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Experimental Analysis of solar air heater duct provided with circular chips as artificial roughness elements

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ABSTRACT: Second law analysis shows that solar energy can be efficiently used for process heating in industries, drying agriculture products etc. The poor thermal efficiency of solar air heaters is generally due to the low heat transfer coefficient between absorber plate and fluid flowing through the duct which is due to existence of thin boundary layer known as viscous sub-layer which makes the insulate situation between heated surface and heat convicting fluid. In the last few years, there has been a growing interest in using roughness underside of the absorber plate which is a very effective method for improving the heat transfer coefficient. By using roughness causes to break viscous sub-layer results into the high thermal performance of the solar air heaters. However, the major drawback of this approach is accompanied by higher friction factor that causes excessive centrifugal fan power for propelling the forced air through the duct. In view a designer needs to carefully examine shape and orientation of roughness elements in order to choose the best fit roughness geometry for intended application. Thus experimental study has been carried out for roughened duct provided with circular chips as roughness geometry. The investigation has covered the range of relative roughness height (e/Dh) from 0.0185 to 0.0366, relative roughness pitch (p/e) from 10 to 40 and relative print diameter (d/Dh) from 0.146to 0.07334. Based on the experimental data, values of Nusselt number (Nu) and friction factor (fr) have been determined for different values of roughness and operating parameters. In order to determine the enhancement in heat transfer and increment in friction factor values of Nusselt number and friction factor have been compared with those of smooth duct under similar flow conditions. Correlations for Nusselt number and friction factor have been developed for solar air heater duct provided such artificial roughness geometry.

KEYWORDS: Friction factor, Nusselt number, Relative roughness height, Relative roughness pitch, Relative print diameter.

I. INTRODUCTION

A conventional solar air heater is essentially a flat-plate collector with a blackened absorber plate, transparent cover at the top, and insulation at the bottom and on the sides. The whole assembly is encased in a sheet-metal container, and the system is referred to as a "solar flat-plate air collector."The working fluid is air which is blown either below or above the absorber plate. The thermal efficiency of solar air heaters has been found to be generally poor because of their inherently low heat transfer capability between the absorber plate and air flowing in the duct due to the smooth surface of duct. This low value of heat transfer coefficient is generally attributed due to the presence of viscous sub layer. In order to make the solar air heaters economically viable, their thermal efficiency needs to be improved by enhancing the heat transfer coefficient. In order to attained higher heat transfer coefficient, the laminar sub-layer formed the vicinity of the absorber plate is to be broken and the flow at the heat-transferring surface is made to be turbulent. This can be done by providing modifications near the heat transferring surface such as artificial roughness on the surface of absorber plate which helps in creating turbulence and thereby increasing the rate of heat transfer. Usually extended surfaces like fins are used in the solar air heaters. Various attempts has been made to enhance the heat transfer rate Prasad and Saini [3,4] reported an experimental investigation of fully developed turbulent flow in a solar



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017

air heater duct having small diameter protrusion wires fixed on absorber plate. Nusselt number and friction factor correlations were developed by using experimental data. An enhancement in Nusselt number and friction factor was observed over smooth duct of the order of 2.38 and 4.25 times respectively corresponding to relative roughness height of 0.033 and relative roughness pitch of 10. Gupta et al. [8] established optimum design parameters under actual climatic conditions for roughened solar air heaters for varying relative roughness height (e/D) and for a relative roughness pitch (P/e) of 10 at an angle of attack (a) of 60 deg. Geometry of roughened absorber plate is inclined ribs An enhancement of heat transfer and friction factor was obtained of the order of 1.8 and 2.7 times respectively. Maximum heat transfer coefficient and friction factor values were obtained at an angle of attack of 60 deg respectively in the range of investigated parameters. Momin et al. [11] investigated solar air heater having V-shaped ribbed roughness on the absorber plate. This experimental investigation covered relative roughness height (e/D) of 0.02-0.034 and angle of attack (α) of 30–90 deg for a fixed relative roughness pitch (P/e) of 10. It was reported that V-shape ribs with an angle of attack (α) of 60 deg enhanced Nusselt number by 1.14 and 2.30 times and friction factor by 2.30 and 2.83 times over inclined ribs and smooth plate respectively. Karmare and Tikekar [14] investigated artificially roughened duct with metal grit ribs. It is reported that plate having roughness parameters (e/D) = 0.044 and (P/e) = 17.5resulted optimum performance and as compared to smooth duct. Aharwal et al. [15] investigated effect of artificial roughness by using an inclined non-continuous rib arrangement in a rectangular duct .Maximum enhancement in Nusselt number and friction factor as compared to smooth duct was observed to be2.59 and 2.87 times respectively.

