm is provided. A short entrance length has been chosen because for a roughened the fully developed thermal boundary layer establishes in short length of 2 to 3 hydraulic diameters. A 353 mm long exit section is installed to remove any downstream effect on the test section. It may be noted that ASHRAE Standard 93-97[45] recommends entry and exit length of $2.5\sqrt{WH}$ and $5\sqrt{WH}$ *respectively*, ie.177 and 353 mm respectively for the duct. The entire set-up is insulated with 25mm thick thermocol. The top side of heated test section carries about 1 mm thick G.I. plate with required rib roughness on the lower side and this forms the top wall of the duct, while bottom wall is formed by 32mm wood with insulation below it. The top side of the entry and exit sections of the duct is covered with smooth faced 12mm plywood. 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 circular G.I. pipe provided was based on pipe 49 diameter d1 which is minimum of 5d1 on the upstream side and 10d1 on the downstream side of the orifice plate. In the present experimental set-up we used 900mm (17d1) pipe length on the downstream side and 600mm (11.3d1) on the upstream side. Butt welded 0.36 mm copper-constantan thermocouples, calibrated against mercury thermometer of 0.10 centigrade least-count, and used to measure the air and the heated plate temperature at different locations. A digital voltmeter is used to indicate the output of thermocouples. The test was performed on the clear sunny days of May at Bhopal between 11:00 hrs. - 16:00 hrs. The air flow rate through the ducts is regulated with the help of the gate valves. All joints are properly sealed to make them air leak proof.

B. Instrumentation

- **Roughened Plate:** It is a G.l. sheet (1mm thick) with dimension 1500mm x 20mm in size and artificially roughened at the bottom of plate. The top surface of the plate was painted black with black board paint. Artificial roughness was produced on the bottom side by Double arc manner. The Geometry of Roughness plate shown in Figure 2.4.
- **Air Handling Equipment**: The exit section of the duct has been connected to a blower as shown in fig. 2.1. The blower driven by a three phase 440 V, 3.5 kW, A.C. motor sucked in air through the duct. The gate valve was used at the blower outlet of air through the duct.
- **Air Flow Measurement**: The flow rate of air through the duct was measured by means of a pre-calibrated orifice meter. It is provided in circular pipe, a vertical U-tube manometer with water as manometric fluid

measured the pressure drop across the orifice plate which leads to the measurement of air flow through the duct.

- **Pressure Drop in Ducts**: A digital micro manometer was used for the measurement of pressure drop across the test section.
- **Solar Radiation Measurement**: The solar radiation intensity was measured by digital solar power meter.

C. Experimental Procedure

All components of the experimental set-up and the instruments have been checked for proper operation. The glass covers of the collectors were cleaned before starting the experiments. The blower is then switched on and the joints of the set-up are checked for air leakage with soap bubble technique. Micro manometer is functioning well and vertical U-tube manometer is properly levelled. Blower is switched on and the flow control valve is adjusted to give a predetermined rate of airflow to the test section. In order to reduce the effect of in accuracy of the calculation of the heat transfer coefficient, the temperature of the air through the duct has been maintained greater than 10ºC and the temperature difference between the heated plate and the bulk air temperature has been kept above 20ºC. During the experimentation the temperature of air entering the duct ranges between 38°C to 43.6°C according to the local atmospheric conditions. The temperature of the air at the outlet of the test section ranges between about 41°C to 73°C. All readings have been noted that under steady state condition which was assumed to have been obtained when the plate and air outlet temperature did not deviate over a 15 min. period. The steady state for each run has been observed to arrive in about 2 to 2.5 hrs. After the steady state has reached the plate temperature, the inlet and exit air temperatures and the pressure drop across the duct and across the orifice plate have been recorded. For one rib configuration 5 runs have been conducted at air-flow rated corresponding to the flow Reynolds numbers between 2,000 and 11,000. By fixing Reynolds number reading was taken at different time from 11:00 hrs. - 16:00 hrs. and observing the variation of heat transfer and friction factor with the insolation.

