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Heat Transfer and Friction Factor Analysis for Solar Air Heater Duct Roughened Artificially With Zig-Zag Flow Pattern Broken Arc Ribs

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***Abstract*—Experiments conducted to study the heat transfer and friction of artificial rib grooves in the hot walls of large pipes show that the Nusselt number can be increased more than in water-core pipes while maintaining a low coefficient of friction. Try the Reynolds number between 3000 and 21,000; relative roughness height is 0.0181 to 0.0363; the effect of these values on the heat transfer coefficient and friction coefficient is discussed, and the results are compared with the results of ribbed and smooth pipes under similar conditions. This study clearly shows that the difference between the rib-groove configuration is higher than the rib center, while the friction coefficient of the rib-groove configuration is small. Slightly higher and rib height than rectangular shape with similar height. Performance conditions have been determined. In order to properly predict the results, a correlation was established between the Nusselt number and the friction coefficient.**

***Keywords*— solar air heater, artificially roughness geometry, Reynolds number, Nusselt number, friction factor, rib roughness.**

I. INTRODUCTION

In It has been observed that the heat transfer coefficient between the absorber plate and working fluid of solar air heater is generally low. It is attributed to the formation of a very thin boundary layer at the absorber plate surface commonly known as viscous sub-layer. This convective heat transfer coefficient can be increased by providing the artificial roughness on the heat transferring surface (Webb, 1987). It has been found that the artificial roughness applied on the heat transferring surface breaks the viscous sub-layer, which reduces thermal resistance and promotes turbulence in a region close to artificially roughened surface. Although the application of artificial roughness in the duct of a conventional solar air heater has been shown to be an efficient method of enhancement of thermal efficiency of solar air heater, however, the use of artificial roughness results in higher friction losses leading to excessive power requirement for the air flow through the duct. It is therefore desirable that the turbulence must be created only in the region very close to the heat transferring surface i.e. in the viscous sub-layer only where the heat transfer takes place and the core flow should not be unduly disturbed so as to avoid excessive friction losses. This can be done by keeping the height of the roughness elements to be small in comparison with the duct dimensions (Saini, 2004). 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, cooling of turbine blades and electronic components. Nikuradse, 1950, Dipprey and Sabersky, 1963 developed friction similarity law for heat and momentum transfer analogy for fluid flow in rough pipes. Webb et al. (1971) developed heat transfer and friction factor correlations for turbulent air flow in tubes having rectangular repeated rib roughness. Han and Park, 1988, Han et al., 1989 conducted extensive experiments to simulate the flow through turbine blades. For developing and fully developed turbulent flow, heat transfer and friction characteristics of ducts with rib turbulators on two opposite walls of the square and rectangular ducts have been extensively studied. The results show that angled or inclined ribs give higher heat transfer than transverse ribs, and narrow aspect ratio ducts perform better than wide aspect ratio ducts. The angled ribs give higher heat transfer rate than the transverse ribs because of the secondary flow induced by the rib, in addition to breaking the viscous sub-layer and producing local wall turbulence. The concept of combined turbulence promoters in the form of rib-groove, with groove at the centre of ribs was introduced by Zhang et al. (1994). Their results show that the heat transfer performance of the rib-grooved roughened duct is much better than that of the rib-roughened duct. The rib-grooved roughened walls enhance the heat transfer 3.4 times and pay six times the pressure drop penalty, whereas the rib roughened walls, with similar rib height and rib spacing, enhance the heat transfer 2.4 times and pay about the same pressure drop penalty. They also reported that the flatter velocity profile and higher turbulence intensity are responsible for producing higher heat transfer. The use of such roughness on the absorber plate of solar air heater makes the fluid flow and the heat transfer characteristics distinctly different from those found in the case of two roughened walls and four heated walls duct. Prasad and Saini, 1988, Saini and Saini, 1997 applied the artificial roughness in the form of circular wires and expanded metal respectively, in the form of artificial roughness on the inner surface of absorber plate of solar air heaters. They reported a considerable enhancement in heat transfer coefficient and friction factor and hence the enhancement of the efficiency of solar air heater.

The objective of the present investigation is to generate friction and heat transfer data pertinent to the heating of air in a rectangular duct with rib-grooved transverse repeated rib roughness on one broad heated wall for Reynolds numbers varying between 3000 and 21,000. The statistical correlations for Nusselt number and friction factor in terms of roughness parameters have been developed and the effect of system and operating parameters has been discussed.

