CFD INVESTIGATIONS OF CIRCULAR AND SQUARE SECTIONED RIB FITTED SOLAR AIR HEATER

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ABSTRACT

A 2-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 square 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 = 3800–18,000). It is found that the thermal enhancement factor values vary between 1.42 and 1.74 for circular rib and between 1.51 and 1.762 for square rib.

Keywords: Ansys, CFD, Fluent,Heat Transfer ,Rib.

I. INTRODUCTION

Due to a growing world population and increasing modernization, global energy demand is projected to more than double during the first half of the twenty-first century and to more than triple by the end of the century. Presently, the world's population is nearly 7 billion, and projections are for a global population approaching 10 billion by midcentury. Future energy demands can only be met by introducing an increasing percentage of alternative fuels. Incremental improvements in existing energy networks will be inadequate to meet this growing energy demand. Due to dwindling reserves and ever-growing concerns over the impact of burning carbon fuels on global climate change, fossil fuel sources cannot be exploited as in the past.

A solar collector is a special kind of heat exchanger, which collects solar radiant energy, and transfers it to a fluid- usually water or air. It converts solar radiation to thermal energy of fluid and delivers the heated fluid for use.A conventional solar air heater generally consists of an absorber plate with a parallel plate below forming a passage of high aspect ratio through which the air to be heated flows. However, the value of the heat transfer coefficient between the absorber plate and air is low and this result in lower efficiency. For this reason, the surfaces are sometimes roughened or longitudinal fins are provided in the airflow passage. A roughness element has been used to improve the heat transfer coefficient by creating turbulence in the flow. However, it would also result in increase in friction losses and hence greater power requirements for pumping air through the duct. In order to keep the friction losses at a low level, the turbulence must be created only in the region very close to the duct surface, i.e. in laminar sub layer. A further disadvantage associated with the use of solar air heater is that

large volume of fluid has to be handled. As a result, the pressure becomes an important parameter and has to be kept in prescribed limits. Air density is 0.001th of water, and thus for the same energy input, air can be heated covering much greater volumetric flow rate [1-5].

Solar air heaters, because of their inherent simplicity, are cheap and most widely used as collection device. 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. In order to attain higher heat transfer coefficient, the laminar sub-layer formed in the vicinity of the absorber plate is broken and the flow at the heat-transferring surface is made turbulent by introducing artificial roughness on the surface. Various investigators have studied different types of roughness geometries and their arrangements.

The heat transfer between the absorber surface (heat transfer surface) of solar air heater and flowing air can be improved by either increasing the heat transfer surface area using extended and corrugated surfaces without enhancing heat transfer coefficient or by increasing heat transfer coefficient using the turbulence promoters in the form of artificial roughness on absorber surface. The artificial roughness on absorber surface may be created, either by roughening the surface randomly with a sand grain/sand blasting or by use of regular geometric roughness. It is well known that in a turbulent flow a laminar/viscous sub-layer exists in addition to the turbulent core. The artificial roughness on heat transfer surface breaks up the laminar boundary layer of turbulent flow and makes the flow turbulent adjacent to the wall. The artificial roughness that results in the desirable increase in the heat transfer also results in an undesirable increase in the pressure drop due to the increased friction; thus the design of the flow duct and absorber surface of solar air heaters should, therefore, be executed with the objectives of high heat transfer rates and low friction losses [6-13].Han et al. [14] experimentally investigated the effects of rib shape, angle of attack and pitch-to-rib height ratio on friction factor and heat transfer coefficient. Verma and Prasad [15] developed the relations to calculate the average friction factor and Stanton number for artificial roughness of absorber plate by small diameter protrusion wire. Park et al. [16] presented the results of heat transfer and friction factor data measured in five short rectangular channels with turbulence promoters. Author investigated the combined effects of the channel aspect ratio, rib angle of attack, and flow Reynolds number on heat transfer and pressure drop in rectangular channels with two opposite ribbed walls. The channel aspect ratio (width to height, W/H, ribs on side W) varied from ¼ to ½ , to 1, 2 and 4, while the corresponding rib angles of attack were 90°, 60°, 45°, and 30°, respectively. The Reynolds number range was 10,000–60,000. The results suggested that the narrow aspect ratio channels (W/H < 1) gave much better heat transfer performance than the wide aspect ratio channels (W/H > 1). For the square channel (W/H = 1), the 60°/45°angled ribs provided the best heat transfer performance. For the narrow aspect ratio channel (W/H=1/4 or 1/2), the 45°/60°angled ribs were recommended while the 30°/45°angled ribs were better for wide aspect ratio channels (W/H = 4 or 2).Taslim and Lengkong [17] experimentally investigated the heat transfer and friction in channel roughened with angled V-shaped and discrete ribs on two opposite walls for Reynolds number ranging from 5,000 to 30,000. The results showed that the 90° transverse ribs produced the lowest heat transfer performance. The 45° angled V-shaped ribs produced the highest heat transfer performance in comparison to other rib configurations. For V-shaped ribs facing downstream of flow, the one with lowest blockage ratio had better heat removal rate. The discrete ribs also produced better performance in comparison to the transverse

