CFD ANALYSIS OF HEAT TRANSFER AND FRICTION CHARACTERISTICS OF SOLAR AIR HEATER DUCT USING BROKEN 'S' SHAPED RIBS ROUGHNESS

Robin (M.Tech. Scholar), SCOPE College Of Engineering, Bhopal
Dr. S. K. Nagpure (Asso.Prof.), SCOPE College Of Engineering, Bhopal

Abstract

In this paper, results of CFD analysis on heat transfer and friction in rectangular ducts with 'S' shape with gap roughness has been presented. The rib roughness has relative roughness pitch was varied from 8 to 11, arc angle of 30° and relative roughness height of 0.043. The relative gap width of 1.0. The effects of relative roughness pitch on Nusselt number, friction factor and thermo-hydraulic performance parameter have been discussed and results compared with smooth duct under similar conditions. The rough ribs were efficient enough to transfer the desired heat, but they are not economical and are very complex in design and construction. Whereas, roughness gives more area of contact. So, there is enough time to transfer the heat from the ribs to the passing air which touches the ribs and the heat transfer takes place efficiently. Thus, the 'S' shaped rib with different relative roughness pitch, when compared with the smooth plate, it was concluded that the related Nusselt Number and friction factor for 'S' shaped rib with different relative roughness pitch was more efficient. The smooth plate could not transfer the desired heat due to absence of friction; so, they are not preferred practically. One of the most important techniques used are passive heat transfer technique. These techniques when adopted in heat transfer surfaces proved that the overall thermal performance improved significantly. Rib roughness on the underside of the top wall of a duct has been found to substantially enhance the heat transfer coefficient. Surface roughness disturbs the laminar sub-layer in the turbulent flow and promotes local wall turbulence that, in turn, increases the heat transfer from the surfaces. The augmentation in heat transfer accompanies a higher pressure drop penalty of the fluid flow. In this work the maximum value is found to be relative roughness pitch 10 at a Reynolds number of 16000.

Keyword: Solar air heater**,** Nusselt number, Heat transfer, Friction factor, Relative roughness pitch,Relative gap width.

Introduction: 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. In order to make the solar air heaters economically viable, their thermal efficiency needs to be improved by enhancing the heat transfer coefficient. Turbulence promoters either in the form of surface roughness or in the form of three dimensional surface protuberances tends primarily to increase the heat transfer coefficient due to disturbance or destruction of the viscous sub-layer near the wall. The key dimensions of the roughness geometry are the relative roughness height, the relative roughness spacing and the shape of the roughness element. The optimal geometry of roughness depends mostly on dynamic conditions in the boundary layer and on the properties of fluid. Heat transfer enhancement by inserting ribs is commonly used application in tubes and ducts. Ribs improve the heat transfer by interrupting the wall sub layer. This yields flow turbulence, separation and reattachment leading to higher heat transfer rates. Due to the existence of ribs effective heat transfer surface increases. Many researchers have been carried out on heat transfer enhancement achieved by different ribs. 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 and turbine blades. Several investigations have been carried out to study effect of artificial roughness on heat transfer and friction factor for two opposite roughened surface by Han[2,3].Han et al.[4-5],Wrieght et al.[7],Lue et al,[8-10],Taslim et al. and Hwang[12],Han and Park[14],Park et al.[15] developed by different investigators . The orthogonal ribs i.e. ribs arranged normal to the flow were first used in solar air heater and resulted in better heat transfer in comparison to that in conventional solar air heater by Prasad k, Mullick S.C. et al [16].Many investigators Gao x sunden B[17],Han J.C,GlicksmanLR,Rohsenow WM[18],Prasad BN,Saini JS{19],TaslimME,LiT,KercherDm[20],Webb RL,EckertErg,Goldstein RJ[21] have reported in detail the Nu and f for orthogonal and inclined rib-roughened ducts. The concept of V-shaped ribs evolved from the fact that the inclined ribs produce longitudinal vortex and hence higher heat transfer .In principal ,high heat transfer coefficient region can be increased two folds with V-shape ribs and hence result in even higher heat transfer et al. [20].The beneficial effect on Nu and f caused by V-shaping of ribs in comparison to angled ribs has been experimentally endorsed by several investigators Geo X,Sunden B{22],Karwa R.[23],KukrejaRT,LueSC,McMillin RD[24],Lau SC,McMillinRD,Han JC[25], for different roughness parameters and duct aspect ratios . In addition ,multiple V-ribs have also been investigated with the anticipation that the more number of secondary flow cells may result in still higher heat transfer et at LanjewarA,BhagoriaJL,Sarviya RM[26],Hans VS,SainiRP,Saini JS[27].Chao et al.[28]examined the effect of an of angle of attack and number of discrete ribs, and reported that the gap region between the discrete ribs accelerates the flow, which increases the local heat-transfer coefficient. In a recent study ,Chao et al.[29] investigated the effect of a gap in the inclined ribs on heat transfer in a square duct and reported that a gap in the inclined rib accelerates the flow and enhances the local turbulence, which will result in an increase in the heat transfer. They reported that the inclined rib arrangement with a downstream gap position shows higher enhancement in heat transfer compared to that of the continuous inclined rib arrangement. Computational studies have also been used extensively in studying the flow and heat transfer effects in ribbed ducts. The advantage of being able to study both the flow and heat transfer in the entire flow field is worth the effort required to simulate ribbed duct flows, but the whether the channel roughened with ribs of different shape can improve the heat transfer rate. There have been attempts undertaken to overcome the adverse effect by varying the geometry of ribs. Lockett and Hwang employed the non-invasive optical method of holographic inter ferometry to investigate the heat transfer in turbulent flow over square and rounded rib-roughness elements. They found that the heat transfer distribution depends on the Reynolds number for the rounded rib, but independent for square rib geometry. In both cases, the minimum heat transfer occurred at the base of the rear facing rib wall.

