ISSN 2229-5518

Heat Transfer and Pressure Drop Characteristics In A Duct With Roughened Absorber Plate Having Rectangular Transversed Rib Using ANSYS

Gupta A. D, Dr. Varshney L, Dr. Pratihar A. K.

Abstract— The heat transfer coefficient of a solar air heater can be increased by providing artificial roughness on the underside of the absorber plate. A computer simulation in ANSYS has been carried out in the present work to study the heat transfer and pressure drop by using transverse ribs in the form of rectangular protrusion on the absorber plate of a solar air heater under simulated conditions. The computer simulation encompasses the Reynolds number from 4000 to 16000, relative roughness pitch from 6 to 25 and relative roughness height equal to 0.044 m. The effect of these parameters on heat transfer coefficient, pressure drop and thermal efficiency has been studied. It is found that at relative roughness pitch of 10, the heat transfer, pressure drop and thermal efficiency are maximum.In the present work, Nusselt number and friction factor correlation developed as a function of Reynolds number at p/e =10 are given below;

$$Nu = 0.134264 \times Re^{0.66}$$
$$f = 0.26550 \times Re^{-0.181237}$$

Index Terms— ANSYS, Artificial roughness, CFD Analysis,Nusselt Number, Renoldes Number,Solar air heater, turbulent flow.

1 Introduction

For harnessing solar thermal energy, different types of collection devices are in use e.g., flat plate collectors for low temperature applications (below 100 ºC) and focusing collectors for high temperature applications above 100 ºC. A flat plate collector, a blackened metal or glass surface is exposed to solar radiations so that the solar radiations are absorbed and converted into heat energy. Generally, water heating and air heating is done using flat plate collectors. Solar air heaters are very popular among solar thermal applications due to their simple design and low cost.

The thermal efficiency of a solar air heater is generally less because of low heat transfer coefficient between absorber plate and air flowing in the duct. To make solar air heater more effective, thermal efficiency needs to be improved by using enhancement techniques. There are several methods which can be used to increase the efficiency of solar air heaters. Several studies have been published in the literature by Chaube et al[1],[2], [3][4] etc., to increase the efficiency of solar air heaters. Various techniques used to improve efficiency of air heater include finned surface, corrugated surface and surface with artificial roughness etc. In the current scenario, the focus of researchers is on adding artificial roughness beneath the absorber plate due to several advantages over other methods. Various types of ribs are used for creating roughness on absorber plate. Rectangular ribs are preferred due to their ease of manufacturing and better strength

Several experimental investigations and numerical studies have been performed by[5], [6], [7], [8] etc. to study the effects of ribs of different geometries. But, experimental and numerical methods are cumbersome to visualize the effects of ribs on the performance of air heaters. In the present work, the computer simulation of heat transfer, pressure drop and air flow pattern in a rectangular duct with constant heat flux has been carried out to visualize and study the effect of rectangular ribs on the performance of air heater using ANSYS software. Main parameters studied in this work are mass flow rate and relative pitch (ratio of pitch and rib height).

2 Methodology

The performance of an air heater largely depends on the design of absorber plate which absorbs heat from a source and transmits to air. Therefore, an absorber plate must be designed properly before the fabrication of an air heater. System simulation is an easy and convenient way to study the performance of such system under various operating conditions. Rough surfaces are widely used to enhance convective heat transfer by promoting turbulence. The major drawback of increased roughness is the increase in frictional and form drag. Several theoretical studies have been performed to analyze the heat transfer characteristics of various types of rough surfaces. This is desirable due to difficulties in conducting experiments and high turbulence intensities that increase difficulties of measurement. Because of this reason numerical studies are becoming more common. In the turbulent region, simulation technique plays a critical role in determining the results with more accuracy.

ANSYS is a general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numeri-

ISSN 2229-5518

cal method of constructing a complex system into very small pieces called elements. The software implements equations that govern the behavior of these elements and solves them all creating a comprehensive explanation of how the system acts as a whole. These results then can be presented in tabulated or graphical forms.