Saini and Saini [17] investigated effect of arc shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of solar air heater. This experimental investigation covered a relative roughness height (e/D) of 0.0213–0.0422 and relative angle of attack of flow ($\alpha/90$) of 0.3333–0.6666 for a fixed relative roughness pitch (P/e) of 10. Maximum enhancement in Nusselt number and friction factors compared to smooth duct was observed to be 3.6 and 1.75 times respectively. Bhagoria et al. [21] investigated air heater rectangular duct roughened by wedge shaped transverse integral ribs. Range of relative roughness height (e/D), relative roughness pitch (P/e) and rib wedge angle (f) was 0.015-0.033, $60.17f_{1.0264} < p/e < 12.12$ and 8-15 degres pectively. Authors reported an enhancement in Nusselt number and friction factor of the order of 2.4 and 5.3 times respectively as compared to smooth duct. Jaurker et al. [29] reported an experimental investigation on rectangular duct having repeated integral transverse rib groove. Range of relative roughness height (e/D), relative roughness pitch (P/e) and groove position to pitch ratio (g/P) was 0.0181-0.0363, 4.5-10.0 and 0.3-0.7 respectively. Enhancement of Nusselt number of the order of 2.75 times of the smooth duct and 1.57 times of ribbed duct with similar rib height and rib spacing was observed. Whereas ribbed duct with similar rib height and rib spacing provides Nusselt number values of the order of 1.7 times that of smooth duct and friction factor increases in the order of 3.61 times that of smooth duct and 1.17 times that of ribbed duct. Whereas a ribbed duct with similar rib height and rib spacing results in friction factor value of the order of 3 times that of the smooth duct. Layek et al. [23] investigated repeated integral transverse chamfered rib-groove roughness, relative roughness pitch of 4.5–10, chamfer angle of 5–30 deg, relative groove position of 0.3–0.6 and relative roughness height of 0.022-0.04. Authors reported that Nusselt number and friction factor increased by 3.24 times and 3.78 times respectively as compare to smooth duct. Saini and Verma [30] investigated dimple shaped artificial roughness geometry Range of relative roughness height (e/D) and relative pitch (p/e) was 0.018–0.037 14 and 8–12 respectively. Authors reported that Nusselt number and friction factor increased by 1.8 and 1.4 times respectively as compared to smooth duct. Bhushan et al [28] investigated staggered dimple roughness in place of transverse dimple roughness. Range of parameter investigated were relative short way length (S/e) as 18.75-37.50, relative long way length (L/e) as 25.00-37.50, relative print diameter(d/D) as 0.147-0.367, relative roughness height as 0.03, aspect ratio as 10 Under given condition maximum enhancement of Nusselt number and friction factor was 3.8 and 2.2 times respectively in comparison to smooth duct. Maximum enhancement in heat transfer coefficient was reported for relative short way length (S/e) of 31.25, relative long way length (L/e) of 31.25 and relative print diameter (d/D) of 0.294.



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u>

Vol. 6, Issue 7, July 2017

List of Abbreviation		
Ap	Area of Plate	
CP	Specific heat of Air	
d/Dh	Relative Print Diameter	
d/W	Relative Gap Position	
Dh	Hydraulic Diameter of Duct	
e/d	Ratio Of Dimple Depth To Diameter	
e/Dh	Relative Roughness Height	
e/H	Rib To Channel Height Ratio	
f_r	Friction Factor Of Rough Duct	
f_s	Friction Factor Of Smooth Duct	
g/e	Relative Gap	
g/p	Relative Groove Position	
H	Height of Duct	
L	Length of Duct	
L/e	Relative Long Way Length	
ṁ	Mass Flow Rate of Air	
Nu _R	Nusselt Number Rough Duct	
Nus	Nusselt Number Smooth Duct	
P/e	Relative Roughness Pitch	
P/H	Baffles Pitch Ratio	
Q_a	Heat Gained by Air	
r/e	Relative Staggered Rib Size	
Re	Reynolds Number	
S/e	Relative Short Way Length	
s'/s	Relative Roughness Segment Ratio	
T_{F}	Mean Fluid temperature	
Ti	Temperature of Inlet Air	
THP	Thermo-Hydraulic performance	
To	Temperature of Outlet Air	
T_P	Temperature of Plate	
V	Velocity of Air	
W	Width of Duct	
W/H	Aspect Ratio	
W/w	Roughness Width Ratio	
Δh_0	Pressure Drop Across Orifice Plate	
Δh_T	Pressure Drop Across Test Section	
ΔΡο	Pressure Drop Across Orifice Plate	
$\Delta P_{\rm T}$ Pressure Drop Across Test Section		
Greel	c symbol	
α	Angle Of Attack	
φ	Chamter Angle	
β	Open Area Ratio	
ρ	Density of Air	
θ	Kinematic Viscosity of Air	