II. Computational Fluid Dynamics

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. The 2-dimensional solution domain used for CFD analysis has been generated in ANSYS version 14.5 (workbench mode) as shown in Fig.1.The solution domain is a horizontal duct with broken ‘S’ shaped ribs roughness on the absorber plate at the underside of the top of the duct while other sides are considered as smooth surfaces.

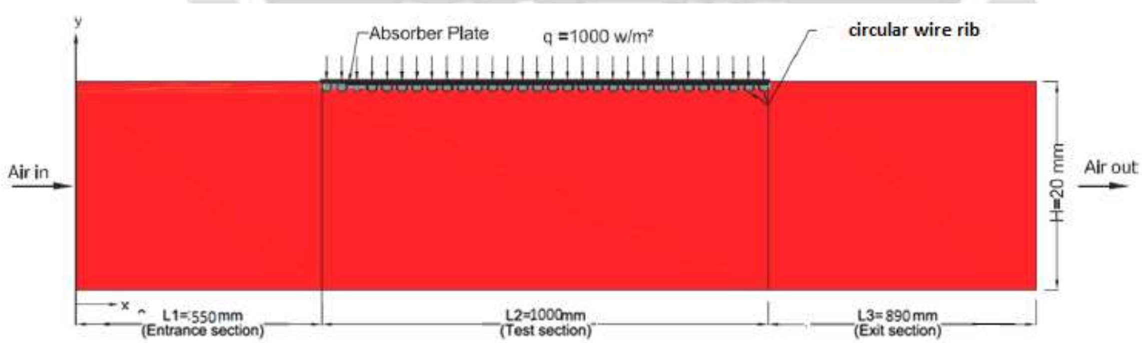


Fig.1. Showing the geometric dimension of the working model

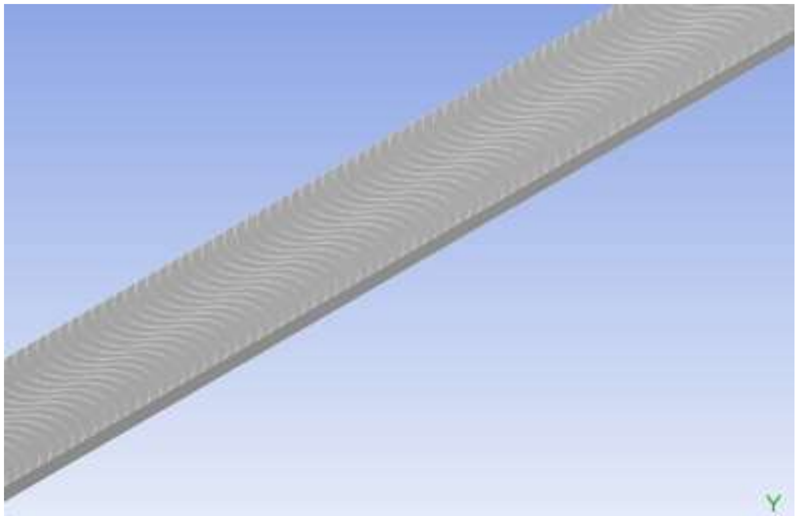


Fig. 2 Geometry of Broken ‘S’ shaped rib

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 of hydraulic diameter. The exit section is used after the test section in order to reduce the end effect in the test section. The top wall consists of a 0.5 mm thick absorber plate made up of aluminum. Artificial roughness in the form of small