Use of general purpose software ANSYS has been used with FLOTRON CFD module

In the present work FLOTRAN CFD is used for 2-D analysis of air flow in a rectangular duct with constant heat flux on one side.

In the present study the behavior of fluid flow has been considered to be uniform along the width of the plate and the ANSYS FLOTRAN element, FLUID141 has been used to analyze flow, pressure and temperature distributions in a single phase viscous fluid. For these elements the ANSYS program calculates velocity components, pressure and temperature from the conservation of three properties: mass, momentum and energy.

3 Problem Formulation

The following methodology is adapted for modelling and analysing heat transfer and pressure drop in the duct in simulated condition:

3.1 Selection of flow domain

For analysis of solar air heater duct for heat transfer and pressure drop as shown in Fig 1. A system has been chosen following the studies made as per ASHRAE Standerd[9]. A rectangular duct with height (H) of 40 mm, rib height (e) of 3.4 mm, rib width of 5.8 mm and pitch (p) of 34 mm has been considered for the analysis. In the experimental details proposed byKarwa [10], the thickness of the heating plate is only 1 mm, which is very small in comparison to the surface area normal to the heat flow.Hence, the Biot number is very small (less than 10^{-3}). This allows us to neglect the internal resistance in comparison to convective resistance as recommended by P. K. Nag [11]]. Therefore, the uniform heat flux of 1000 W/m^2 is given on ribbed surface, neglecting the conduction resistance within the plate.

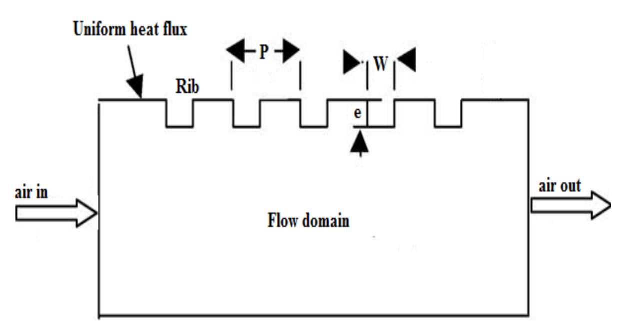


Fig.1. Flow domain

3.2 Flow regime

1. Flow is turbulent.
2. Flow of gas is considered to be ideal.
3. The effect of temperature on fluid density, viscosity and thermal conductivity is taken into account.
4. On the basis of Mach no. flow is considered to be compressible.

3.3 Finite element mesh

In order to obtain best results, mapped meshing has been used where in the region near the walls has highest gradients due to turbulence so a much denser meshing is created there to capture significant effects. Meshing has been show in Fig. 2.

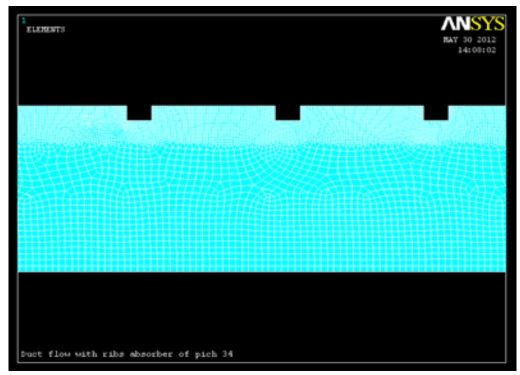


Fig.2. Mapped mesh sample

3.4 Boundary conditions

The following boundary conditions are applied for obtaining results using FLOTRON CFD:

1. Condition of no slip is applied for air solid contact on the duct wall and absorber surface.
2. Pressure at the outlet of duct is considered atmospheric hence gauge value is taken to be zero.
3. Uniform heat flux at the absorber surface.
4. Adiabatic condition on wall of duct.
5. Mean velocity used at the inlet is constant velocity according to mass flow rate.
6. Constant inlet temperature equal to ambient temperature.

