Statistical Analysis of Heat Transfer and Fluid Flow Features of Solar Air Heater

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Abstract: **The application of artificial roughness in the form of fine wires or ribs of different geometry on the heat transfer surface has been recommended to increase the heat transfer coefficient by several investigators. The use of artificial roughness on the underside of the absorber plate disturbs the viscous sub-layer of the flowing medium. It is well known that in a turbulent flow a sub-layer exists in the flow in addition to the turbulent core. The purpose of the artificial roughness is to make the flow turbulent adjacent to the wall in the sub-layer region. The present work was undertaken with the objectives of extensive investigation on the 90° shaped ribs as artificial roughness on the surface of duct. Experimental setup for heat transfer and pressure loss has been design and developed. Data were collected for local heat transfer and pressure loss of these artificially roughened ducts. Results of artificially roughened duct have been compared with those of a smooth duct under similar flow condition to determine heat transfer and friction factor.**

Keywords- **Heat Transfer, Friction Factor. Roughness.**

I. INTRODUCTION

The aim to reach higher and higher efficiency levels for turbo gas engines, so as to reduce energy resources consumption and global emissions, pushed towards a gradual increase of maximum cycle temperature. At the same time, a constant search for new materials and casting techniques allowing to increases metal temperature, as well as the development and improvement of heat transfer is always more effective though complex cooling configurations is being carried out to ensure the required system reliability. Among various cooling techniques, turbulence promoters (ribs) are probably one of the most effective techniques, thanks to their ability to combine high heat transfer coefficients (mainly due to the increase of near wall flow turbulence level) to limited pressure losses.

The transfer or removal of excessive heat from system components is essential to avoid the damaging effects because of burning or overheating. Therefore, the enhancement of heat transfer is an important subject of thermal engineering. The heat transfer from surfaces may in general be enhanced by increasing the heat transfer coefficient between a surface and its surroundings, by increasing the heat transfer area of the surface, or by both. In most cases the heat transfer is concerned with the application of vibration and simultaneously with artificial surface roughness techniques as a combined turbulence promoter for convective heat transfer enhancement.

1.1) Techniques for Enhancement of Heat Transfer: There are two basic techniques of enhancement of heat transfer, first one is Active technique and second is Passive technique.

1.1.1) Active techniques:
In active techniques there is requirement of direct input of additional power to change the flow structure of fluid. Fluid injection from transverse direction of flowing fluid and create vibrations on fluid flowing surfaces.

1.1.2) Passive technique:
Do not require direct input of external power to change the flow structure of fluid. For example artificial roughened surfaces by providing ribs on the surfaces.

1.2) Improvement of Heat Transfer Coefficient between Duct and Air:

A few techniques used for improvement of heat transfer from surface are discussed below:

1.2.1) Use of Artificial Roughness on Surfaces of Duct:
The application of artificial roughness in the form of fine wires or ribs of different geometry on the heat transfer surface has been recommended to increase the heat transfer coefficient by several investigators. The use of artificial roughness on the underside of the absorber plate disturbs the viscous sub-layer of the flowing medium. It is well known that in a turbulent flow a sub-layer exists in the flow in addition to the turbulent core. The purpose of the artificial roughness is to make the flow turbulent adjacent to the wall in the sub-layer region. Artificial roughness can be produced by machining, casting, forming, welding and other methods.

II. LITERATURE REVIEWS

Aharwal et. al. [1] carried out experiments to study the effect of a gap in the inclined rib on the heat transfer and fluid flow characteristics of heated surface. They used a rectangular duct of aspect ratio of 5.83 to conduct experiments on one rib roughened surface. Experimental data have been collected to determine Nusselt number (heat transfer coefficient) as a function of roughness and flow parameters in the form of repeated ribs. In order to understand the mechanism of heat transfer through a roughened duct having inclined rib with and without gap, the detailed analysis of the fluid flow structure is required. Therefore the detailed velocity structures of fluid flow inside a similar roughened duct as used for the heat transfer analysis were obtained by 2-Dimensional Particle Image Velocimetry (PIV) system and the heat transfer results were correlated with the flow structure. It was found that inclined rib with a gap (inclined discrete rib) had better heat transfer performance compared to the continuous inclined rib arrangement. Further the inclined discrete rib with relative gap width (g/e) of 1.0 gives the higher heat transfer performance compared to the other relative gap width.