II. LITERATURE REVIEW

Anil kumar et al [1] have conducted an experiment investigation of heat transfer and friction analysis in the flow of air in rectangular ducts having multi V-shaped rib with gap roughness on one broad wall. The investigation encompassed by varying Reynolds number (Re) 2000-20,000, relative gap width (g/e) values of 0.5-1.5, relative roughness pitch (P/e) values of 6- 12, relative roughness height (e/D) values of 0.022-0.043, relative gap distance (Gd/Lv) values

of 0.24-0.80, relative roughness width ratio (W/w) values of 1-10, angle of attack (α) 30^{0}-75^{0}.The results are compared with the results of ribbed and smooth duct under similar floe

conditions. The investigation clearly demonstrates that the maximum enhancement in Nusseltnumber (Nu) and friction factor (f) is 6.74 and 6.37 times of smooth duct, respectively. Themaximum value of friction factor occurs for multi V-shaped with gap rib with relative

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Sukhmeet singh et al [2] have conducted the heat and fluid flow characteristics of rectangular duct having its one broad wall heated and roughened with periodic discrete V-down rib are experimentally investigated. The investigation encompassed by varying Reynolds number (Re) 3000-15000, relative gap position (d/W) values of 0.20-0.80, relative gap width (g/e) values of 0.5-2.0, relative roughness pitch (P/e) values of 4-12, angle of attack (α) 30^{0}-75^{0}, relative roughness height values of (e/Dh) values of 0.015-0.043. The experimental investigation clearly demonstrates that the maximum increase in Nusselt (Nu) number and friction factor (f) over the smooth duct is 3.04 and 3.11 folds respectively.

K.R.aharwal et al [3] have carried out an experimental investigation of heat transfer and friction characteristics of solar air heater ducts having integral inclined discrete ribs on absorber plate. The experiment encompassed by varying Reynolds number (Re) 3000-18000, relative gap position (d/W) value of 0.16-0.5, relative gap width (g/e) values of 0.5-2.0, roughened duct width to height ratio (W/H) values of 5.83, angle of attack value of (α) 30^{0}- 90^{0}, relative roughness pitch (P/e) value of 4-10. The experimental investigation clearly demonstrates that maximum enhancement in Nusselt number (Nu) and friction factor (f) is observed to be 2.83 and 3.60 times of smooth duct, respectively.

Narinderpal singh Deo et al [4] have conducted an experiment study to investigate heat transfer, friction factor and thermo-hydraulic performance characteristics of flow in a rectangular duct used artificially roughened on one side with multi-gap V-down ribs combined with staggered ribs. The experiment encompassed by varying Reynolds number (Re) 4000-12,000, angle of attack (α) value of 40^{0}-80^{0}, gap width to rib height (g/e) ratio f 1. The rib to pitch height (P/e) values of 4-12 and rib height to hydraulic diameter (e/Dh) ratio from 0.026 to 0.057. The experiment investigation clearly demonstrates the maximum enhancement in Nusselt number (Nu) and thermohydraulic performance is observed to be 3.34 and 2.45 times respectively.

Rajesh Maithani et al [5] have conducted an experiment study for enhancement of heat transfer coefficient of solar air heater having roughened air heater having roughened air duct artificially roughened in the form of V-ribs with symmetrical gaps as turbulence promoter. The experiment encompassed by varying Reynolds number (Re) ranging from 4000 to 18,000, relative roughness pitch (P/e) value of 6-12, number of gaps (Ng) values of 1-5, relative roughness height (e/D) value of 0.043, angle of attack (α) value of 30^0-75^0. The investigation clearly demonstrates the maximum enhancement in Nusselt number (Nu) and friction factor (f) is observed to be 3.6 and 3.67 times of smooth duct, respectively.

Rajendra karwa [6] has conducted an experiment investigation of heat transfer and friction in rectangular duct and revealed the effect of inclined, V-continous, V-discrete and transverse pattern. The ribs in V-pattern are tested for both pointing downstream (V-down) and upstream (V-up) to the flow. The angle of inclination of the ribs in inclined and v-patterns is 60^0. The enhancement in the Stanton number over the smooth duct was up to 147%, 137%,134% and 142% for the V-down continuous, V-up continuous, V-up discrete and V-down discrete rib arrangement respectively. Performance comparison of the different rib patterns for equal pumping power shows that the V- down discrete arrangement gives the best heat transfer performance.