The present investigation was, therefore, carried out with the objective of extensive experimentation on circular chip geometry as artificial roughness provided on to the underside of one wall of the duct of a solar air heater, in order to collect data on heat transfer and fluid flow characteristics. The experimental data are presented in the form of Nusselt



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u>

Vol. 6, Issue 7, July 2017

number and friction factor plots as function of Reynolds number for different roughness parameters. Using experimental data collected during extensive experimental study, correlations for Nusselt number and friction factor have also been developed.

II. INDOOR EXPERIMENTAL PROGRAM

2.1 Experimental setup

As in case of actual solar air heater it is not possible to perform experimentation on all absorber plate at a time because intensity of solar irradiation is changing time to time so that indoor experimental test facility has been designed and fabricated to study the effect of different roughness pitch, height and diameter on the heat transfer characteristics of a solar air heater duct. In experimental setup facility has been provided to give average global solar irradiation. A schematic diagram of experimental setup is shown in Figure 2.1



The experimental setup is as per the ASHRAE standard 93-77. It recommends that entry section should be $\geq 2.5\sqrt{WH}$ and exit section should be $\geq 2.5\sqrt{WH}$. The solar air heater duct is made up of wooden panels of 18mm thickness. The inner surfaces of the duct are coated by smooth sun-mica so as get desirable results. The length of entry section is kept as 500 mm so as to get fully developed flow of air. The length of exit section is kept as 300 mm so as to reduce the end effects such as back flow etc. The top section of duct is insulated with the help of asbestos sheet and glass wool so as to reduce heat loss from the top wall. The adjustable door is fixed to the duct which acts as a top wall of air duct. After the exit section of duct there is a plenum section to collect the air from the duct. The detail specification is given in Table 2.1.

2.1 Table: Specification of air duct.			
Parameter	Dimensions		
Cross-section of duct (W*H)	$300 \text{ mm} \times 30 \text{ mm}$		
Length of entry section	500 mm recommended $\geq 5\sqrt{(WH)}$		
Length of test section L	1000 mm		
Length of exit section	300 mm recommended $\geq 2.5\sqrt{(WH)}$		
Length of plenum	300 mm		
Aspect ratio	W/H= 300/30 =10		
Hydraulic diameter	2WH/(W+H) = 54.54 mm		



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017



Figure 2.1.2The pictorial view of the experimental setup

2.2 Materials and equipments.

- 1. A centrifugal blower is fitted with an electric motor to induce air from the duct the capacity of the blower is 70 CFM, 1 H.P, Volts -180/230, Single phase with Rpm 2800 .The inlet of the blower is fitted to the exit of the air duct and at the exit of blower the control valve is fitted to control the velocity of air flow.
- 2. Orifice Meter to measure the flow rate, water tube manometer range: 0 to 50 mm of water, Micro manometer, an electric plate heater having a size of 1000 mm × 300 mm by combining a series and parallel loops of Nichrome wire having Combined Capacity up to1000 Watts.
- 3. Temperature measurement unit in conjunction with 12 K-type Thermocouples at equal distance on heating Section wall and 5 for measuring Inlet and Outlet temperature of air.
- 4. Control Panel consist of Ammeter, Voltmeter and Dimmerstat6 Ampere Variac and 2500 Watt.

2.3 Preparation of absorber plate

Absorber plate is made up of GI sheet of 1.30 mm thickness. The length of absorber plate is 1000 mm and width is 300 mm. Roughness elements in the form of flat circular chips are prepared with turret machine. The circular chips are of diameter 4mm, 6mm and 8 mm in diameter and 1 mm, 1.5 mm and 2mm of thickness with angle of attack 90°. Roughness is created on the lower surface of absorber plate by adhering chips at different pitch value with the help of Araldite adhesive solution. The area covered by this chips on absorber plate is considered as an adiabatic as no heat is transferred through this area. Top side of this plate is exposed to the heater plate. In total nine roughened plates are fabricated and one plate is kept as smooth so as to compares results obtained. Figure (a), (b) and (c) shows the pictorial view of Roughness elements and absorber plate receptively.