They conclude that in the present investigation study the effect of gap width on the heat transfer coefficient has been investigated and attempt has also been made to determine the optimum width of a gap in the inclined rib to form discrete rib for the better heat transfer performance. Investigations have been carried out for the relative gap width of 0.5 to 2.0 and the Reynolds number range of 3000-18000 at relative roughness height of 0.037. It was found that inclined rib with relative gap width (g/e) of 1.0 gives the highest value of heat transfer coefficient compared to the other relative gap width. To investigate the effect of gap width on the flow field, a Two Dimensional Particle Image Velocimetry (2-D PIV) system is used. The level turbulence intensity at different relative gap width has been measured. It is observed that the highest value of turbulent intensity is observed at the relative gap width of 1.0 which is in the line of heat transfer measurements. These results of PIV showed good agreement between the heat transfer results and the flow analysis results.

Saini and Saini [4] investigate solar air heater having artificial roughness in the form of arc- shape parallel wire. The effect of system parameters such as relative roughness height (e/d) and arc angle (a/90) have been studied on Nusselt number (Nu) and friction factor (f) with Reynolds number (Re) varied from 2000 to 17000. The maximum enhancement in Nusselt number has been obtained as 3.80 times corresponding the relative arc angle (a/90) of 0.3333 at relative roughness height of 0.0422. However, the increment in friction factor corresponding to these parameters has been observed 1.75 times only.

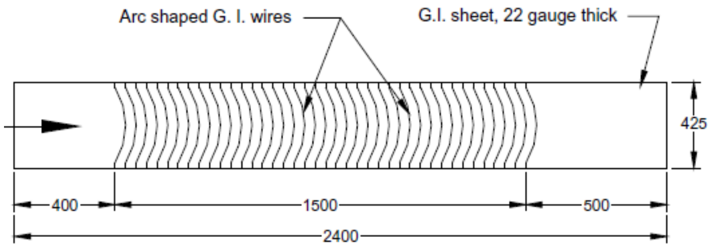

Fig. 2.1 Absorber plate.

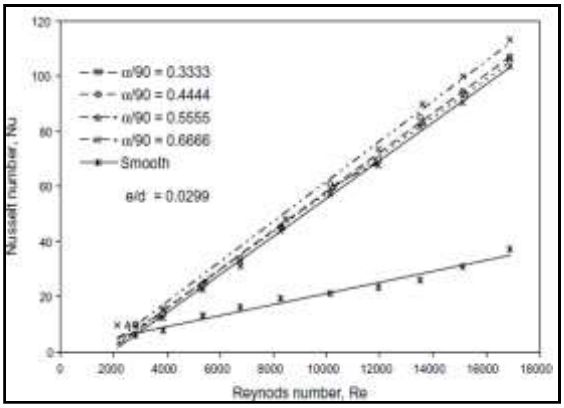

Fig. 2.2 Variation of Nusselt number with Reynolds

Bruno Facchini et al [3] carried out experimental study concerning performances at various aspect ratios of rib roughened channels (ribs inclined at a = 60_), commonly employed in turbine blade cooling systems. Experimental tests were performed with transient TLC technique on a PMMA model, properly designed to reproduce large-scale cooling ducts with different aspect ratios (AR = 1,2, 3). To validate the test model and the data acquisition and reduction procedure, a smooth channel was initially studied and relative results were found in good agreement with common correlations for heat transfer and pressure losses. A novel aspect of this study is represented by the investigation of the initial region, so as to evaluate the entrance effect influence on heat transfer coefficient of such area. Performed tests showed that, for all geometries, there is a first part (2–3 hydraulic diameters) where Nusselt is 20–25% below the value reached for fully developed flow conditions. Then it increases and exceeds such value up to 100% in the AR = 2 channel, even if globally the entrance region shows a moderate raise (maximum 20% for AR = 2). Such behavior significantly differs from the smooth channel case, where there is a monotonous decrease of heat transfer up to the developed conditions. This should warn cooling system designer from the choice of always considering fully developed conditions as a conservative assumption.