3.5 Solution of the problem

In this step, monitoring of solution is done for convergence and stability of the analysis by observing the rate of change of the solution and the behavior of relevant dependent variables. These variables include velocity, pressure, temperature and (if necessary) turbulence quantities such as kinetic energy (degree of freedom ENKE), kinetic energy dissipation rate (ENDS) and effective viscosity (EVIS). An analysis typically requires multiple restarts.

In last step the post process output quantities are calculated and the results in the output are obtained at various nodes. These values can be utilized to estimate:

1. Convective heat transfer coefficient
2. Temperature distribution in duct
3. Pressure distribution in duct

4 Results & Discussion

When fluid enters a closed channel at a uniform velocity, the

ISSN 2229-5518

fluid particles in the layer in contact with wall of the channel come to complete rest. This layer also causes the fluid particles in the adjacent layers to slow down gradually due to friction loss. As a result, velocity gradient develops along the channel. As the roughness element lies under the absorber plate, the flow becomes turbulent because of reattachment points or brakeage of hydrodynamic boundary layer at regular intervals. In this process heat transfer coefficient, friction factor and pumping power of fluid increase due to the presence of this artificially roughened rectangular duct, the behavior of Nusselt number with different operating parameters of solar air heater having rectangular ribs on the backside of the absorber plate. The effect of roughness and operating parameters on the heat transfer coefficient has been examined and a comparison of performance of roughened solar air heater with that of conventional solar air heater having smooth duct has been made.

4.1 Validation of ANSYS results for the smooth Duct

(a) For Nusselt number (Nu)

The values of Nusselt number (Nu) obtained from the ANSYS results have been compared with the values obtained from Dittus-Boelter equation in Fig. 3. It has been observed that the variation in results obtained from ANSYS lie within ±6.5% of the predicted value by Dittus-Boelter equation which establishes the authenticity of the results obtained from ANSYS.

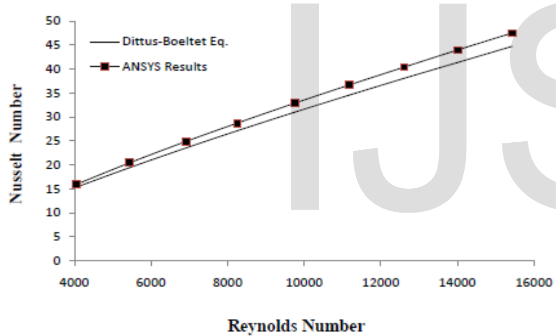


Fig.3 Comparison of Nusselt number predicted by ANSYS and estimated by Dittus Boelter equation for smooth duct

(b) Pressure drop (ΔP) for smooth duct

The pressure drop across duct length obtained from ANSYS has been compared with that obtained from Blasius equation indicated by Eq. 1. The comparison has been shown in Fig. 4. It has been observed from the ANSYS results that the variation between predicted and ANSYS results lie within ± 9%.The reason for variation in values obtained by Blasius equation and ANSYS results is that the ANSYS results also include minor pressure loses at the entrance and exit sections which are not included in Blasius equation. The comparison indicates that the results obtained through ANSYS are reliable and in good agreement with those by Blasius equation

$$f = 0.085\,\mathrm{Re}^{0.25}$$

$$head\_loss = \frac{4flv^2}{2dg} \qquad \text{………….(1)}$$

$$\Delta P = \text{head loss} \times \rho \times g$$

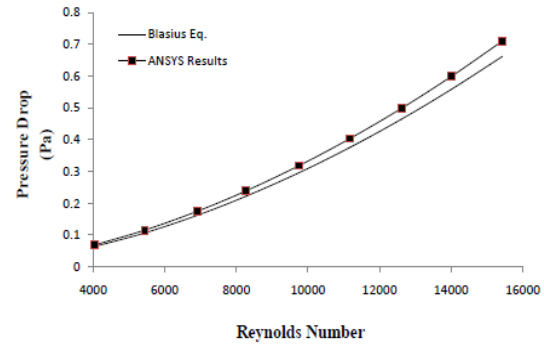


Fig.4 Comparison of pressure drop predicted by ANSYS and estimated by Blasius equation for smooth duct

After conforming the reliability of ANSYS results it was decided to study the effect of operating and system parameters on the performance of roughened plate solar air heater having different geometries. The systematic results have presented in following sections.

