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Numerical Investigation of Heat Transfer and Friction Characteristics of Solar Air Heater Duct using Broken Double Arc Shaped Ribs Roughness

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Abstract— Solar air heating is a developing renewable energy technology used for various applications, presently solar air heaters have numerous applications in space heating, cooling and ventilation. These solar air heaters have low heat transfer efficiency which can be improved using geometrical modifications like optimizing duct geometry or adding artificial roughness. The use of artificial roughness on the underside of the absorber plate is an effective and economic way to improve the thermal performance of a solar air heater. Several experimental investigations, involving different types of roughness elements, have been carried out to improve the heat transfer from the absorber plate to air flowing in solar air heaters. In this paper the CFD analysis on heat transfer and friction in rectangular ducts roughened with broken double arc-rib has been presented. The relative gap width (g/e) is varied from 0.5 to 2.5 and other parameter are constant. The effects of gap width (g/e) on Nusselt number, friction factor and thermo-hydraulic performance parameter have been discussed and results compared with smooth duct under similar conditions. It is found that the maximum heat transfer and friction characteristic at a gap width of 1.0.

Keywords: Solar air heater, Nusselt number, Heat transfer, Relative gap width

I. INTRODUCTION

It has been observed that the heat transfer coefficient between the absorber plate and working fluid of solar air heater is generally low. It is attributed to the formation of a very thin boundary layer at the absorber plate surface commonly known as viscous sub-layer. This convective heat transfer coefficient can be increased by providing the artificial roughness on the heat transferring surface. It has been found that the artificial roughness applied on the heat transferring surface breaks the viscous sub-layer, which reduces thermal resistance and promotes turbulence in a region close to artificially roughened surface. Although the application of artificial roughness in the duct of a conventional solar air heater has been shown to be an efficient method of enhancement of thermal efficiency of solar air heater. 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. 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 whether the channel roughened with ribs of different shape can improve the heat transfer rate. Hence many techniques have been investigated on enhancement of heat transfer rate and decrease the size and cost of the involving equipment especially in heat. Heat transfer enhancement by turbulence promoters either in the form of surface roughness tends primarily to increase the heat transfer coefficient. The use of rib roughness in the form of repeated ribs has been found to be an efficient method of enhancing the heat transfer to fluid flowing in the duct. Ribs improve the heat transfer by interrupting the wall sub layer.

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 arc shaped ribs roughness on the absorber plate at the underside of the top of the duct while other sides are considered as smooth surfaces.

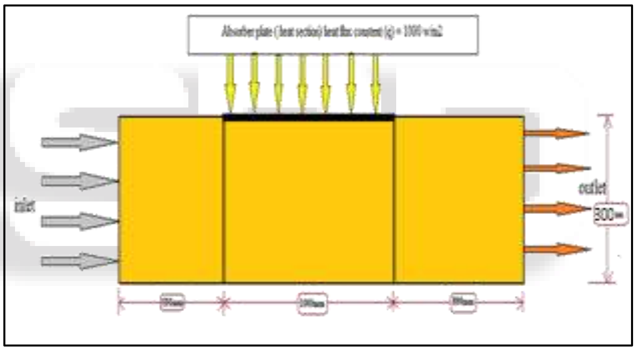


Fig. 1: Showing the geometric dimension of the working model

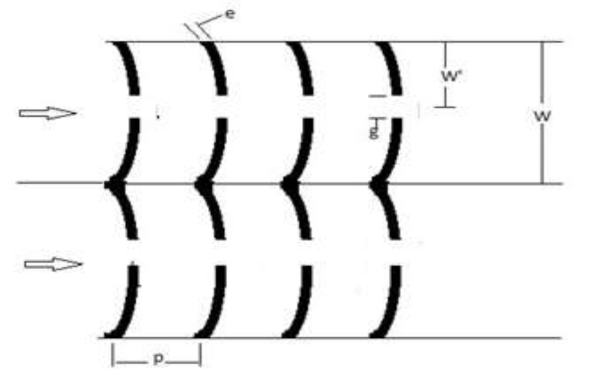


Fig. 2: Schematic diagram of broken double arc rib

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/m2 is considered for computational analysis.