III. GOVERNING EQUATION AND SETUP

3.1 Governing equation for turbulent flow kinetic energy k

Multiplication of each of the instantaneous Navier Stokes equation by the appropriate fluctuating velocity components and addition of all the results, followed by a repeat of this process on the Reynolds equations, subtraction of the two resulting equations gives the turbulent kinetic energy as follows

$$\frac{\partial(\rho k)}{\partial t} + div(\rho k U) = div\left(-\overline{p'u'} + 2\mu\overline{u'e'_{ij}} - \rho\frac{1}{2}\overline{u'_i.u'_iu'_j}\right) - 2\mu\overline{e'_{ij}e'_{ij}} - \rho\overline{u'_iu'_j}.E_{ij} \quad (3)$$

(I) (II) (III) (IV)
(V) (VI) (VII)

Equations for mean and turbulent kinetic energy look very similar. However RHS of the equation shows that the change in turbulent kinetic energy is mainly governed by turbulent interections. Term (VII) in both the equations is equal in magnitude, but opposite in direction.

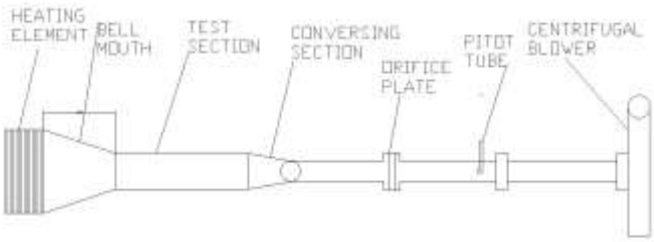

Fig.3.2 Schematic Diagram of Experimental Setup

3.3 Parts of Experimental Setup:

1) Rectangular Test Section
2) Heating Element
3) Honeycomb Section(Conversing)
4) Conversing Section to attach Test Section with Pipe
5) Orifice Plate and U tube manometer
6) Pitot Tube
7) Flexible pipe
8) Centrifugal Blower and Motor

3.4 Procedure to Perform Experiment:

1) Apply the thermographic liquid crystal sheet to a clean surface of the test section. Liquid crystal sheets used in this case works in the temperature range of 40˚C to 50˚C.
2) Subject the treated surface to known temperature levels by heater. Heat the incoming air through the heater provided at the front part of experimental set up while maintaining the test section at room temperature so as to notice the color response of liquid crystal sheet applied over the test section.
3) Measure and record the temperature response of TLC. Temperature response of sheet is captured by High Mega Pixel Camera with suitable light source setting and the temperature is recorded by digital thermometer (Cr-Al).
4) After taking temperature and colour image of the different points of test section, we made a relation between colour and temperature.
5) Now the color response of the sheet and their variation with temperature can be used to get exact temperature of any point of the test section.
6) Also record the pressure loss and head difference of U-Tube manometer to get the velocity and mass flow rate of flowing fluid at the pitot tube and orifice meter.
7) Calculations and data analysis on the basis of recorded data and given equations.

IV. MODEL FORMULATION

4.1 Roughness geometry and range of parameters.

The values of system and operating parameters of this investigation are listed in Table 4.1. The relative roughness pitch (P/e) value is selected as 8, based on the optimum value of this parameter reported in the literature . Similarly, the value of the angle of attack is chosen as 60˚, to achieve maximum enhancement of heat transfer . The arrangements of ribs on the absorber plate are shown in Fig. 4.4(a–c). In order to investigate the effect of the Values of parameters gap position and gap width on the enhancement of heat transfer and friction factor, the relative gap position (d/W) is varied from 0.33 to 0.5 of the width of duct from the trailing edge of the rib. The range of Reynolds number and relative roughness height (e/Dh) has been chosen based on the requirement of the solar air heater

_Sr. no.	Parameters	Values
1	Relative roughness pitch (P/e)	8
2	Rib height (e) and width (b)	2mm
3	Relative roughness height (e/Dh)	0.0225
4	Duct aspect ratio (W/H)	3
5	Reynolds number (Re)	3000–18000
6	Angle of attack (a)	60˚
7	Relative gap position (d/W)	0.33 and 0.5

Table.4.1 fixed parameters

4.2 Computational Fluid Dynamics Approach

Computational fluid dynamics or CFD is the analysis of systems involving fluid flow, heat transfer and associated phenomena such as chemical reactions by means of computer-based simulation. The technique is very powerful and spans a wide range of industrial and non-industrial application areas.

4.2.1 CFD Analysis

4.2.1.1 Computational Domain

The 2-D computational domain used for CFD analysis having the height (H) of 60 mm and width (W) 180 mm and total length of 250mm as shown in Fig 4.3.