Arvind kumar et al [7] have conducted an experiment investigation of heat transfer and friction factor correlation and revealed the effect for artificially roughened solar air heater duct with discrete W-shaped ribs. The experiment encompassed by varying Reynolds number (Re) from 3000 to 15000, relative roughness height (e/Dh) in the range of 0.0168-0.0338, relative roughness pitch (p/e) 10. The investigation clearly demonstrates the maximum enhancement of Nusselt number was found 1.67 for angle of attack 60^0, for relative roughness height of 0.0168 and 2.16 found for relative roughness height 0.0338 for same angle of attack.

III. ARTIFICIAL ROUGHNESS

Amongst all available methods of enhancing turbulence in convective heat transfer, artificial roughness is believed to be an efficient method. Therefore in order to increase heat Transfer co-efficient, various designs flow arrangement presented at the heat transferring surface is to be made turbulent. However excessive turbulence leads to increase power requirement and such power is obtained from fan or blower to make the air flow through the duct. It is desirable that the turbulence must be created only in region very close to heat transferring surface i.e. laminar sub layer only. To minimise the friction losses, height of roughness element should be kept small in comparison with the duct dimensions. There are several physical parameters that characterize the arrangement and shape of the roughness over the absorber plates. Among all the parameters roughness element height (e) and roughness element pitch (P) are the most important parameters. These parameters are usually specified in terms of dimensionless parameters, i.e. relative roughness height (e/D) and relative roughness pitch (P/e). The roughness elements can be two dimensional transverse, angle or circular arc continuous or broken arc ribs. There are two generally two by which breaking of laminar sub layer is possible so that turbulence is created in the heat transfer zone and finally heat transfer co-efficient increases to a large extent. Out of these two methods the first method is for improving the heat transfer co-efficient by providing artificial roughness below the absorber plate so that turbulence is created in laminar sub layer. Therefore, artificial roughness increases heat transfer area which is essential condition for improving heat transfer co-efficient. The solar air heater is considered to be a rectangular channel with one rough surface and three smooth surfaces therefore roughness is provided only on the surface on which solar energy incident. The easiest method of providing artificial roughness on absorber plate is having ribs on it.

The other method is providing porous packing inside the duct it provides high heat transfer area, energy absorption in depth and finally turbulence is increased which is essential condition for increasing heat transfer co-efficient between the absorber plate and the flowing air in duct. This method is suitable for increasing the thermal performance of conventional solar heater. In the first method of providing artificial roughness on absorber plate is having ribs on the absorber plate. Ribs are various types like angle circular arc ribs, V-shaped ribs, wedge shaped ribs. Ribs no doubt increase in heat transfer co-efficient by increasing turbulence in laminar sub-layer but due to this increase in turbulence friction layer in duct will occur. Therefore more pumping power is required to overcome these friction losses so that air will propel in the duct smoothly and easily. it is found from many investigations that in solar air heater the Reynolds number ranges from 3000 to 15000. Therefore it becomes necessary that turbulence is created only in laminar sub-layer so that heat transfer increases with minimum

pumping power required. Therefore it becomes very necessary that hydraulic performance have been investigated for various roughness elements so that it gives an idea about optimum artificial roughness design which gives maximum heat transfer between the absorber plate and the air flowing in to the duct with minimum power requirement to force the air in to duct. There are several parameter which decide shape and arrangement of roughness element, first roughness element have height (e) and pitch (P). There is a dimensionless parameter (e/D) that is relative roughness height D is equivalent diameter of air passage and relative roughness (P/e) which decide the roughness arrangement.

3.1 Effect of roughness parameters:

The geometry of the artificial roughness can be of different shapes and orientations. Flow patterns were discussed with respect to different types of artificial roughness element below.

3.2 Effect of relative roughness height (e/Dh):

For all the values of Reynolds number investigated, the monotonic increase in Nusselt number and friction factor has been observed with increase in relative roughness height. This increase of Nusselt number and friction factor due to increase of relative roughness height may be because of rib sticking out more into core the flow thus increasing turbulence. Besides this, increase in relative roughness height cause more obstruction to main flow which enhances secondary flow.