diameter galvanized iron (G.I) wires is considered at the underside of the top of the duct on the absorber plate to have roughened surface, running perpendicular to the flow direction while other sides are considered as smooth surfaces. A uniform heat flux of 1000 w/m^2 is considered for computational analysis.

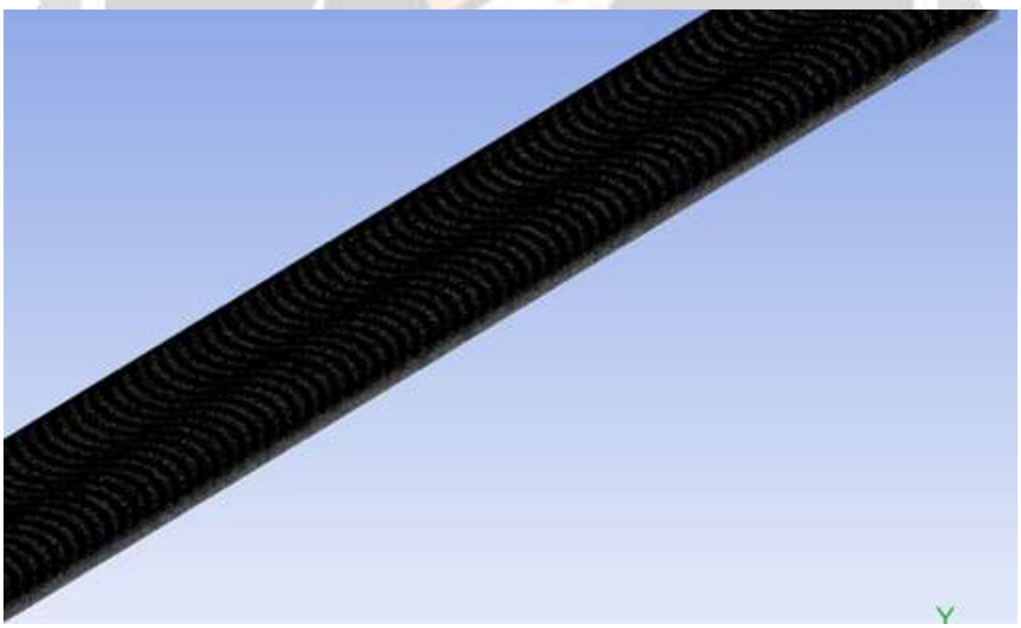


Fig.3 **Meshing of computational Domain for broken ‘S’ ribs Roughness**

A non-uniform mesh is shown in Fig.3Present mesh contained 191,061 quad cells with non-uniform quad grid of 0.21 mm cell size. This size is suitable to resolve the laminar sub-layer. For grid independence test, the number of cells is varied from 113,431 to 207,147 in five steps. It is found that after 191,0481 cells, further increase in cells has less than 1% variation in Nusselt number and friction factor value which is taken as criterion for grid independence.