4.2 Effect of Reynolds number on convective heat transfer coefficient

Figure 5 shows the variation of convective heat transfer coefficient with Reynolds number. As the Reynolds number increases, the heat transfer coefficient also increases for different values of relative roughness pitches. The variation in heat transfer coefficient is low at small Reynolds numbers while it is large at higher Reynolds numbers. This behavior seems due to increased turbulence at higher Reynolds numbers and also due to breakage of thermal boundary layer at higher Reynolds numbers.

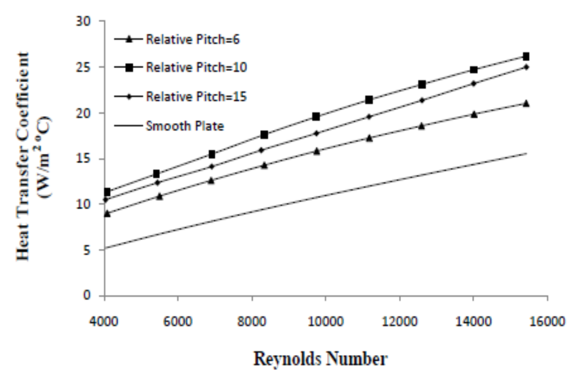


Fig.5 Comparison of smooth and roughened plates for different values of convective heat transfer coefficient

4.3 Effect of Reynolds number on Nusselt number

Figure 6 shows the variation of Nusselt number with Reynolds number. As the Reynolds number increases, the Nusselt number also increases for different values of relative roughness pitches. The variation in Nusselt number is low at small Reynolds numbers while it is large at higher Reynolds numbers due to breakage of thermal boundary layer at higher Reynolds

ISSN 2229-5518

numbers and large turbulence.

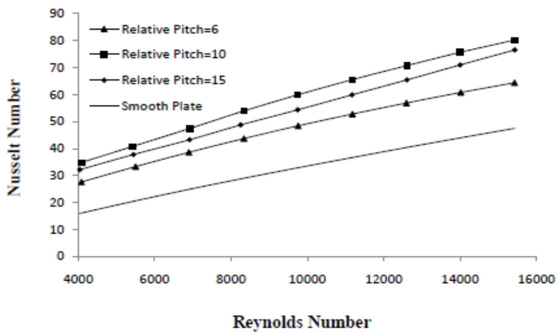


Fig.6 Comparison of Smooth and Roughened Plates for Different Values of Nusselt number

4.4 Effect of Reynolds number on heat utilization

Utilzation of heat given to system is an indicaton of performance of the collector. In order to estimate the value of heat utilization, losses have been determined for different value of Reynolds number. The total heat supplied minus losses gives the amount of heat utilization. Fig. 7 shows the variation of heat utilized in duct with Reynolds numbers for different values pitch. From the plot, it is observed that the heat utilization increase with increase in Reynolds number for all the values of pitch. This effect is due to amount of losses for different values of Reynolds number. The losses for low Reynolds number are higher as compared to losses at increased Reynolds number. Also it is observed that variation in heat utilization for different pitch for same Reynolds number is insignificant at higher values of Reynold numbers. This phenomenon is probably due to the fact that collector is operating at lower temperatures where losses become insignificant.

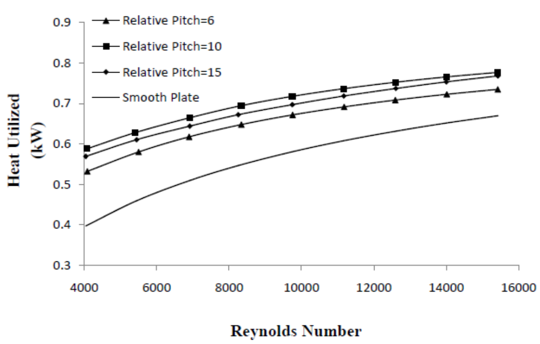


Fig.7 Comparison of smooth and roughened plates for different values of heat utilizes Boelter equation for smooth duct

4.5 Effect of mass flow rate on temperature of air at exit of duct

Figure 8 shows the variation of temperature of air at exit with mass flow rate. As the mass flow rate increases, the heat carrying capacity of air increases. Because of this heat utilization also increases, but the rate of increase of heat capacity is more significant than the rate of increase of heat utilization. Therefore the temperature of air at exit decreases as a result increase in mass flow rate.