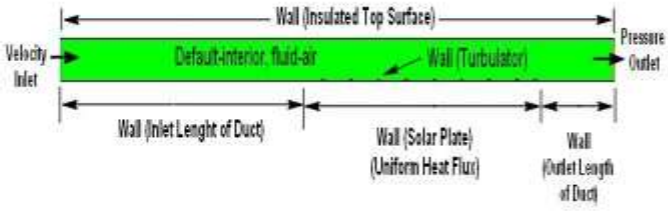

Figure 4.2 Two dimensional computational domain.

Complete duct geometry is divided into three sections, namely, entrance section, test section and exit section. A short entrance length is chosen because for a roughened duct, the thermally fully developed flow is established in a short length 2–3 times hydraulic diameter. The exit section is used after the test section in order to reduce the end effect in the test section. Roughness geometry inclined discrete shape on the roughened absorber plate has been shown in Fig. 4.4 (a,b,c).

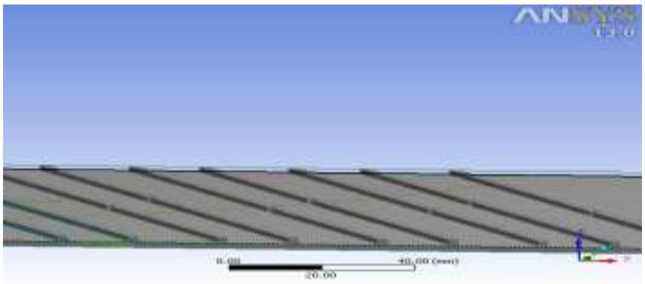

Fig.4.3.a roughness geometry

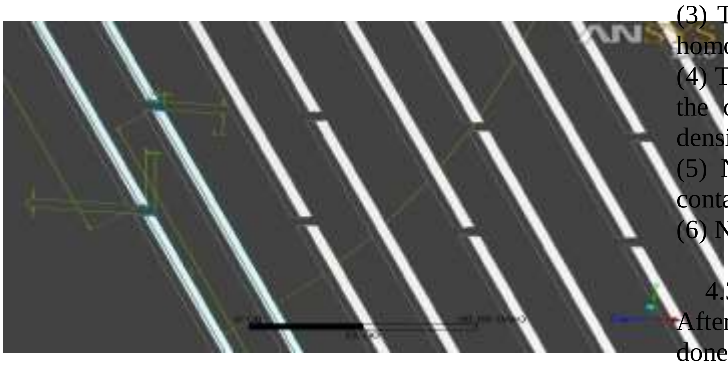

Fig.4.3.b roughness geometry

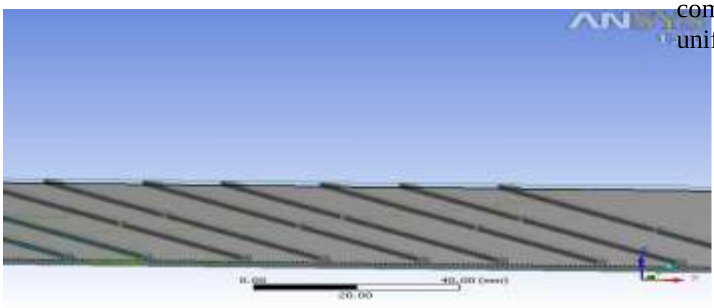

Fig.4.3.c roughness geometry

Operating and geometrical parameters of the artificially roughened solar air heater duct for computational analysis are shown in table 4.2.

In the present study, FLUENT Version 13. 0 is used for analysis.

The following assumptions are imposed for the computational analysis.

(1) The flow is steady, fully developed, turbulent and two dimensional.

(2) The thermal conductivity of the duct wall, absorber plate and roughness material are independent of temperature

Parameter	Value
Roughness pitch	20mm
HeatFlux, I	1000 W/m2
Reynoldsnumber range	3000-15000
Ductdepth, H	66mm
Ductwidth, W	195mm
Hydraulicdiameter,Dh	0.09883 m
Ribheight, e	4mm
Plate ength, L	350mm

Table 4.2: Parameters of the artificially roughened solar air heater duct for computational analysis

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(3) The duct wall, absorber plate and roughness material are homogeneous and isotropic.

(4) The working fluid, air is assumed to be incompressible for the operating range of solar air heaters since variation in density is very less.

(5) No-slip boundary condition is assigned to the walls in contact with the fluid in the model.

(6) Negligible radiation heat transfer and other heat losses.