ribs.Karwa et al. [18] experimented on integral chamfered rib roughness on the heated wall and reported that the chamfer angle of 15° gives the maximum heat transfer. Most of the investigations carried out so far have been with ducts of circular cross-section or of rectangular section having two opposite roughened walls and with all the four walls heated. It needs to be mentioned that for the application of this concept of enhancement of heat transfer in the case of solar air heaters, roughness elements have to be considered only on one broad wall, which is the only heated wall. This application makes the fluid flow and heat transfer characteristics distinctly different from those found in the case of two roughened walls and four heated wall ducts. In solar air heaters, only one wall of the rectangular air passage is subjected to uniform heat flux (isolation) while the remaining three walls are insulated. It has recently been proposed by several investigators that providing artificial roughness on the absorber plate could substantially enhance the heat transfer capability of a solar air heater.Chaube et al. [19] performed a computational analysis of heat transfer augmentation and flow characteristics due to artificial roughness in the form of ribs on a broad heated wall of a rectangular duct of a solar air heater for turbulent flow (Reynolds number range of 3,000–20,000). A 2-D analysis of heat transfer and fluid flow through an artificially roughened solar air heater was carried out using commercially available CFD software, FLUENT 6.1. Shear stress transport k-ω turbulence model was selected by comparing the predictions of different turbulence models with experimental results available in the literature. The results obtained from CFD approach were reported to be closer to the experimental results. Based on the computational results following conclusions were drawn: 1. It was observed that the 2D analysis model itself yields results which were closer to the experimental ones as compared to 3D models. The 3D model required much higher memory and computational time compared to 2D ones. Hence, it was sufficient to employ a simpler 2D model which was more economical with the memory and computational time requirements. 2. In the inter-rib region, the model predicted well near the central high heat transfer area but it under predicted around ribs. 3. The peaks of local heat transfer were found at the reattachment points. 4. The turbulence intensity was found maximum at the peak of the local heat transfer coefficient in the inter-rib regions. 5. The peaks of the local heat transfer coefficient and the reattachment lengths decreased stream wise in the successive inter-rib regions up to 3–4 ribs and then the distribution became periodic with peak values and the reattachment lengths remained unchanged. 6. The highest heat transfer was achieved with chamfered ribs but the best performance index was found with rectangular rib of size 3x5 mm.Ryu et al. [20] have studied numerically friction and heat transfer in the flow in rib-roughened channels with one smooth wall .Reynolds-averaged Navier–Stokes equations, coupled with the k–ω turbulence model with a special near-wall treatment, were solved by a finite-volume method. The roughness elements cross-section was square, triangle, semicircle and cosine wave. The roughness function was found to be a function of the rib shape and pitch ratio but was independent of the absolute rib size.Layek et al. [21] studied and found that the artificial roughness in the form of chamfered rib groove on the absorber plate results in considerable enhancement of heat transfer. This enhancement is, however, accompanied by a substantial increase in the friction factor. It is, therefore, desirable to select the roughness geometry such that the heat transfer coefficient is maximized while keeping the friction losses at the minimum possible value. Considering the heat transfer and friction characteristics can fulfill this requirement of the collector simultaneously.Prasad et al. [22] determined the optimal thermo hydraulic performance of three sides artificially roughened solar air heater of high aspect ratio. For a particular set of values of roughness and flow parameters the optimal thermo

hydraulic performance condition always corresponds to an optimal value of roughness Reynolds number.Singh et al. [23] presents thermo-hydraulic performance comparison of rib roughness under investigation, 'V- down ribs with gap' and similar reported rib roughness geometries used in solar air heater duct. Five rib roughened plates having relative roughness pitch of 4, 6, 8, 10 and 12 have been tested. The Nusselt number and friction factor were found to be highest for relative roughness pitch of 8. Maximum enhancement in Nusselt number and friction factor has been found to be 2.70 and 2.86, respectively. Thermo-hydraulic performance parameter ranged from 1.27 to 1.93. Thermo-hydraulic comparison with similar rib geometries show that the present roughness geometry performs better for Reynolds number range of 3000–12,000.