(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017



Figure 2.3 :(a) Different roughness elements used. (b) Photographic view of absorber plate roughnesd by circular chips. (c) Arrangement of roughness elements on absorber plate

2.4 Experimental procedure

For each experimental run, initially all the instruments viz., mille-voltmeter, micro-manometer- tube manometer were checked. All the joints were checked to ensure no leakage from the system. Also the correctness of the measuring instruments is performed before starting experimentation. The system is allowed to reach the quasi-static state before the data recorded. During the experimentation, the following parameters need to measure for each set of the flow rate:

1) Pressure drop across the test section (ΔPT).

2) Pressure drop across orifice plate (ΔPO).

3) Temperature of absorber plate (Tp).

4) Temperature of air at inlet (Ti) and outlet (To) of the test section.

Steady state values of the test plate and air temperatures at various locations were obtained for a given heat flux and the mass flow rate of air. Heat transfer rate to the air, Nusselt number, friction factor have been computed from the data roughened as well as smooth plate.

2.5 Data Reduction

Steady-state values of the plate and air temperature in the duct at various locations were used to determine the values of useful parameters. Mass flow rate, m, velocity of air v, heat supplied to the air Q and heat transfer coefficient h, where calculated by using

Following expressions

Pressure difference $\Delta Po = \rho * g * \Delta h$ Mass flow rate $m = C_d * A_o * \left(\frac{2\rho\Delta P_O}{1-\beta^4}\right)^{0.5}$ Velocity of Air $v = \frac{m}{\rho WH}$ Hydraulic diameter $D_h = \frac{4WH}{2(W+H)}$ Reynolds Number $Re = \frac{VD_h}{\vartheta}$ Heat gained by air $Q_a = m * C_p * \left(T_{o_{avg}} - T_{i_{avg}}\right)$ Heat transfer coefficient $h = \frac{Q_a}{A_p * (T_{Pavg} - T_{Favg})}$

Where the temperature $T_{P_{avg}}$ and $T_{F_{avg}}$ are average values of the absorber plate and fluid (air) temperature, respectively the average value of plate temperature $T_{P_{avg}}$ was determined from the detailed temperature profile of the absorbing plate. It was found that the temperature of the absorber plate varies predominantly in the flow direction



(An ISO 3297: 2007 Certified Organization) Website: www.ijirset.com Vol. 6, Issue 7, July 2017

only and is linear. The variations in the direction normal to the flow were found to be negligible. The plate temperature was determined at different locations. The average plate temperature Tp_{avg} was determined by taking average temperature of all thermocouples located at different locations on plate the air temperature variations were found to be linear. Since the temperature variations of the absorber plate in the direction normal to the flow were found to be negligible in that direction, the air temperature variations in that direction can be assumed to be negligible. The temperature variation in the direction normal to the absorber plate could not be measured because of very small duct depth and were assumed to be negligible. The air temperature T_f was determined as an average of the temperatures measured at few location of the duct cross-section along the flow direction at in late and out late. The convective heat transfer coefficient was then used to obtain the average Nusselt number from the following expression:

Nusselt Number $Nu = \frac{hD_h}{k}$ Friction factor $f = \frac{2\Delta P_T D_h}{4\rho LV^2}$

The friction factor was determined from the measured values of pressure drop ΔP_T across the test section length, L. The validity of the test setup was verified by conducting experiments for smooth duct under similar conditions. The values of Nusselt number and friction factor determined for smooth duct were compared with the values obtained from correlations of Dittus-Boetler equation for the Nusselt number and modified Blasius equation for the friction factor. Nusselt number for a smooth rectangular duct is given by Dittus-Boetler equation.

 $Nu = 0.023 Re^{0.8} Pr^{0.4}$

Friction factor for a smooth rectangular duct is given by Blasius equation as $Fr = 0.085 Re^{-0.25}$

III. RESULTS AND DISCUSSIONS

The effect of flow and roughness parameters on heat transfer and friction characteristics for flow of air in a rectangular duct under the present investigation are presented and discussed under this part of the paper. In order to conduct the validity test of the duct, experimental data were collected on duct having smooth absorbing surface. The average values of Nusselt number and friction factor for the duct having smooth plate were determined. Fig. 3.1 (a) and (b) show the comparison of the experimental values of Nusselt and friction factor for roughened solar air heater with those predicted by correlations proposed by Modified Dittus- Boelter correlation for Nusselt number and by Modified-Blasius correlation for friction factor. It can be seen from these figures that the values obtained experimentally are comparable.