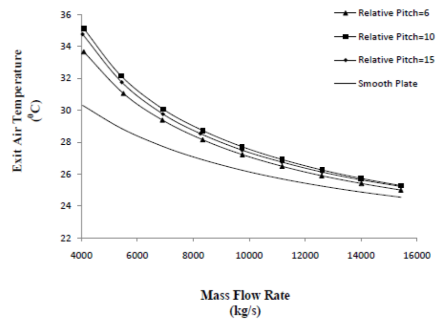


Fig.8 Comparisons of smooth and roughened plates for different values oftemperature of air at exit of duct

4.6 Effect of Reynolds number on thermal efficiency

Figure 9 shows the variation of thermal efficiency with Reynolds number. As the Reynolds number increases, the heat transfer coefficient increases and collector operates at lower temperature of absorber plate which reduces the heat loss and increases thermal efficiency. Maximum efficiency is obtained for plate having relative roughness pitch (p/e)=10 in the entire range of Reynolds number under study.

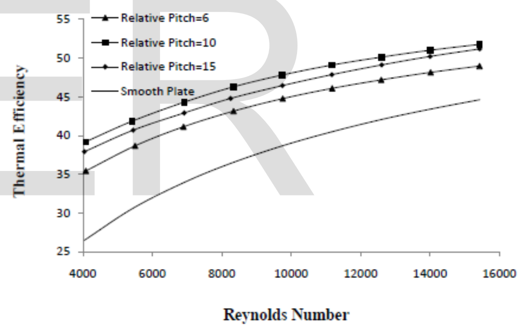


Fig.9 Comparison of Smooth and Roughened Plates for Different Values of thermal efficiency

4.7 Effect of relative roughness pitch on convective heat transfer cofficient

The variation of convective heat transfer coefficient with relative roughness pitch parameter for the geometry under study can be explained on the basis of flow separation shown in Fig 10(a), (b), (c). For plate with higher p/e ratio, the flow does not reattach after it detaches from each rib before it reaches the succeeding rib. Also, the maximum heat transfer is achieved near the reattachment point. Hence, it is concluded that at p/e=10, the convective heat transfer coefficient is maximum. It is observed that for roughness of low relative pitch (p/e), flow does not attaches the absorber plate and for high relative roughness flow continously attach the absorber plate and form laminar sublayer. In both cases heat transfer coefficient is low. It is found that for relative roughness 10, the attachment and deattachment is almost instantaneous, hence heat transfer coefficient is maximum for p/e =10.