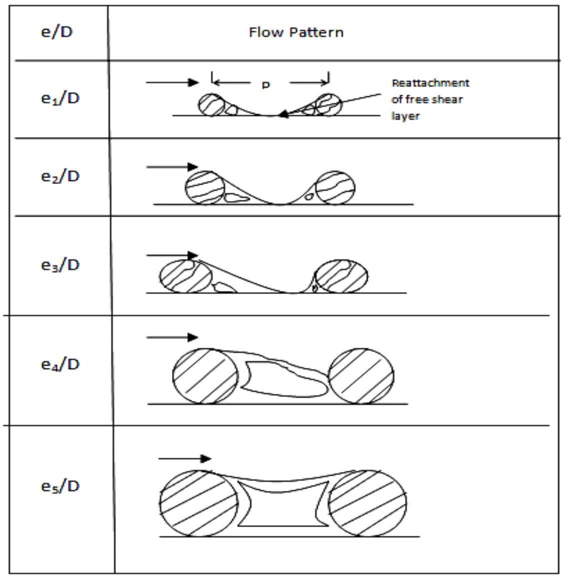


Figure.3.1 Effect of relative roughness height (e/Dh)

IV. EXPERIMENTAL DETAILS

4.1 Introduction: The experimental set-up and parts which is used in experimental set-up are discussed. Solar air heater with zig-zag flow pattern broken arc ribs is experimentally studied. The aim of this work is to study the heat transfer and friction characteristics using artificial roughness on underside of absorber plate of solar air heater. Artificial roughness is created on GI sheet which has been used as an absorber plate. The artificial roughness has been providing with the help of a spherical indenter in the form of protrusions arranged in angular fashion. The effect of roughness and operating parameter on heat transfer and frictio factor has been studied.

4.2 Experimental details

For testing absorber plates roughened with broken arc rib with zig-zag flow pattern, a outdoor test facility has been designed as per guidelines given in ASHRAE standard. The experimental schematic diagram set-up including the text 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 1500 mm x 150 mm x 100mm inner cross section. The experimental set-up is a rectangular channel with forced convection flow having entrance, test and exit sections. The components of experimental set up consists blower, plastic rectangular duct, GI sheet, U-tube manometer, micro-manometer, variable transformer, thermocouples and milli-voltmeter as shown in Figure. The duct is having Dimensions for inner cross-section as 1500 mm x 150 mm x 100 mm. The test section has a length of 1000mm and length of entry and exit section was provided as 200 mm and 300 mm respectively which are taken as per the ASHRAE standards. 15 mm thick thermocoal was used as an insulating material outside of the rectangular plastic duct to reduce the heat losses from absorber plate of the solar air heater. In total 5 thermocouples were provided over the test section for measuring the plate temperatures. All the thermocouples were connected to millivoltmeter through selector switch. The mass flow rate of air was measured by means of calibrated orifice meter connected with a U-tube manometer. Control valves were provided to control the flow. A digital millivoltmeter was used to measure the output of the thermocouples. The pressure drop across the test section was measured with the help of U-tube manometer having least count of 0.01 mm of water. Mean temperature of plate (Tpm) is calculated by weighed average method. It is based on the assumption that thermocouples area located at a point indicates average temperature of an of the absorber plate covered by that thermocouple.