4.3. Mesh Generation

After defining the computational domain, uniform meshing is done by rectangular elements. In creating this mesh, it is desirable to have more cells near the plate because we want to resolve the turbulent boundary layer, which is very thin compared to the height of the flow field. Fig. 4 shows the non-uniform quadrilateral meshing.

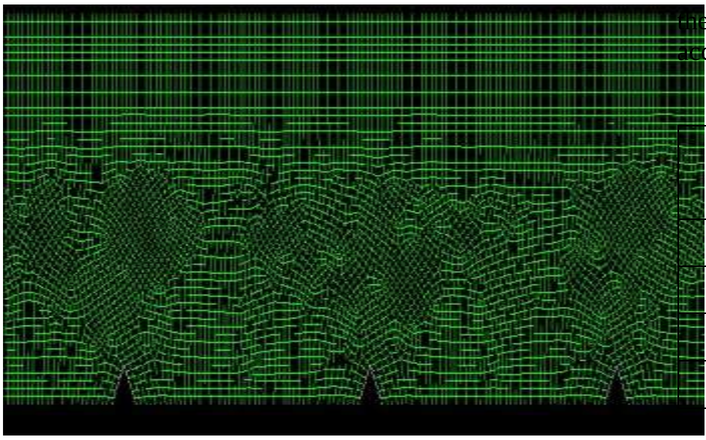

Figure 4A non-uniform quadrilateral meshing.

4.4. Boundary condition

Table 4.3 shows the detail of boundary conditions. The left edge was specified as the duct inlet and right edge as the duct outlet. Top edge was specified as top surface and bottom edges were inlet length, outlet length and solar plate. All internal edges of rectangle 2d duct were defined as turbulator wall.

4.5 Solver

ANSYS FLUENT Version 13.0 is used as a solver with k-epsilon turbulence model. The modeled turbulence kinetic energy, k, and its rate of dissipation, ε, are obtained from the following transport equations for k-ε model.

V. RESULTS AND DISCUSSION

Velocity 1 m/s

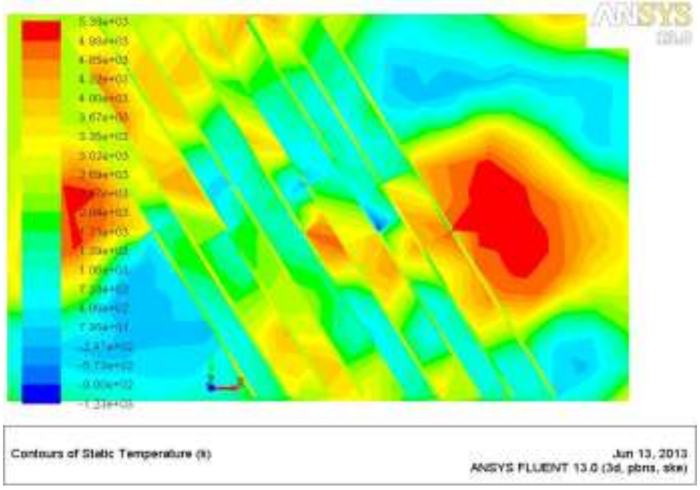

Fig.5a temperature contour for Reynolds number 5234.5

Velocity 2 m/s

The effects of various flow and roughness parameters on the heat transfer and friction characteristics for flow of air in a roughened rectangular duct are presented below. The results have been compared with those obtained in case of smooth duct operating under similar operating conditions to discuss the enhancement of heat transfer and friction factor on account of artificial roughness.

S.no.	Velocity (m/s)	Reynolds Number	Nus	Fs
1	1	5234.5	19.296	0.0092
2	2	10469	33.59	0.0078
3	2.5	13086.3	40.163	0.0074
4	3	15703.5	46.5	0.00706