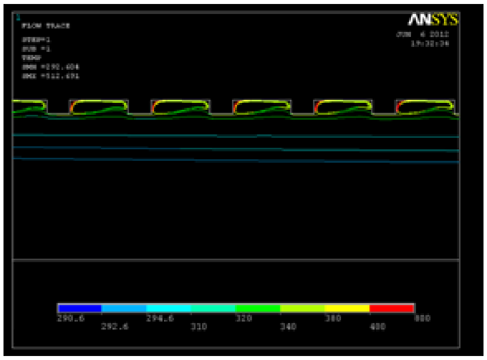


Fig.10 (a) Flow pattern for relative roughness pitch (p/e) =6

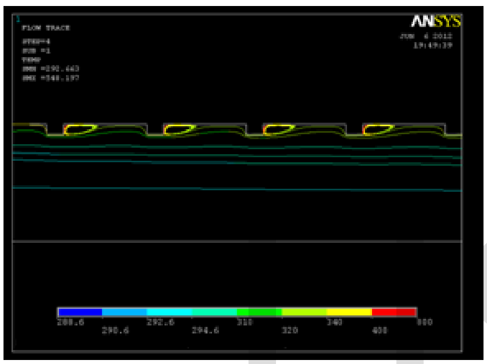


Fig.10 (b) Flow pattern for relative roughness pitch (p/e) =10

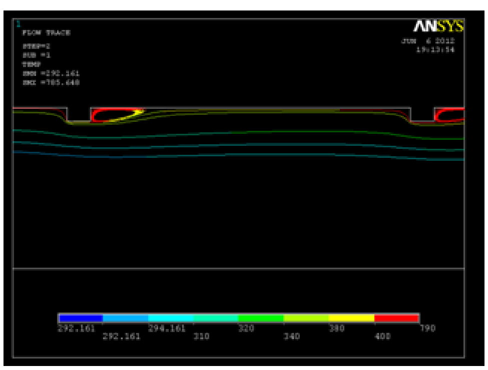


Fig.10 (c) Flow pattern for relative roughness pitch (p/e) =25

4.8 Effect of relative roughness pitch on temperature of air at exit of duct

The variation of temperature of air at exit of duct with relative roughness pitch parameter has been shown in Fig. 11. It has been observed that for a given mass flow rate, increase in the value of p/e up to 10, temperature increases due to increase in heat utilization and after that it starts to decrease as heat utilization decreases. Temperature of air at exit decreases for all the relative pitches under study. The temperature of air at exit of duct is found to be highest at p/e=10 because for all mass flow rates, the heat utilization is maximum at this relative roughness pitch.

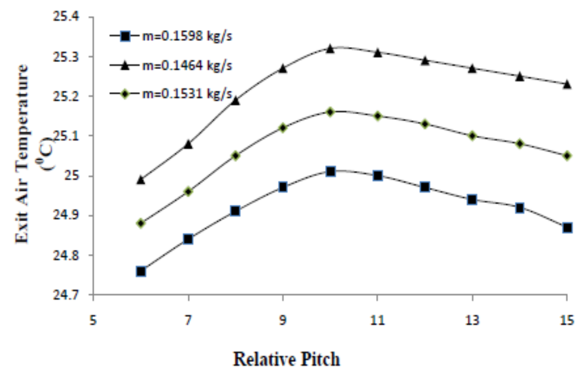


Fig.11 Variation of thermal eficiency in duct with relative pitch

4.9 Effect of relative roughness pitch on thermal efficiency

Variation of thermal efficiency with relative roughness pitch parameter has been depicted in Fig. 12. It is observed that for p/e ratio from p/e 6 to 10 thermal efficiency increases, after that it start decreasing. Thermal efficiency is maximum at p/e=10. Optimum value of thermal efficiency is being obtained at p/e=10 because heat transfer coefficient is maximum for this value.

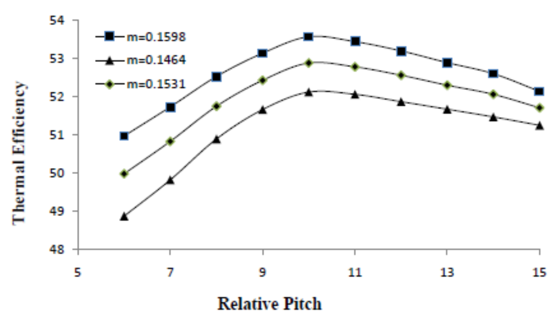


Fig.12 Variation of thermal eficiency in duct with relative pitch

4.10 Effect of Reynolds number on pressure drop in the duct

Figure 13 shows the variation of pressure drop in the duct with varying Reynolds numbers. Results show that for all plates, pressure drop increase with increase in Reynolds number for rough as well as smooth plate. Similar trend in the variation of pressure drop in the duct has been obtained for all plates having different relative roughness pitch. This is due to the fact that pressure drop in the duct is significantly frictional pressure drop. As Reynolds number increases the velocity gradient in the duct also increases which increase frictional force and hence pressure drop.