4.3 Experimental set-up

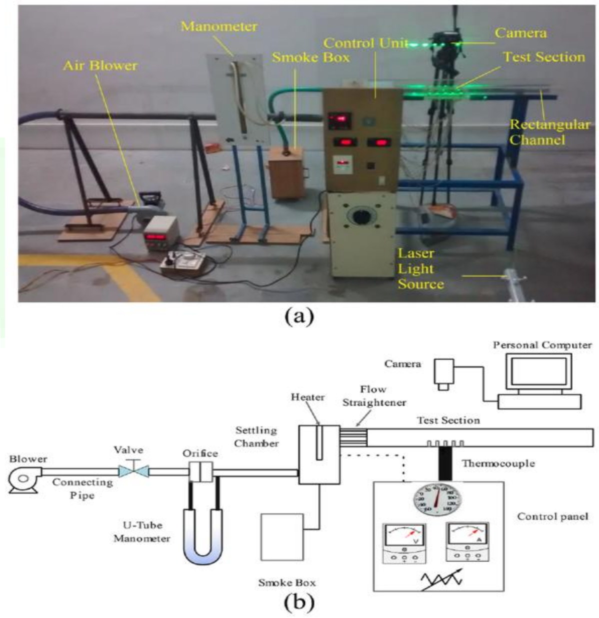


Figure 4.1: Pictorial view of experimental setup

For measuring temperature at outlet of test section, five thermocouples were arranged along width of the duct after the test section. Thermocouple was used for measuring temperature at entry of test section. To measure output of thermocouples in degree Celsius, a digital temperature indicator along with selector switch was used. Digital micro-manometer (least count=0.001Pa) was used to determine pressure drop across the test section.

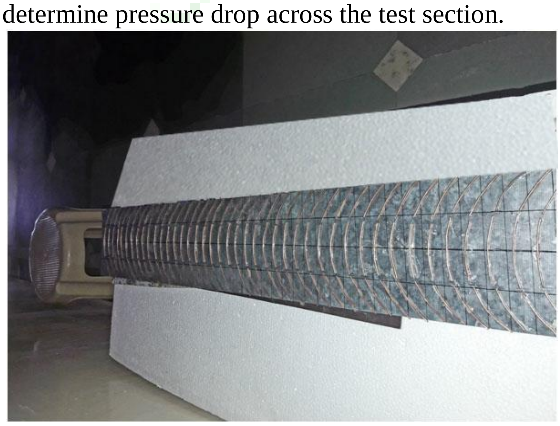


Figure 4.2 pictorial view of roughened absorber plate

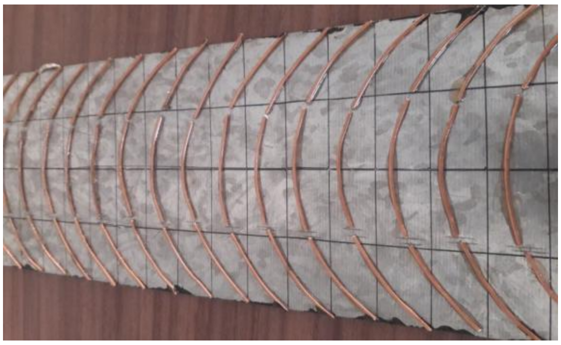


Figure 4.3 Roughened absorber plate

Figure 4.4: back view of absorber plate

4.4 component of experimental set-up:

Figure 4.5: Pictorial view of material use in rectangular duct

Figure 4.6: pictorial view of blowe

4.4.2 Variac

Figure 4.7: pictorial view of variac

4.4.3 Digital solar power meter

Figure 4.8: pictorial view of digital solar power meter

4.4.5 Digital Anemometer

Figure 3.9: pictorial view of digital anemometer

4.4.6 Milli-voltmeter

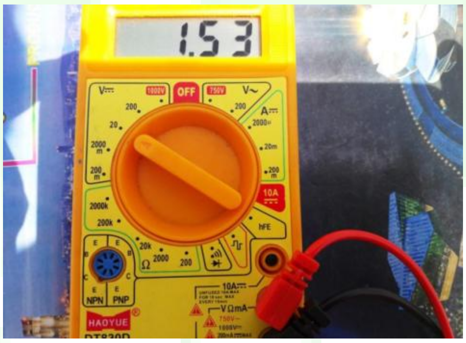


Figure 4.10: pictorial view of milli-voltmeter

3.4.7 U tube manometer

Figure 4.11: pictorial view of U-tube manometer

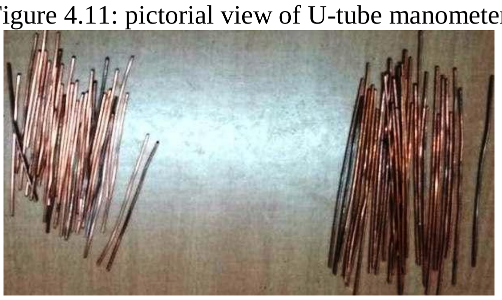
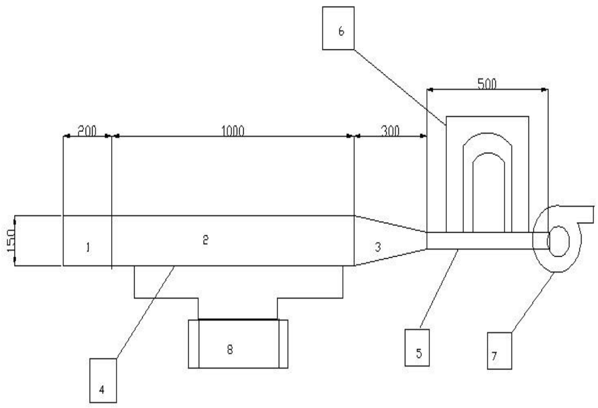