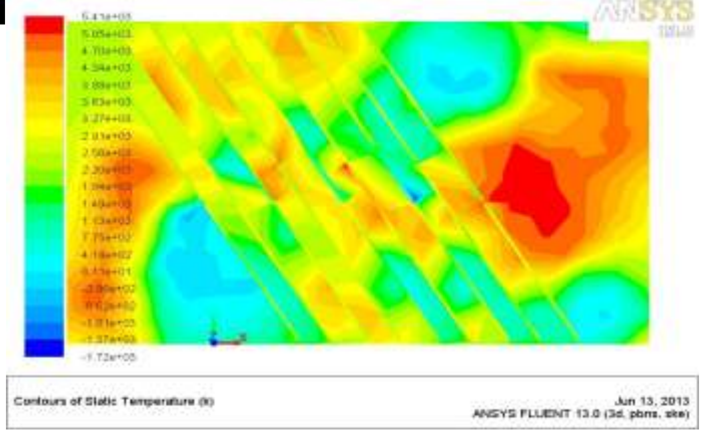

Fig.5b temperature contour for Reynolds number 10469

Velocity 2.5 m/s

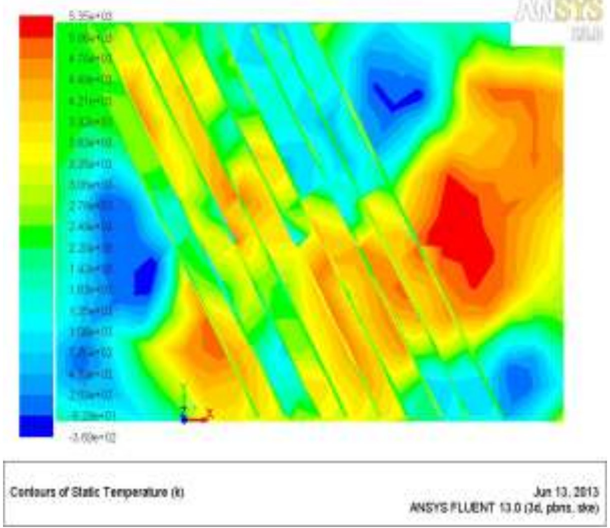

Fig.5c temperature contour for Reynolds number 13086.3

Velocity 3 m/s

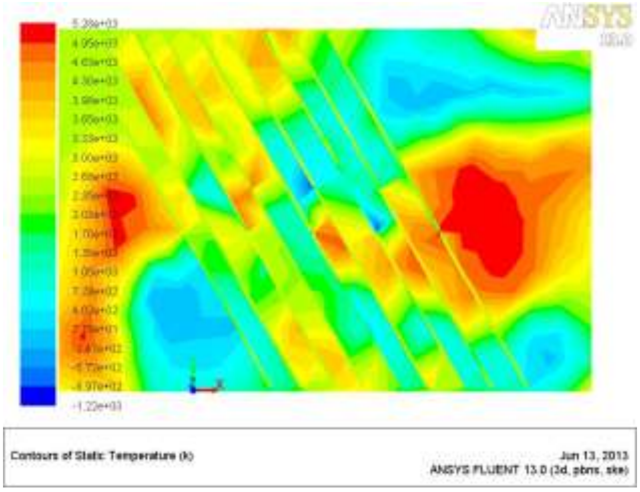

Fig.5d temperature contour for Reynolds number 15703.5

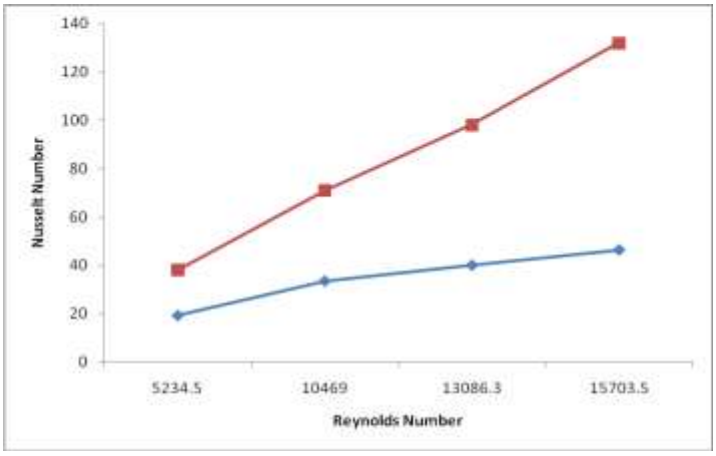

Fig.5.1 Nusselt number variation for smooth and rough surface for different reynold number

VI. CONCLUSIONS

The present work was undertaken with the objectives of extensive investigation on the 900 shaped ribs as artificial roughness on the surface of duct. Experimental setup for heat transfer and pressure lose have been design and developed. Data were collected for local heat transfer and pressure loss of these artificially roughened ducts. Results of artificially roughened duct have been compared with those of a smooth duct under similar flow condition to determine heat transfer and friction factor. The major conclusions from this investigation are given below:

1.As per the guide lines of ASHRE standard 93-77 an experiment setup comprising artificially roughened duct equipped with measuring and control instruments has been designed and developed to investigate the local heat transfer and pressure loss characteristics.