In the present simulation governing equations of continuity, momentum and energy are solved by the finite volume method in the steady-state regime. The numerical method used in this study is a segregated solution algorithm with a finite volume-based technique. The governing equations are solved using the commercial CFD code, ANSYS Fluent 14.5. No-slip conditions for velocity in solid surfaces are assumed and the turbulence kinetic energy is set to zero on all solid walls. The top wall boundary condition is selected as constant heat flux of 1000 W/m^2 and bottom wall is assumed at adiabatic condition. A uniform air velocity is introduced at the inlet while a pressure outlet condition is applied at the exit. The Reynolds number varies from 2000 to 16000 at the inlet. The mean inlet velocity of the flow is calculated using Reynolds number. Constant velocity of air is assumed in the flow direction. The temperature of air inside the duct is also taken as 300 K at the beginning. At the exit, a pressure outlet boundary condition is specified with a fixed pressure of $1.013x10^5$ Pa.

III. RESULTS AND DISCUSSION

A. Heat Transfer Characteristics and Friction Factor Characteristics

Fig.4 shows the effect of Reynolds number on average Nusselt number for different values of relative roughness pitch (P/e) and fixed other parameter. The average Nusselt number is observed to increase with increase of Reynolds number due to the increase in turbulence intensity caused by increase in turbulence kinetic energy and turbulence dissipation rate.

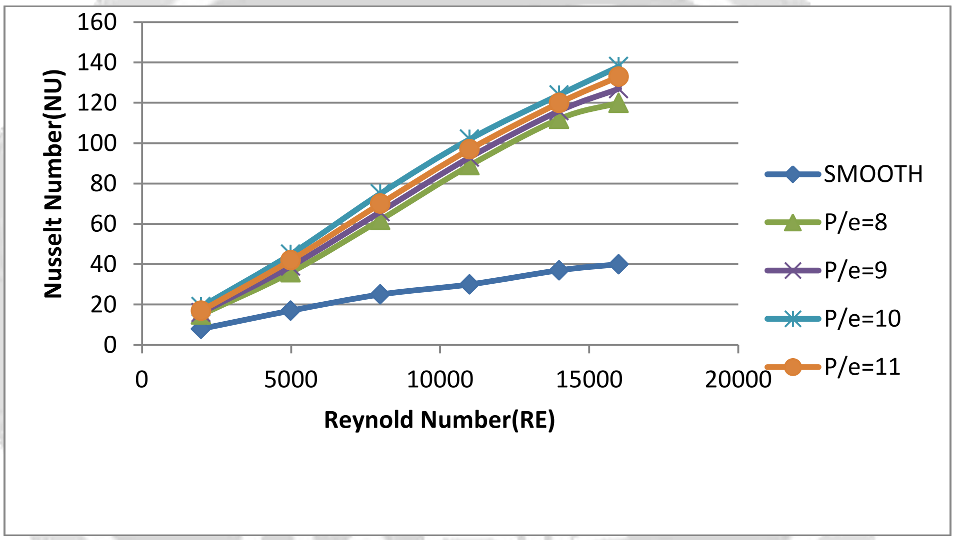
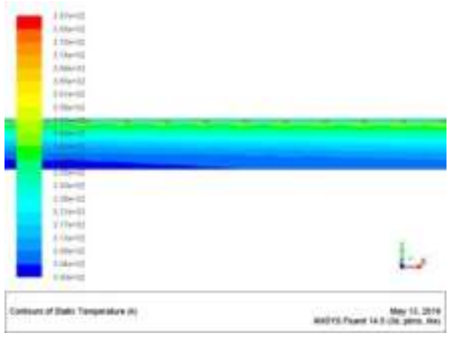
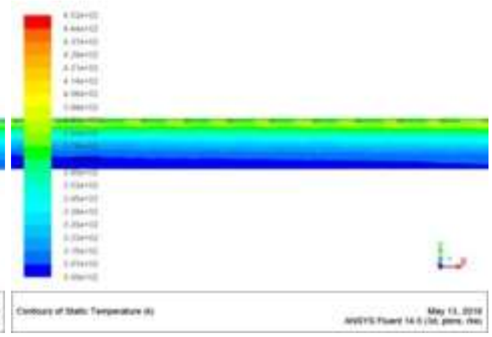


Fig. 4. Variation of Nusselt number with Reynolds number for different Values of relative roughness pitch (P/e).