Figure4 .12: pictorial view of cut piece wire

4.5 Physical and numerical specifications of set-up

4.5.1 Experimental setup



1. Entry section
2. Test section
3. Exit section
4. Absorber plate
5. G.I.pipe
6. Inclind u-tube manometer
7. Electric air blower
8. Micromanometer
9. Control valve
10. Constant voltage transformer
11. Digital temperature indicator

4.5.2 Data used for experimental set-up

4.5.3 Experimental condition parameters

Table 4.1 operating parameter used for geometry

S. No	Parameter	Ranges
|1	Reynolds number (Re)	3500-15,000
2	Test length (l)	1000 mm
3	Hydraulic diameter	43 mm
4	Angle of attack (α)	60⁰
5	Area of plate	1000*150 mm²
6	Thickness of plate (t)	1 mm
7	Relative gap position (d/w)	0.65
8	Relative gap size (g/e)	1 mm
9	Relative roughness pitch (p/e)	10mm
10	Relative height ratio (e/D)	0.0233
11	Plate material and wire	G.I. sheet ,copper

4.6. Experimental procedure

For all experimental runs, initially all the instruments of experimental set up view checked for their proper functioning. To prevent against any air leakage, all joints were checked thoroughly for proper sealing. The experimental data on rib roughened duct was collected as per recommendations of ASHRAE standard []. Keeping similar flow conditions, a smooth duct was also investigated to check the usefulness of broken arc rib zig-zag flow pattern. For each test run the data such as inlet air temperature and pressure drop across orifice plate and test section, were recorded when system acquired a steady state. Each run was considered in steady state, when temperature did not show a change for a minimum of 20 min. when change in the operating condition is made, it takes 30-40 min. to reach quasi-steady state before the data were recorded. In order to reduce the effect of inaccuracy in the measurement of temperature, which strongly affects the accuracy of the calculation of heat transfer coefficient,
the temperature of the air through the duct has been maintained greater than 10 degree centigrade and the temperature difference between the heated plate and the bulk air temperature has been kept above 20^0C during the experimentation the temperature of the air at the entering the duct, range between 38 to 45^0C according to the local local atmospheric condition. The temperature of air at the outlet of test section is between 40^0C to 75^0C. 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 deviate over a 15 min. After the steady state has been reached, the voltage, current, the plate temperature, the inlet and exit temperature, the pressure drop across the duct and across the orifice plate have been recorded. For each plate configuration 5 runs have been conducted at different air flow rate corresponding to Reynolds number between is 3000 to 15000.

4.7. Experimental methodology

4.7.1 Air flow measurements:

The rate of flow of air through the duct was measured by means a pre calibrated orifice meter. It is used in circular pipe. An inclined tube manometer with water as a manometric fluid and a least count of 0.01 mm of water was used as a measurement of pressure drop across the orifice plate and this pressure drop is measured by inclined U tube manometer.

4.7.2. Pressure drop measurement in ducts

A U-tube manometer fig having a least count of 0.0025 mm was used for measurement of pressure drop across the test section. The micro manometer consists of a movable reservoir and an inclined transparent tube connected to the movable reservoir through flexible tubing. The reservoir is mounted on a sliding arrangement with screw having pitch of 1.5 mm and a graduated dial having 520 divisions each division showing a movement of 0.0025 mm of the reservoir.

4.8 Data Handling

The collected experimental data were analysed to determine the effect of influencing parameters on heat and friction factor in the flow.