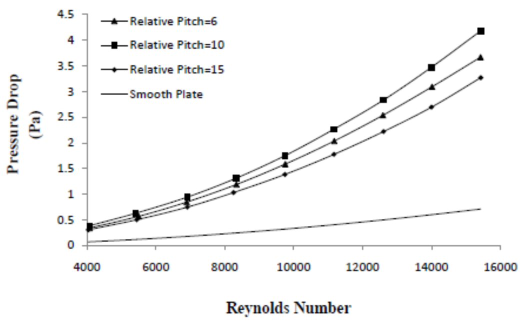


Fig.13 Comparison of Smooth and Roughened Plates for Different Values of pressure drop in duct

4.11 Effect of relative roughness pitch on pressure drop in the duct

Figure 14 shows the variation of pressure drop in the duct with relative roughness pitch parameter. It has been observed that with increase in the value of p/e, pressure drop in the duct firstincrease up to p/e=10 then it starts decreasing for all the plates under study. Maximum pressure drop is found for p/e ratio 10. In duct with smooth plate, laminar sub layer is formed adjacent to surface, hence a part of flow remains laminar. But in presence of ribs this laminar flow is disturbed and becomes turbulent. In flow simulation, maximum disturbance has been observed for relative roughness pitch p/e=10, hence maximum pressure drop occur at p/e=10.

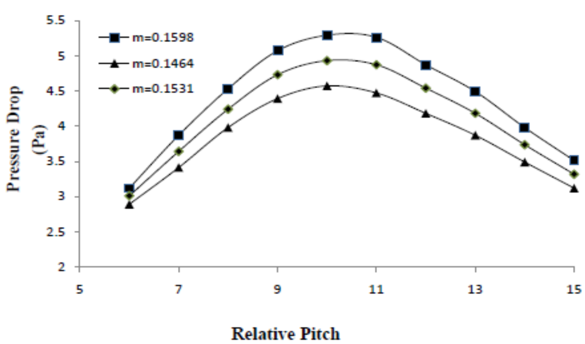
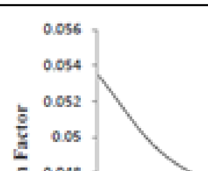


Fig.14 Variation of pressure drop in duct with relative pitch

4.12 Variation of friction factor with Reynolds number

The plot for variation in friction factor with Reynolds number has been shown in Fig. 15(a) and Fig.15(b) for smooth and roughened plate respectively. Friction factor for smooth as well as roughened plates decrease as Reynolds number increases.



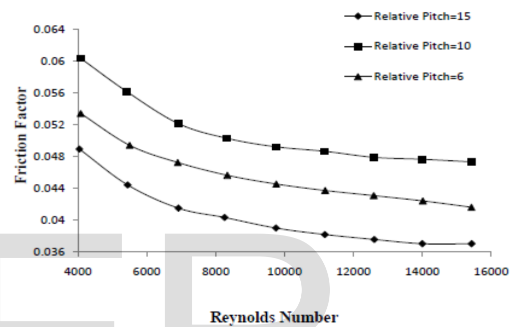


Fig.15(b) Variation of Exit air temperature in duct with relative pitch

5 Flow Pattern Studies

On the basis of graphical output of ANSYS the following efects were observed, by which above results can be explained.

5.1 Effect of ribs

The most important effect produced by the presence of a rib on the flow pattern is the generation of two flow separation regions, one on each side of the rib. The vortices so generated are responsible for the turbulence and hence the enhancement in heat transfers as well as in the friction losses takes place. A considerable influence of the presence of ribs is more pronounced in turbulence intensity distribution. Fig. 16 shows the pressure distribution to show flow separation region on each side of the rib.