This work basically describes the characteristics of the heat transfer and friction in a square duct where circular-shaped ribs are placed transversely to the main stream direction on one wall. The objectives of this paper are to fulfill two aspects; i.e., to provide detailed information of average heat transfer and flow friction characteristics in ribbed passages, and to compare the thermal performance of different configurations of ribbed duct. The specific objectives of the paper are to develop a computational model and solve the governing equations on the computational domain and finally compare the thermal performance of different configurations of ribbed duct.

II. CFD ANALYSIS

The 2-dimensional solution domain used for CFD analysis has been generated in ANSYS version 12.1 (workbench mode) as shown in Fig. 1. The solution domain is a horizontal duct with circular and square rib roughness on the absorber plate at the underside of the top of the duct while other sides are considered as smooth surfaces.

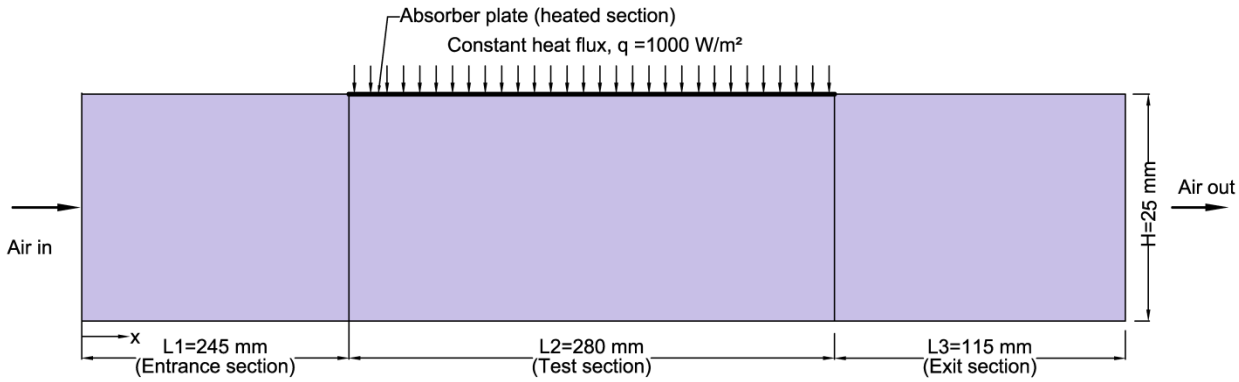

Figure 1 Computational Domain

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 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

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computational analysis. Meshing of the domain is done using ANSYS ICEM CFD V12.1 software (Fig. 2).

Figure 2 Meshing of computational Domain

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 12.1. A second-order upwind scheme is chosen for energy and momentum equations. The SIMPLE algorithm (semi-implicit method for pressure linked equations) is chosen as scheme to couple pressure and velocity. The convergence criteria of 10^{-3} for the residuals of the continuity equation, 10^{-6} for the residuals of the velocity components and 10^{-6} for the residuals of the energy are assumed. A uniform air velocity is introduced at the inlet while a pressure outlet condition is applied at the outlet. Adiabatic boundary condition has been implemented over the bottom duct wall while constant heat flux condition is applied to the upper duct wall of test section.

III. RESULTS AND DISCUSSION

The effects of relative roughness height and Reynolds number 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 ducts operating under similar operating conditions to discuss the enhancement in heat transfer and friction factor on account of artificial roughness.

Fig. 3 shows the effect of Reynolds number on average Nusselt number for different values of relative roughness height (e/D) and fixed value of roughness pitch (P). 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.

Effect of the relative roughness height (e/d) on heat transfer is also shown typically in Fig. 3. 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 height (e/d) for fixed value of roughness pitch (P). This is due to the fact that heat transfer coefficient is low at the leading edge of the rib and high at the trailing edge. Higher relative roughness height

produced more reattachment of free shear layer which creates the strong secondary flow.