Fig.3.1 (a) Comparison of experimental and theoretical values of Nusselt number for smooth duct. (b) Comparison of experimental and theoretical values of friction factor for smooth duct.



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Figs. 3.2 and 3.3have been drawn to represent the Nusselt number derived from the test results as a function of the roughness and operating parameters. Fig. 10 shows the Nusselt number as a function of Reynolds number for relative roughness height (e/D_h) values of 0.0185, 0.0275 and 0.0366 and for a fixed value of relative print diameter (d/D_h) of 0.146. The plot is prepared to compare the heat transfer rate for a duct having circular chip shape geometry of different values of relative roughness height (e/D_h) . Similar trend has been obtained for Nusselt number for all the values of relative print diameter. It can be seen from the figures that for given values of roughness parameters, the Nusselt number monotonously increases with increase of Reynolds number. It is seen from Fig. 3.2that Nusselt number has been found maximum corresponding to relative roughness height, (e/D_h) of 0.0366. Due to increasing in relative roughness height volume of separation of the flow by the roughness element increases which result in to higher heat transfer coefficient occurs in the vicinity of the reattachment region. For relative roughness height (e/D_h) of 0.0185 and 0.0275 less volume separations occur hence the values of heat transfer coefficient correspond to these values of (e/D_h) are found to be minimum.



Fig 3.2 Variation of Nusselt number with Reynolds number and for different values of Relative roughness height (e/Dh) for fixed value of relative print diameter (d/Dh) =0.146

Similarly, larger relative print diameter (d/D_h) might produced more reattach reattachment of laminar sub layer hence the maximum value of heat transfer is found to be corresponding to relative print diameter (d/D_h) of 0.146. Fig. 3.3shows the Nusselt number as a function of Reynolds number for relative print diameter (d/D_h) values of 0.146, 0.1101, and 0.07334 and for a fixed value of relative roughness height (e/D_h) of 0.0366. It is seen that Nusselt number increases withincrease in relative print diameter, (d/D_h) and has been found maximum corresponding to relative print diameter of 0.146.



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017



Fig.3.3 Variation of Nusselt number with Reynolds number and for different values of Relative print diameter (d/Dh) for fixed values of Relative roughness height (e/Dh) 0.0366.

Figs. 3.4 and 3.5show the variation in friction factor with Reynolds number for different values of roughness parameters. It is seen from the figures that friction factor decreases monotonously with increase in Reynolds number for different roughness parameters. Fig. 3.4shows the friction factor as a function of Reynolds number for relative roughness height (e/D_h) values of 0.0185, 0.0275 and 0.0366 and for a fixed value of relative print diameter (d/D_h) of 0.0366. The plot is prepared to compare the friction factor for a duct having circular chip shape geometry of different values of relative roughness height (e/D_h). Similar trend has been obtained for friction factor for all the values of relative print diameter. Friction factor has been found minimum for relative roughness height (e/D_h) corresponding to 0.0185. Similarly Fig.3.5shows the variation of friction factor for relative print diameter (d/D_h) of 0.0366. It is seen that friction factor corresponding to relative print diameter, (d/D_h)of 0.07334 has been found maximum while friction factor for (d/D_h)of 0.146 is found to be minimum. As relative print diameter decreases number of attachment and detachment goes on increasing which results into higher frication factor.



Fig.3.4 Variation of Friction factor with Reynolds number and for different values of Relative roughness height (e/Dh) for fixed value of relative print diameter (d/Dh) 0.146.



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017



Fig.3.5 Variation of frication factor with Reynolds number and for different values of Relative print diameter (d/Dh) for fixed values of Relative roughness height (e/Dh) 0.0366.