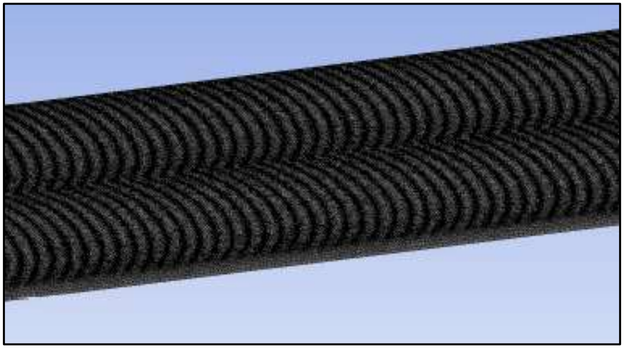


Fig. 3: Meshing of computational Domain for double arc-rib with gap

A non-uniform mesh is shown in Fig.3 present 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/m2 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 1.013x105 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 gap width (g/e) and fixed other parameters. 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.

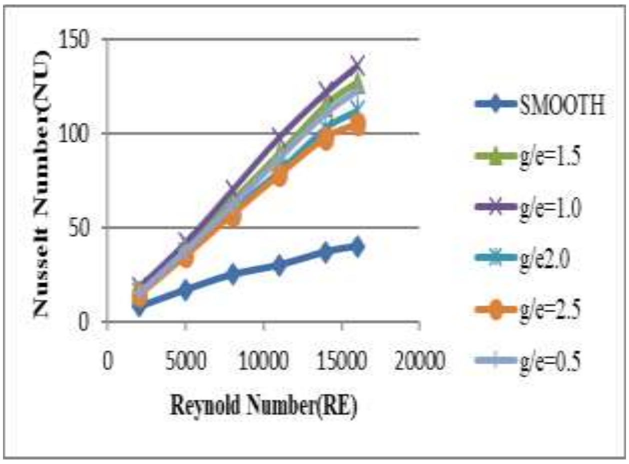
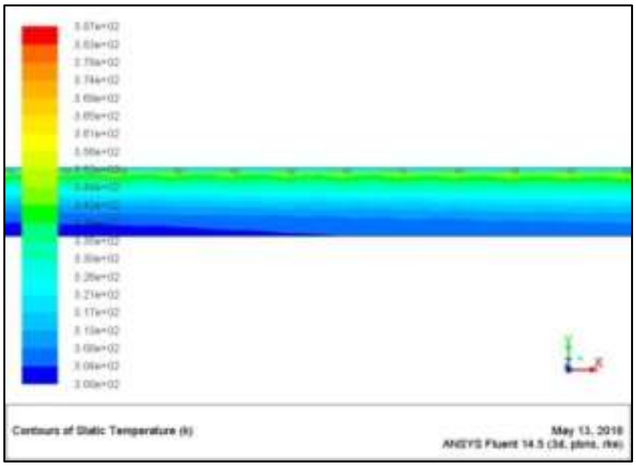


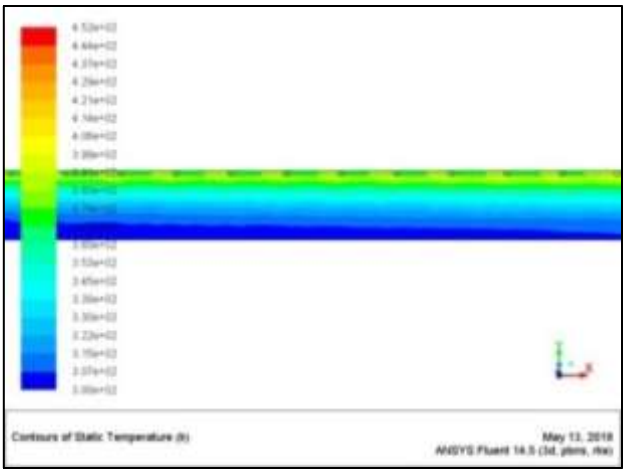
Fig. 4: Variation of Nusselt number with Reynolds number for different Values of relative gap width (g/e).

Effect of the relative gap width (g/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 gap width (g/e) of up to 1.0 and then decrease for a fixed value of roughness pitch (P). The roughened duct having gap in double arc shaped with relative gap width (g/e) of 1.0 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 2.51 times that of smooth duct for relative gap width (g/e) of 1.0 at a Reynolds number of 16000.