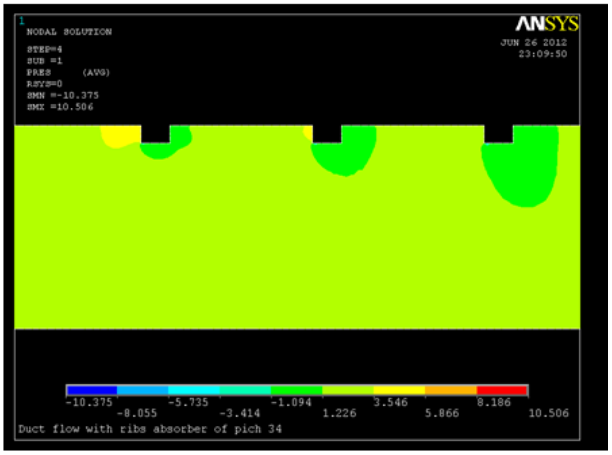


Fig.16 Pressure distribution to show flow separation region

5.2 Effect of relative roughness pitch

Figure 10 (a), (b) and (c) show the flow patterns as a function of relative roughness pitch. Due to flow separation downstream of a rib, reattachment of the shear layer does not occur for a pitch ratio of less than about 8. Maximum heat transfer has been found to occur in the vicinity of a reattachment point. For relative roughness pitch considerably less than about 8, the reattachment will not occur at all, resulting in the decrease of heat transfer enhancement. However, an increase in pitch beyond about 10 also results in decreasing the enhancement..

6 Conclusion

On the basis of investigation, following conclusions have been drawn:

1 Based on the analysis, it has been found that the performance of solar air heater can be enhanced by providing artificial roughness in the form of ribs on the underside of the absorber plate than the conventional flat plate solar air heater.

2 Nusselt number has been found to be highest for an absorber plate having relative roughness pitch equal to 10.

3 Thermal efficiency increases with increases in mass flow rate.

References

[1] A. Chaube, P. K. Sahoo, and S. C. Solanki, "Analysis of heat transfer augmentation and flow characteristics due to rib roughness over absorber plate of a solar air heater," Renew. Energy, vol. 31, no. 3, pp. 317–331, 2006.

[2] A. Lanjewar, J. L. Bhagoria, and R. M. Sarviya, "Heat transfer and friction in solar air heater duct with W-shaped rib roughness on absorber plate," Energy, vol. 36, no. 7, pp. 4531–4541, 2011.

[3] S. Kiwan, "Size optimization of conventional solar collectors," vol. 23, no. 5, pp. 373–378, 1998.

[4] B. N. Prasad, "Thermal performance of artificially roughened solar air heaters," Sol. Energy, vol. 91, pp. 59–67, 2013.

[5] K. R. Aharwal, B. K. Gandhi, and 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," Renew. Energy, vol. 33, pp. 585–596, 2008.

[6] J. L. Bhagoria, J. S. Saini, and S. C. Solanki, "Heat transfer coefficient and friction factor correlations for rectangular solar air heater duct having transverse wedge shaped rib roughness on the absorber plate," Renew. Energy, vol. 25, pp. 341–369, 2002.

[7] A. S. Yadav and J. L. Bhagoria, "A numerical investigation of square sectioned transverse rib roughened solar air heater," Int. J. Therm. Sci., vol. 79, pp. 111–131, 2014.

[8] V. S. Hans, R. P. Saini, and J. S. Saini, "Heat transfer and friction factor correlations for a solar air heater duct roughened artificially with multiple v-ribs," Sol. Energy, vol. 84, no. 6, pp. 898–911, 2010.

[9] ASHRAE standard 93, Method of Testing to Determine the Thermal Performance of Solar Collectors. 30329. Atlanta, GA: American Society of Heating, Refrigerating & Air Conditioning Engineers, 2003.

[10] R. Karwa, "Experimental studies of augmented heat transfer and friction in asymmetrically heated rectabgular ducts with ribs on the heated wall in transverse, inclined, v-continous and v-discrete pattern," Int. Commun. Heat Mass Transf., vol. 30, no. 2, pp. 241–250, 2003.

[11] P. K. Nag, Heat Transfer, 8th ed., ISBN 0-07-059114-8. Page No. 697, 2007.