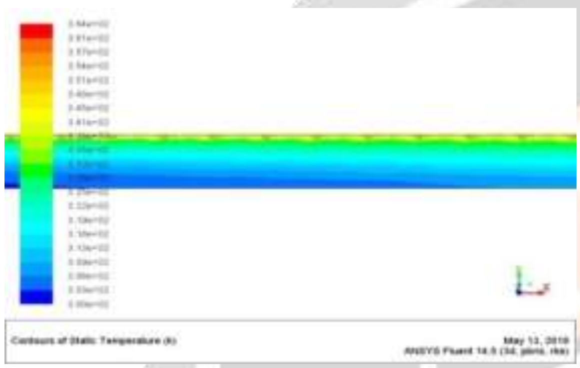
Effect of the relative roughness pitch (P/e) on heat transfer is also shown typically in Fig. 4. It can be seen that the enhancement in heat transfer of the roughened duct with respect to the smooth duct also increases with an increase in Reynolds number. It can also be seen that Nusselt number values increases with the increase in relative roughness pitch (P/e) of up to 10 and then decrease for a fixed value of other parameter. The roughened duct having broken 'S' shaped with relative roughness pitch (P/e) of 10 provides the highest Nusselt number at a Reynolds number of 16000. For circular rib the maximum enhancement of average Nusselt number is found to be 2.47 times that of smooth duct for relative roughness pitch (P/e) of 10 at a Reynolds number of 16000.The heat transfer phenomenon can be observed and described by the contour plot of turbulence intensity. The contour plot of turbulence intensity for broken 'S' shaped ribs is shown in Fig.5 (a, b and c).The intensities of turbulence are reduced at the flow field near the rib and wall and a high turbulence intensity region is found between the adjacent ribs close to the main flow which yields the strong influence of turbulence intensity on heat transfer enhancement.



(a)



(b)



(c)

Fig. 5 Contour plot of turbulent intensity for circular rib (a) Re=4000 (b) Re=8000 (c) Re=12000

Fig. 6 shows the effect of Reynolds number on average friction factor for different values of relative roughness pitch (P/e) and fixed value of other parameter. It is observed that the friction factor decreases with increase in Reynolds number because of the suppression of viscous sub-layer.

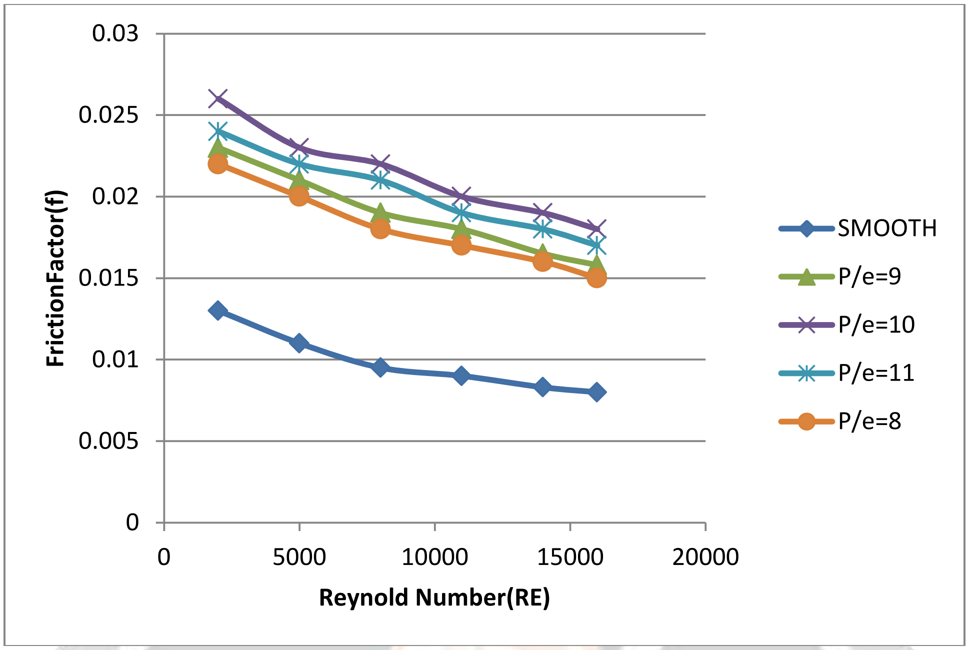


Fig. 6 Comparison between Friction factor and Reynolds number at different relative roughness (P/e)

Figure. 6 also shows that the friction factor decreases with the increasing values of the Reynolds number in all cases as expected because of the suppression of laminar sub-layer for fully developed turbulent flow in the duct. It can also be seen that friction factor values increase with the increase in relative roughness pitch (P/e) up to 10 and then decrease for fixed value of other, attributed to more interruptions in the flow path.