4.8.1. Mass flow measurement

Mass flow rat of air has been determined from pressure drop measurement across the orifice plate using the following relationship

$\mathring{m} = \rho * A * V$

4.8.2. Heat transfer co-efficient

The heat transfer rate has been used to calculate the heat transfer coefficient (h) for test section

4.8.3. Mean air and plate temperatures

The mean air temperature or average flow temperature (Tfm) is simple arithmetic mean of the measure values at the inlet and exit of the test section.

4.9 Experimental result

4.9.1 Experimental values of performance parameters for smooth plate

Table 4.2 Experimental result of smooth plate

s. no.	Solar isolation I (w/m^2)	Reynolds number (Re)	Inlet temp Ti (^{0}c)	Avg. Outlet temp Toav (^{0}c)	Avg. Plate Temp T_{pm} (^{0}c)	Avg. Flow temp T_{fm} (^{0}c)	Nusselt number Nu	Friction factor (f)
1		3601	38.00	47.00	61.50	42.50	14.56	0.0183
2		5045	39.00	48.50	62.00	43.75	19.29	0.0163
3		7120	40.00	49.00	62.80	44.50	25.13	0.0145
4		9406	41.00	51.00	63.50	46.00	31.40	0.0129
5		12427	42.00	52.00	65.00	47.00	39.24	0.0112
6		14445	40.50	50.00	63.00	45.25	44.26	0.0098

4.9.2 Experimental values of performance parameters for roughened plate

Table 4.3 Experimental result of roughened plate

s. no.	Solar isolation I (w/m^2)	Reynolds number (Re)	Inlet temp Ti (^{0}c)	Avg. Outlet temp Toav (^{0}c)	Avg. Plate Temp T_{pm} (^{0}c)	Avg. Flow temp T_{fm} (^{0}c)	Nusselt number Nu	Friction factor (f)
1		3639	39.50	53.00	68.50	46.25	21.33	0.0307
2		5100	40.60	54.50	69.80	47.55	28.57	0.0265
3		7169	42.00	55.70	71.00	48.85	38.17	0.0234
4		9440	43.00	57.00	74.00	50.00	49.17	0.0119
5		12480	44.00	59.00	75.00	51.50	61.03	0.0171
6		14478	42.50	56.00	72.00	49.25	71.04	0.0162

V. RESULTS AND DISCUSSION

The effects of operating and roughness parameters on friction factor and Nusselt number for broken arc zig-zag flow pattern rib roughened duct have been discussed and reported in this section. For comparative evaluation of broken arc rib roughened plate duct, friction factor and Nusselt number for smooth plate duct have also been reported under similar condition. The Nusselt number for smooth rectangular duct is given by Dittus-Beolter equation as

$Nu = 0.0233Re^{0.8}Pr^{0.4}$

Figure 4.1: camparison of experimental and predicted values of Nusselt number smooth duct

Figure shows the variation of Nusselt number with respect to Reynolds number as the Reynolds number increases Nusselt number also increases for all combination of rib parameters.

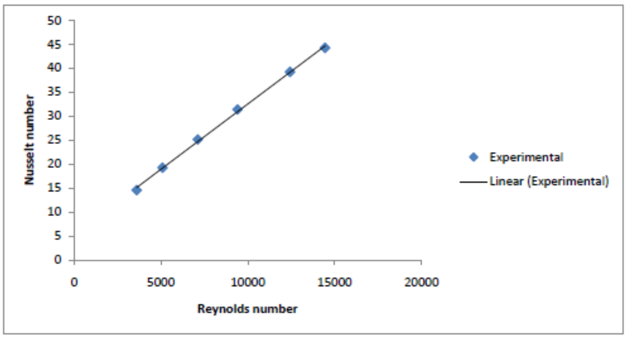


Figure 4.1: camparison of experimental and predicted values of Nusselt number smooth duct

Figure shows the variation of Nusselt number with respect to Reynolds number as the Reynolds number increases Nusselt number also increases for all combination of rib parameters.