Correlations for Nusselt number and friction factor

As discussed earlier that the Nusselt number and friction factor are strong functions of flow and roughness parameters, namely flow Reynolds number (Re) and the roughness dimensions relative print diameter (d/D_h), relative pitch (p/e) and relative roughness height (e/D_h). The functional relationships for Nusselt number and friction factor can therefore be written as

$$\begin{split} Nu_r &= f\left((\text{Re}), (p/e), (e/\text{Dh}), (d/\text{D}_h)\right) \\ FF_r &= f\left((\text{Re}), (p/e), (e/\text{Dh}), (d/\text{Dh})\right) \\ \text{All data points of Nusselt number from experimental data were plotted against Reynolds number as shown in Figure. A regression analysis to fit a straight line through these data points may be represented as \\ ln (Nur) &= 0.9283 \times ln(\text{Re}) - 4.7601 \\ Nur &= 0.0085647 \text{Re}^{0.9283} \end{split}$$



The 0.008564 will in fact be a function of other influencing parameters. Now, taking the parameter roughness pitch (p/e) into consideration, the value of $\ln(\text{Nur/Re}^{0.92830})$ corresponding to all values of $\ln(p/e)$ is plotted against $\ln(p/e)$, as shown in Figure. Regression analysis to fit a straight line through these data points may be represented as



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017

 $\begin{array}{l} ln \; (Nur/Re^{0.92228} = -0.1074 ln(p/e) - 4.388 \\ Nur = 0.01242*(Re^{0.92228})*(p/e)^{-0.1074} \end{array}$



Further, the constant (0.01242) will be the function of relative roughness height (e/Dh)Now, taking the parameter roughness height (e/Dh) into consideration, the value of $Ln(Nur/((Re^{0.9283})(p/e^{1-0.1092})))$ corresponding to all values of ln(e/Dh) is plotted against ln(r/Dh), as shown in Figure. Regression analysis to fit a straight line through these data points may be represented as

 $\begin{array}{l} Ln(Nur)/(Re^{0.9283}(p/e)^{-0.1092})) = 0.2785 \times ln(e/Dh) - 3.3829 \\ Nur = 0.03271 Re^{0.92228} \times (p/e)^{-0.1074} \times (e/Dh)^{-0.276} \end{array}$



Further, the constant (0.03271) will be the function of relative print diameter (d/Dh) Now, taking the parameter relative print diameter (d/Dh) into consideration, the value of $Ln((Nur)/((Re^{0.9283})*((p/e)^{0.1092})\times((e/Dh)^{0.2785})))$ corresponding to all values of ln(d/Dh) is plotted against ln(d/Dh), as shown in Figure. Regression analysis to fit a straight line through these data points may be represented as



(An ISO 3297: 2007 Certified Organization) Website: <u>www.ijirset.com</u> Vol. 6, Issue 7, July 2017



 $Ln((Nur)/((Re^{0.9283})\times((p/e)^{-0.1092})\times((e/Dh)^{0.2785})))=0.275ln(d/Dh)-2.8038$

 $(Nu_r)_{cr} = 0.059427 \text{Re}^{0.9283} \times (p/e)^{-0.1092} \times (e/Dh)^{0.2785} \times (d/Dh)^{0.275}$

A similar procedure has been employed to develop correlation for friction factor . FF_r=f ((Re), (p/e), (e/Dh), (d/Dh))

The final form of correlation for friction factor is obtained as follows

 $(F_r)_{cr} = EXP(9..891) \times (Re^{-1.3879}) \times ((p/e)^{-0.4642}) \times ((e/Dh)^{0.00004}) \times (d/Dh)^{-0.0701}$

IV. CONCLUSIONS

An experimental study has been carried out to predict the heat transfer and flow characteristics of a solar air heaters having roughened duct provided circular chips shape roughness geometry for artificial roughness. It can be concluded that heat transfer can be enhanced considerably as a result of providing circular chip shape roughness geometry on the absorber plate of a solar air heater duct. Nusselt number and friction factor are the strong function of the system and operating parameters. The maximum value of Nusselt number has been found corresponds to relative roughness height (e/D_h) of 0.0366 and relative print diameter (d/D_h) of 0.146. While minimum value of friction factor has been found correspond to relative roughness height (e/D_h) of 0.0185 and relative print diameter (d/D_h) of 0.146. It is therefore, roughness parameters of the geometry can be selected by considering the net heat gain and corresponding power required to propel air through the duct. Further different arrangement mode of the circular chip shape artificial roughness. Based on experimental data collected during investigation, correlations for Nusselt number and friction factor have been developed for the investigated system. These correlations developed for such systems have been found to predict the values of Nusselt number and friction factor reasonably well under the given range of parameters.

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Vol. 6, Issue 7, July 2017

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