2. In the present study actual mechanism of heat transfer of smooth and artificially roughened surface has been studied using liquid crystal sheet which shows different colours corresponds to different temperature at different instant of time

REFERENCES

[1] K.R. Aharwal, B.K. Gandhi, J.S. Saini "Experimental investigation on heat-transfer enhancement due to a gap in an inclined continuous rib arrangement in a rectangular duct of solar air heater", Renewable Energy, Volume 33, Issue 4, Pages 585-596, April 2008.

[2] M.M. Sahu, J.L. Bhagoria "Augmentation of heat transfer coefficient by using 90° broken transverse ribs on absorber plate of solar air heater" Renewable Energy, Volume 30, Issue 13, Pages 2057-2073, 2005.

[3] Varun, R.P. Saini, S.K. Singal "A review on roughness geometry used in solar air heaters", Solar Energy, Volume 81, Issue 11, Pages 1340-1350, 2007.

[4] V. SriHarsha, S.V. Prabhu , R.P. Vedula, Influence of rib height on the local heat transfer distribution and pressure drop in a square channel with 90 continuous and 60 V- broken Applied Thermal Engineering 29 (2009) 2444–2459.

[5] Bruno Facchini, Luca Innocenti, Marco Surace, Design criteria for ribbed channels: Experimental investigation and theoretical analysis, International Journal of Heat and Mass Transfer 49 (2006) 3130–3141

[6] P.R. Chandra, C.R. Alexander, J.C. Han, Heat transfer and friction behavior in rectangular channels with varying number of ribbed walls, International Journal of Heat and Mass Transfer 46 (2003) 481–495.

[7] F. Giglitto, The use of turbulence promoters in gas turbine cooling systems (in italian). Master thesis. University of Florence, 2003.

[8] D. Graham, J. Rhine, The design of transient wall heating experiments for the determination of convective heat transfer using liquid crystal thermography, ASME Paper (2000-GT-658), 2000.

[9] Diego Cavallero, Giovanni Tanda, An experimental investigation of forced convection heat transfer in channels with rib turbulators by means of liquid crystal thermography, Genova, Italy(2001)-16145

[10] C. Camci, K. Kim, S.A. Hippensteele, A new hue capturing technique for the quantitative interpretation of liquid cry stalimages used in convective heat transfer studies, ASME J. Turbomach. 114 (1992) 765–775.

[11] Gee, D.L., and Webb, R.L., 1980, "Forced Convective Heat Transfer in Helically Rib Roughened Tube", Int. J. Heat & Mass Transfer, Vol21, pp. 127-1136

[12] Han, J.C., 1984, "Heat Transfer and Friction in Channels with Two Opposite Rib Roughened Walls", ASME J. Heat Transfer, Vol 6, pp. 774-781.

[13] Prasad, K., and Mullick, S.C., 1985, "Heat transfer Characteristics of a Solar Air Heater Used for Drying Purposes", J. Applied Energy, Vol 13, pp. 55-64.

[14] Miin, L.T., and Hwang, J.T., 1993, "Effect of Ridge Shapes on Turbulent Heat Transfer and Friction in a Rectangular Channel", Int. J. Heat & Mass Transfer, Vol 30, pp. 931- 940.

[15] Gupta, D., Solanki, S.C., and Saini, J.S., 1997, "Thermohydraulic Performance of Solar Air Heater with Roughened Absorber Plate", J. Solar Energy, Vol 61, No.1, pp. 33-42. [16] Prasad, B.N. and Saini, J.S., Effect of artificial roughness on heat Transfer and friction factor in a solar air heater, Solar Energy, Vol. 41, No. 6, 555-560, 1988.

[17] Karwa, R., Solanki, S.C., Saini, J.S., 2001, “Thermohydraulic Performance of Solar Air Heater having Integral Chamfered Rib Roughness on Absorber Plates”, Int. J. Energy, Vol 26, pp. 161-176.
[18] Momin, A.M.E., Saini, J.S., Solanki, S.C., 2002, “Heat Transfer and Friction in Solar Air Heater Duct with V-Shaped Rib Roughness on Absorber Plate”, Int. J. Heat & Mass Transfer, Vol 45, pp. 3383-3396.
[19] Nikuradse, J., 1970,”Laws of Flow in Rough Pipes”, NACA, Technical Memorandum 1292