The friction factor for a smooth rectangular duct is given by the Modified Blasius equation as

$f = 0.085Re^{-0.25}$

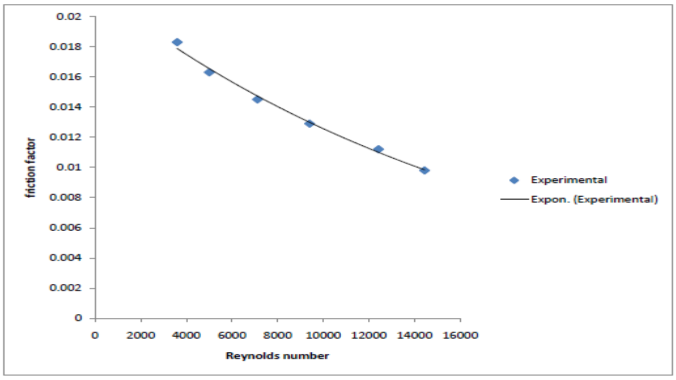


Figure 4.2 Comparison of experiment and predicted values of friction factor for smooth duct

Figure shows that with increase of Reynolds number, the friction factor decreases. This is due to viscous sub-layer suppression.

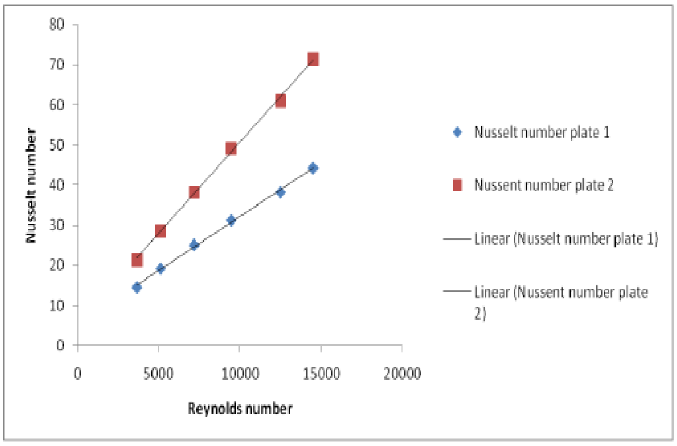


Figure 4.3 Variation of Nusselt with Reynolds number for broken arc zig zag pattern rib roughned duct and smooth duct

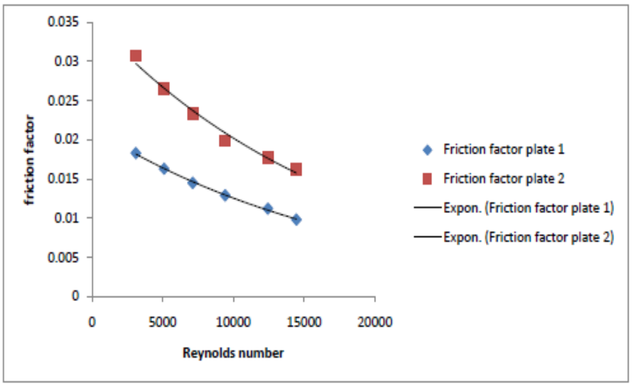


Figure 4.4 Variation of friction factor with Reynolds number for broken arc zig zag pattern rib roughened duct and smooth duct.

VI.CONCLUSION AND FUTURE SCOPE

The On the basis of experimental investigation of heat transfer, Nusselt number, friction factor and thermo-hydraulic performance of solar air heater duct provided with heated plate having broken arc with zigzag pattern ribs geometry of artificial roughness, the following conclusions can be drawn from the present work:

- For duct roughened with broken arc ribs with zigzag flow pattern, the maximum enhancement in Nusselt number and friction factor over that smooth duct have been found to be 1.61 and 1.67 times, respectively.
- The increase in Reynolds number, Nusselt number increases for all combination of rib parameter. It is also observed that as Reynolds number increases, increase in Nusselt number is more for broken arc ribs with zigzag flow pattern in comparison to continuous arc rib.
- The value of friction factor decreases with increase of Reynolds number for different roughness geometry that means loss of pressure drop decreases.
- Roughened absorber plate increases the heat transfer coefficient as compared to smooth rectangular duct under similar operating conditions at higher Reynolds number.
- Heat transfer enhancement in case of broken arc ribs with zigzag flow pattern is
- Less as compared to V-rib with gap and inclined rib with gap but due to lesser frictional losses, thermo hydraulic performance of broken arc with zigzag flow pattern rib geometry is better for the range of Reynolds number which is considered appropriate for solar air heater operation.
- Solar air heaters with roughened absorbers perform better as compared to smooth heaters. Broken arc ribs with zig-zag flow pattern have significant effect on heat transfer.

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