B. Thermo-Hydraulic Performance

It has also been observed from Figures 4 and 6 that the maximum values of Nusselt number and friction factor correspond to relative roughness pitch (P/e) of 10, thereby, meaning that an enhancement in heat transfer is accompanied by friction power penalty due to a corresponding increase in the friction factor. Therefore, it is essential to determine the effectiveness and usefulness of the roughness geometry in context of heat transfer enhancement and accompanied increased pumping losses. In order to achieve this objective, Webb and Eckert [23] proposed a thermo-hydraulic performance parameter ‘η’, which evaluates the enhancement in heat transfer of a roughened duct compared to that of the smooth duct for the same pumping power requirement and is defined as,

$$\text{Thermal enhancement factor} = \frac{{}^{\text{Nu}}/_{\text{Nu}_s}}{\left({}^{f}/_{f_s}\right)^{\frac{1}{3}}}$$

The value of this parameter higher than unity ensures that it is advantageous to use the roughened duct in comparison to smooth duct. The thermo-hydraulic parameter is also used to compare the performance of number of roughness arrangements to decide the best among these. The variation of thermo-hydraulic parameter as a function of Reynolds number for different values of relative roughness pitch (P/e) and investigated in this work has been shown in Fig. 7. For all values of relative roughness pitch (P/e), value of performance parameter is more than unity. Hence the performance of solar air heater roughened with broken ‘S’ shaped ribs is better as compared to smooth duct.

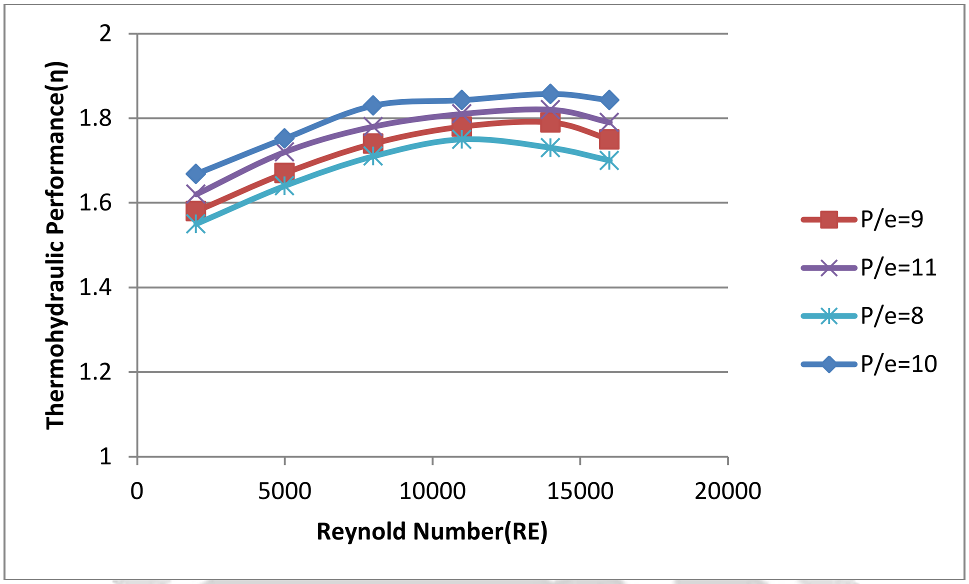


Figure No. 7 Thermo-hydraulic performance parameter as a function of Reynolds Number for different relative roughness pitch (P/e)

Conclusion:

The Numerical investigations were conducted on solar air heater duct roughened with broken 'S' shaped ribs. The following conclusions are drawn from the present study:

A 3-dimensional CFD analysis has been carried out to study heat transfer and fluid flow behavior in a rectangular duct of a solar air heater with one roughened wall having circular and broken 'S' rib roughness. The effect of Reynolds number and relative roughness pitch on the heat transfer coefficient and friction factor have been studied. In order to validate the present numerical model, results have been compared with available experimental results under similar flow conditions. CFD Investigation has been carried out in medium Reynolds number flow (Re = 2000–16,000). The following conclusions are drawn from present analysis:

1. The Renormalization-group (RNG) k-ε turbulence model predicted very close results to the experimental results, which yields confidence in the predictions done by CFD analysis in the present study. RNG k-ε turbulence model has been validated for smooth duct and grid independence test has also been conducted to check the variation with increasing number of cells.
2. The roughened duct having broken 'S' shaped rib with relative roughness pitch (P/e) of 10 provides the highest Nusselt number at a Reynolds number of 16000.
3. For rectangular rib the maximum enhancement of average Nusselt number is found to be 2.38 times that of smooth duct for relative roughness pitch (P/e) of 10 at a Reynolds number of 11000.
4. The roughened duct having broken 'S'shaped rib with relative roughness pitch (P/e) of 10 provides the highest friction factor at a Reynolds number of 3500.
5. For broken 'S' shaped rib the maximum enhancement of average friction factor is found to be 3.32 times that of smooth duct for relative roughness pitch (P/e) of 10.

It is found that the thermal hydraulic performance of relative roughness pitch (P/e) of 10 is maximum.

References

[1] Sharma AK and Thakur NS. "CFD based Fluid Flow and Heat Transfer Analysis of a V-Shaped Roughened Surface Solar Air Heater". International Journal of Engineering Science and Technology, 2012, 5, 2115-2121.

[2] Kaltschmitt M, Streicher W, Wiese A. Renewable energy: technology, economics and environment. 1st ed. New York: Springer; 2007.

[3] AtulLanjewar, J.L. Bhagoria, R.M. Sarviya, "Experimental study of augmented heat transfer and friction in solar air heater with different orientations of W-Rib roughness Experimental Thermal and Fluid Science 35 (2011) 986–995.

[4] Li X. Green energy: Basic concepts and fundamentals. 1st ed. New York: Springer; 2011.

[5] Maczulak A. Renewable energy: sources and methods. 1st ed. New York: Infobase Publishing; 2011

[6] Foster R, Ghassemi M, Cota A. Solar energy: renewable energy and the environment. New York: CRC Press Taylor & Francis group; 2010.
[7] Kalogirou S. Solar energy engineering: processes and systems. London: Academic Press; 2009.
[8] Yadav AS, Bhagoria JL, Renewable energy sources- An application guide. International Journal of Energy Science 2013; 3(2): 70-90.
[9] Sukhatme SP, Nayak JP. Solar energy. 3rd ed. New Delhi: Tata McGraw Hill; 2011.
[10] J.C. Han, P.R. Chandra, C.R Alexandra, "Heat transfer and friction behaviors in rectangular channels with varying number of ribbed walls" International Journal of Heat and Mass Transfer 46 (2003) 481–495.