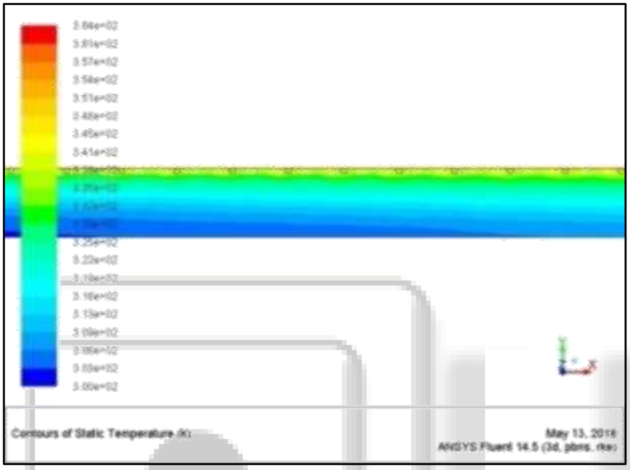
The heat transfer phenomenon can be observed and described by the contour plot of turbulence intensity. The contour plot of turbulence intensity for broken double arc shaped ribs is shown in Fig.5 (a, b and c).The intensities of turbulence are reduced at the flow field near the rib and wall and a high 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 gap width (g/e) and fixed value of roughness pitch. It is observed that the friction factor decreases with increase in Reynolds number because of the suppression of viscous sub-layer.

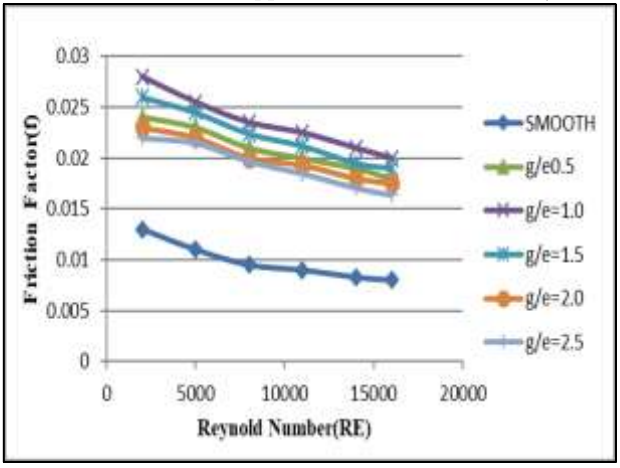


Fig. 6: Variation of Friction factor and Reynolds number at different gap width (g/e)

Fig. 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 gap width (g/e) up to 1.0 and then decrease for fixed value of roughness pitch, 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 gap width of 1.0, 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 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,

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

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 gap width (g/e) and investigated in this work has been shown in Fig. 7. For all values of relative gap widths, value of performance parameter is more than unity. Hence the performance of solar air heater roughened with broken double arc shaped ribs is better as compared to smooth duct.

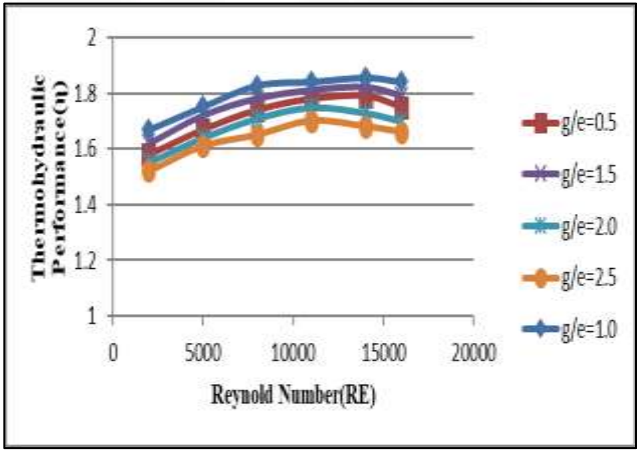


Fig. 7: Thermo-hydraulic performance parameter as a function of Reynolds Number for different relative gap width (g/e)

IV. CONCLUSION

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 double arc-rib rib roughness. The effect of Reynolds number and relative roughness pitch on the heat transfer coefficient and friction factor have been studied. The following conclusions are drawn from present analysis:

1) The roughened duct having broken double arc shaped rib with relative gap width of 1.0 provides the highest Nusselt number at a Reynolds number of 16000.
2) For rectangular rib the maximum enhancement of average Nusselt number is found to be 2.52 times that of smooth duct for relative gap width of 1.0 at a Reynolds number of 11000.
3) The roughened duct having broken double arc- rib with relative gap width of 1.0 provides the highest friction factor at a Reynolds number of 3500.
4) For broken double arc-rib the maximum enhancement of average friction factor is found to be 3.52 times that of smooth duct for relative gap width of 1.0 at a Reynolds number of 3800.
5) It is found that the thermal hydraulic performance of relative gap width of 1.0 is maximum.

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