The following parameters were measured during the experiments:

1. Pressure drop across the orifice plate
2. Inlet air temperature of collectors
3. Outlet air temperature of collectors
4. Temperature of plate
5. Solar radiation intensity

IV. RESULT AND DISCUSSION

A. Experimental Data

Experimental data have been collected for one roughness plate. Many runs have been taken with different mass flow rate for each day. For one day, by fixing mass flow rate temperature reading is collected for different insolation at different time duration. Same procedure is repeated on next day with variation in mass flow rate. The experiment data has been used to determine the heat transfer coefficient (h), Nusselt number (Nu), Friction factor (f). The effect of various flow and roughness parameters on thermal efficiency for the flow of air in rectangular ducts of roughness. Double arc reverse shaped of the present investigation on outdoor set-up are being discussed below. Results have been compared with those of smooth duct under similar flow and geometrical roughness condition to see the enhancement in heat transfer coefficient and friction factor.

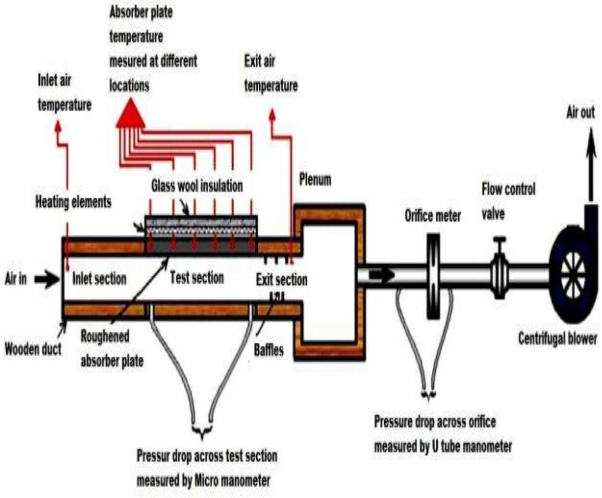

Fig. 2.1 Schematic Diagram of Experimental Outdoor Set-Up

Fig. 2.2 Photograph of Outdoor Experimental Set-Up

Fig. 2.3 Smooth Plate

Fig. 2.4 Roughened Plate with 12mm pitch and 30º arc angle

The aim of our analysis is to increase collector efficiency with using artificial roughness on collector plate. The effect of various flow and roughness parameters (e/D=0.0270) on rectangular plate for same pitch (12 mm) result have been compared with the Reynolds number Vs Nusselt number, Reynolds number Vs friction factor, Reynolds number Vs thermal efficiency, Nusselt number Vs Thermal efficiency, Reynolds number Vs heat transfer coefficient and insolation vs heat transfer coefficient at different Reynolds number. Increasing the mass flow rate, there occurs, increase in efficiency. However, the outlet temperature of air increases significantly. 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 radiation value change in range and it reaches in the midday. Fig 3.1 & 3.2 shows Validation of Nusselt Number and Friction Factor (Experimental value and Formulated Values) with Reynolds Number. From Figure 3.1 & 3.2 found 14%-15.5% difference in Valdiation of Nusselt Number by Experimental values. Similarly 9.4%-14.6% difference in Valdiation of friction factor by Experimental values .Fig.3.3 & 3.4 shows the variation of heat transfer coefficient & Nusselt number with Reynolds number for Double Arc reverse shaped roughness. It is found that with the increase of Reynolds number heat transfer increases because Nusselt number is nothing but the ratio of conductive resistance to convective resistance to the heat flow and as Reynolds number increases thickness of boundary layer decreases and hence convective resistance decreases which in turn increase the Nusselt number. If the Reynolds number is low then the thermal boundary layer remains unbreakable, which offers resistance to heat flow and hence low heat transfer coefficient results. It can be seen from fig 3.4 that the Nusselt number values increase rapidly with the Reynolds number increase. Fig 3.3 & 3.4 also shows the comparison of roughened plate with smooth plate. Fig.3.5 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 highest (60.23%) for rough plate at higher mass flow rate (0.02015 kg/s) and at same mass flow rate the efficiency of smooth plate is 48.65%. When the Reynolds number is low the roughness elements lie within the thermal boundary layer and velocity of air is not sufficient to break the thermal boundary layer and hence heat transfer to the air occurs with less efficiency. But with the large Reynolds number, mass flow rate is more and velocity of air is sufficient to break the thermal boundary layer which leads to increase the thermal efficiency of collector plate. Fig 3.6 showing, at Re = 3000 with the increase of insolation heat transfer coefficient increases. And we get the highest Heat Transfer Coefficient (h = 21.15 W/m^2 ºC) at Insolation (I = 1045 W/m^2) with Re = 11000 Fig 3.7 shows the variation of Thermal efficiency with Nusselt number. It is clearly seen from the figure that Thermal efficiency increases with Nusselt number up to an extent and we got the highest thermal efficiency (η th=60.23) at Nu = 30.93.