[11] Han JC. Heat transfer and friction in channels with two opposite rib-roughened walls. J. Heat Transfer 1984; 106 (4): 774-81.
[12] Bertram RW. Testing and standards for thermal solar collectors, Solar Energy Conversion II (Selected lectures from the 1980 International Symposium on Solar Energy Utilization, London), Edited by A. F. Janzen and R. K. Swartman. New York: Pergamon Press; 1980, pp. 73-91.
[13] Bhushan B, Singh, A review on methodology of artificial roughness used in duct of solar air heaters. Energy 2010; 35: 202–12.
[14] Quaschning V. Understanding renewable energy systems. 3rd ed. London: Earthscan; 2005.
[15] Garg HP, Gupta CL. Flat plate collector experimental studies and design data for India. Proc. ISEC Congress New Delhi 1978; 733-738.
[16] Verma SK, Prasad BN. Investigation for the optimal thermo-hydraulic performance of artificially roughened solar air heaters. Renewable Energy 2000; 20: 19–36.
[17] Han JC, Chandra PR, Lau SC. Local heat/mass transfer distributions around sharp 180 degree turns in two-pass smooth and rib roughened channels. J. Heat Transfer 1988; 110 (February): 91–98.
[18] Han JC, Glicksman LR, Rohsenow WM. 1978. An investigation of heat transfer and friction for rib-roughened surfaces. International Journal of Heat and Mass Transfer 1978; 21(8):1143–56.
[19] Gupta D, Solanki SC, Saini JS. Heat and fluid flow in rectangular solar air heater ducts having transverse rib roughness on absorber plates. Solar Energy 1993; 51(1): 31–7.
[20] Zhang YM, Gu WZ, Han JC. Heat transfer and friction in rectangular channels with ribbed or ribbed-grooved walls. J. Heat Transfer 1994; 116 (1): 58–65.
[21] Park JS, Han JC, Huang Y, Ou S. Heat transfer performance comparisons of five different rectangular channels with parallel angled ribs. International Journal of Heat and Mass Transfer 1992;35(11):2891-903..
[22] Taslim ME, Lengkong. 45 degree staggered rib heat transfer coefficient measurements in a square channel. ASME J. Turbomach. 1998; 120: 571– 580.
[23] Karwa R, Solanki SC, Saini JS. Heat transfer coefficient and friction factor correlations for the transitional flow regime in rib-roughened rectangular ducts. Int J Heat Mass Transfer 1999; 42: 1597–15.
[24] Momin AME, Saini JS, Solanki SC. Heat transfer and friction in solar air heater duct with v-shaped rib roughness on absorber plate. Int J Heat Mass Transfer 2002; 45: 3383–96.
[25] Anil SinghYadav, J.L.Bhagoria, "Heat transfer and fluid flow analysis of solar air heater: A review of CFD approach", Renewable and Sustainable Energy Reviews 23 (2013) 60–79.

[26] Han JC, Zhang YM. High performance heat transfer ducts with parallel broken and V-shaped broken ribs. International Journal of Heat and Mass Transfer 1992;35(2):513-23.
[27] Chaube A, Sahoo PK, Solanki SC. Analysis of heat transfer augmentation and flow characteristics due to rib roughness over absorber plate of a solar air heater. Renewable Energy 2006; 31: 317-31.
[28] Ryu DN, Choi DH, Patel VC. Analysis of turbulent flow in channels roughened by two-dimensional ribs and three-dimensional blocks. Part I: Resistance. International Journal of Heat and Fluid Flow 2007; 28: 1098–1111.
[29] Layek A, Saini JS, Solanki SC. Heat transfer and friction characteristics for artificially roughened ducts with compound turbulators. International Journal of Heat and Mass Transfer 2007; 50: 4845–54.
[30] Kamali R, Binesh AR. The importance of rib shape effects on the local heat transfer and flow friction characteristics of square ducts with ribbed internal Surfaces. International Communications in Heat and Mass Transfer 2008; 35: 1032–1040.
[31] Saini RP, Verma J. Heat transfer and friction factor correlations for a duct having dimple-shaped artificial roughness for solar air heaters. Energy 2008; 33: 1277–87.
[32] Saini SK, Saini RP. Development of correlations for Nusselt number and friction factor for solar air heater with roughened duct having arc-shaped wire as artificial roughness. Solar Energy 2008; 82: 1118–30.

[33] Chaudhary S, Varun, Chauhan MK. Heat transfer and friction factor characteristics using continuous M shape ribs turbulators at different orientation on absorber plate solar air heater. Int J energy and environment 2012; 3(1):33-48.

[34] Bopche SB, Tandale MS. Experimental investigations on heat transfer and frictional characteristics of a turbulator roughened solar air heater duct. Int J Heat and Mass Transfer 2009; 52: 2834–48.

[35] Lanjewar A, Bhagoria JL, Sarviya RM. Heat transfer and friction in solar air heater duct with W-shaped rib roughness on absorber plate. Energy 2011;36:4531-41.

[36] Hans VS, Saini RP, Saini JS. Heat transfer and friction factor correlations for a solar air heater duct roughened artificially with multipleV-ribs. Sol Energy 2010;84:898–91

[37] Singh S, Chander S, Saini JS. Heat transfer and friction factor correlations of solar air heater ducts artificially roughened with discrete V-down ribs. Energy 2011; 36: 5053-64.

[38] Launder BE and Spalding DB. Lectures in mathematical models of turbulence. London, England: Academic Press; 1972.