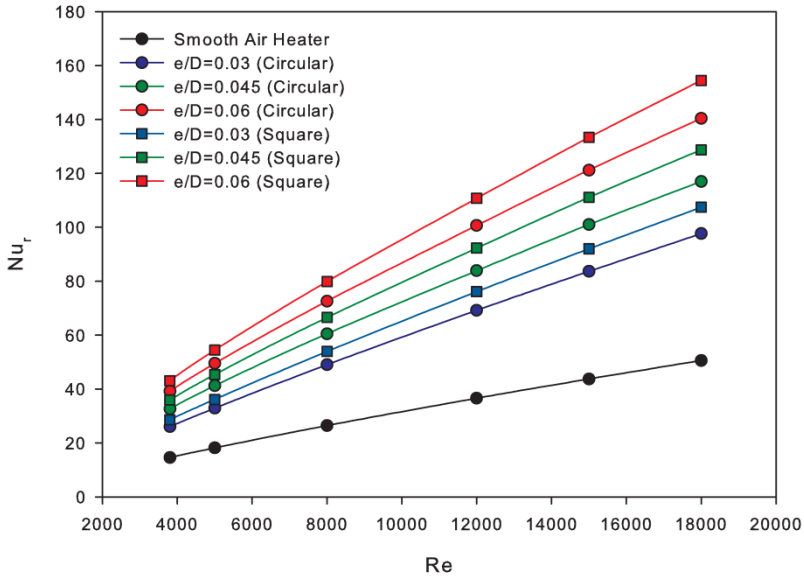

Figure 3Nusselt number Vs. Reynolds number

The heat transfer phenomenon can be observed and described by the contour plot of turbulence intensity. The contour plot of turbulence intensity for circular rib is shown in Fig. 4 (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.

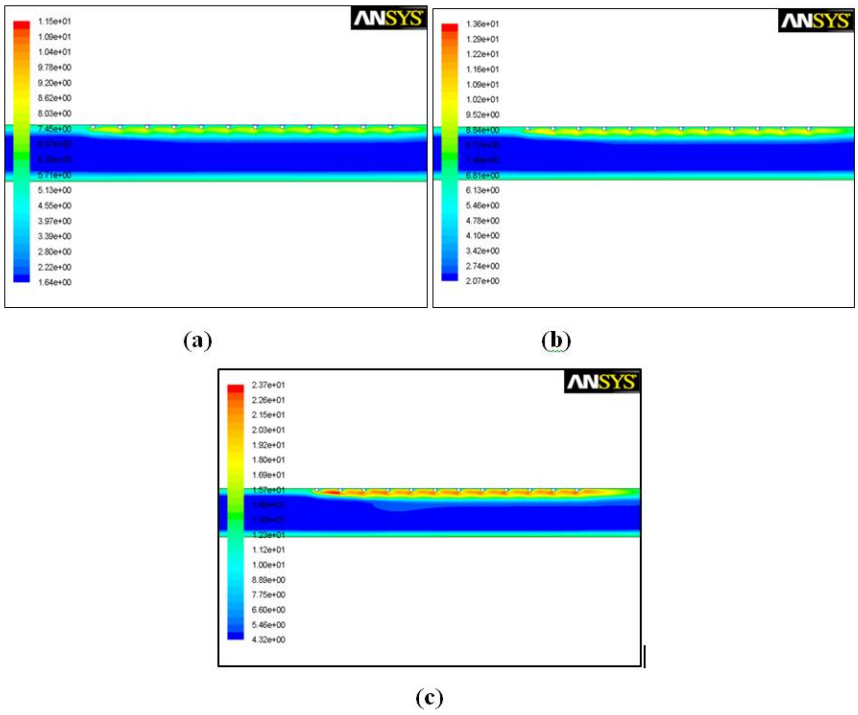

Figure 4 Contour plot of turbulent intensity for circular rib (a) Re=3800 (b) Re=5000 (c) Re=12000

IV. CONCLUSIONS

The following conclusions are drawned from theCFD 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 circular rib with relative roughness height of 0.06 provides the highest Nusselt number at a Reynolds number of 18000. The roughened duct having square rib with relative roughness height of 0.06 provides the highest Nusselt number at a Reynolds number of 18000.
3. It is found that the thermal enhancement factor values vary between 1.42 and 1.74 for circular rib and between 1.51 and 1.762 for square rib.

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