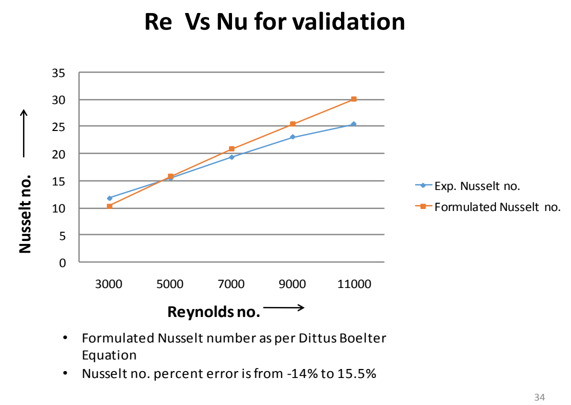

Fig. 3.1 Validation of Nusselt Number (Experimental Vs Formulated Value) with Reynolds Number

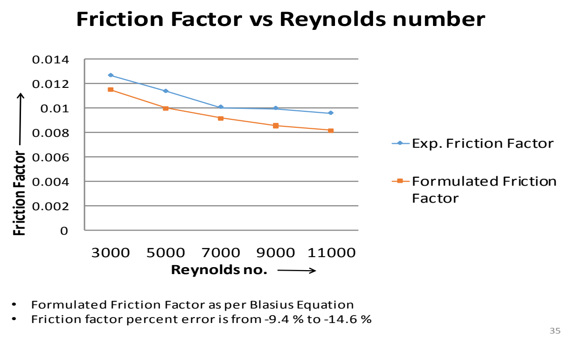

Fig. 3.2 Validation of Friction factor (Experimental Vs Formulated Value) with Reynolds Number

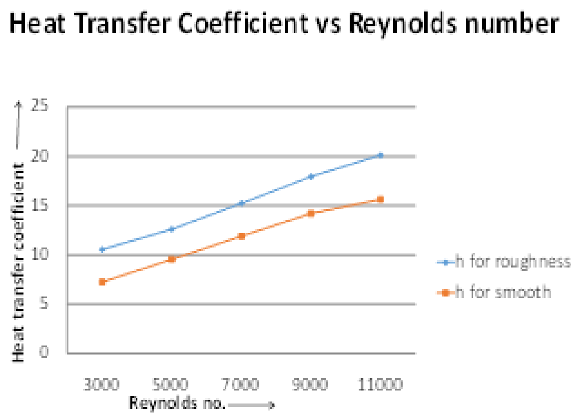

Fig. 3.3 Heat Transfer Coefficient Vs Reynolds Number (For Smooth & Rough Plate)

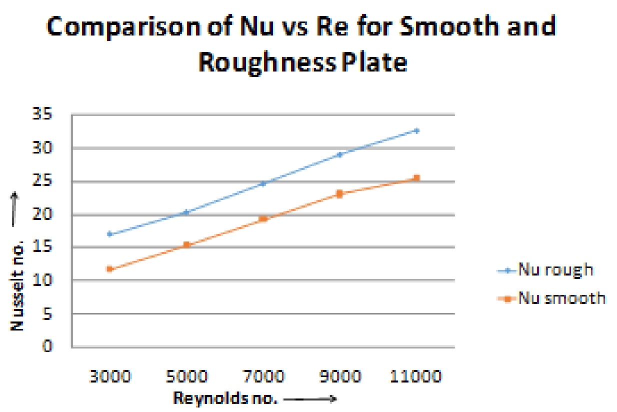

Fig. 3.4 Nusselt Number Vs Reynolds Number (For Smooth & Rough Plate)

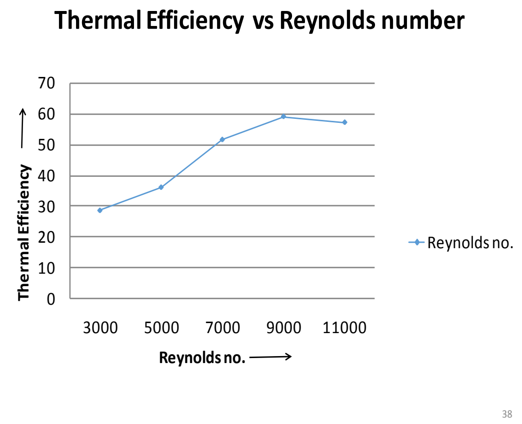

Fig. 3.5 Thermal Efficiency Vs Reynolds Number

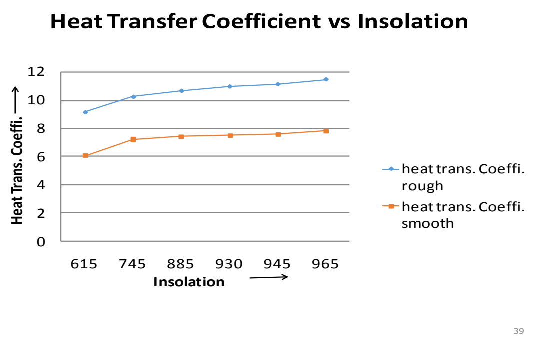

Fig. 3.6 Heat Transfer Coefficient Vs Insolation (For Smooth & Rough Plate)

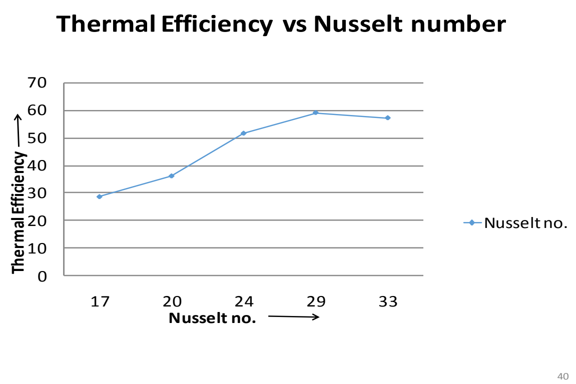

Fig. 3.7 Thermal Efficiency Vs Nusselt Number

V. CONCLUSION

The conclusion can be drawn from the experimental study are:

1. Friction Factor decreases with increase in Reynolds number.
2. Thermal Efficiency increases with increase in Reynolds number.
3. While comparing experimental value of Nusselt number with theoretical value, it is found that the experimental values are within a range of predicted value.
4. The solar air heater with roughened absorber plate performs better as compare to smooth plate solar air heater when intensity of solar radiations is high.
5. Thermal efficiency of roughened duct is increases as compare to smooth duct at same Reynolds number.
6. When the surface roughness increased, the heat transfer and pressure loss increases.
7. The experimental value of thermal efficiency of the absorber plate tested has been compared with the smooth plate. Rough plate with Double arc reverse shaped gives high efficiency **60.23%** at Reynolds number **(9234)** at Insolation **1051 W / m^2**.

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