Analysis of Heat Transfer and Friction behaviors in Rectangular Duct with Two Different Ribs

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Abstract- A computational investigation was carried out to determine heat transfer and friction factors for turbulent flow through rectangular ducts with reveres triangular forms at uniformly varied heights, reveres pentagonal shapes at the same height, and reveres pentagonal shapes at uniformly different heights. ANSYS FLUENT 14.5, a commercial finite volume program, is utilized to Examine and illustrate the kind of flow through the ribbed duct. Air was the fluid in the duct, and friction and heat transfer by calculating the duct's total heat transfer coefficients, factors were ascertained. To achieve completely formed circumstances at the entry and escape. Various short ribs were ribbed into the walls of the air side. The duct's opposite wall, which was adiabatic, had good insulation. The results are given dimensionless regarding Reynolds numbers functions friction factors and Nusselt numbers. It was shown that, as compared to a smooth duct, reveres pentagonal rib form at constant heights and at continuously changing heights improve heat transfer and pressure drop.

Keywords: CFD, Friction Factor, forced convection, Heat transfer, Internal flow, Pressure Drop

INTRODUCTION

Rectangular ducts are frequently seen in heat transfer devices, such as nuclear reactor cooling channels, gas turbine cooling systems, compact heat exchangers, and combustion chamber cooling channels. forced turbulent convection of heat in a rectangle or square duct is among the basic issues with fluid mechanics and thermal physics. Ever since the invention of electrical gadgets and computers, a lot has changed in terms of technology. Smaller, faster computers have facilitated the creation of quicker, denser, and more heat fluxes being generated at the chip and package levels as a result of smaller circuit technology. Over the years, there have been notable developments in the use of air conditioning methods to control higher heat fluxes. Due to its affordability and ease of integration, air cooling remains the most popular technique for cooling electronic components. As turbulence promoters, repeated ribs or tabulators have been employed to increase heat transfer to the coolant flow in a channel. These roughness components rupture the flow's laminar sublayer. Both the pressure drop and the heat transfer are improved, which is a crucial factor to consider when evaluating the overall effectiveness of these flows.

Research has been conducted to predict the effects of wall thickness on heat transfer and friction characteristics. In applications like cooling gas turbine air-foils, rib tabulators are frequently cast on two separate sides of the cooling channels because heat transfer happens from the inner walls of the pressure vessel and the suction sides of the blade.

However, sometimes rib tabulators are positioned on one or all four sides of the cooling channels. The effects of improved internal cooling in channels with one, two, three, or all four rib-roughened walls may be used in various applications, such as electronic devices, heat exchangers, and nuclear reactors, even though turbine blade internal cooling has been the subject of much research in the past. The four-ribbed wall channel results are also utilised to support earlier hypotheses developed in the development of semi-empirical relationships between the roughness functions for heat transfer and friction. M.Y. Ibrahim, J.S. Park, and J.C. Han.[1]- Figure 5.1 illustrates the geometry of the cooling channel examined in this work; square ribs roughened the channel. The aspect ratio of the channel (AR = W/H) was 2.0 and 34 mm was the hydraulic diameter (Dh). Ten ribs in all were arranged on a single wall within the computational domain. The ratio of pitch to hydraulic diameter (p/Dh) was 10.0, while the ratio of rib height to hydraulic diameter (h/Dh) was 0.047. like that employed by C.G. Spezial, S. Sarkar, T.B. Gatski, and Han et al. [2]- The Reynolds stress model and the Speziale–Sarkar–Gatski (SSG) pressure–strain model was used to analyse turbulence. By taking into account the invariance with dynamical systems, the SSG model for the pressure–strain correlation terms in the transport equations for the Reynolds stress components was created. It was anticipated that the SSG Reynolds stress model would yield precise forecasts, particularly for flows with a significant streamline curvature. Previous research by M.E. Taslim and S.D. Spring [3] demonstrated that when the rib cross-section is altered from the area averaged Nusselt transforms from a square to a rectangle (that is, the rib width-to-height ratio (e/h) rises or falls from unity). the number falls between 5000 and 50,000 Reynolds numbers. For the isosceles triangular rib (Case 2), the heat transfer coefficient is observed to rise at Reynolds values between 20,000 and 60,000 as the rib width-to-rib height ratio increases from 1.0 to 2.0. Heat transfer improvements with these triangular ribs were found to be greater than those with rectangular ribs across the full range of measured rib width-to-height ratios at Reynolds numbers ranging from 4000 to 16,000. This was demonstrated by M.E. Taslim and S.D. Spring [4]. B. Sunden and C.O. Olsson [5] evaluated two radiator tubes with ribs and airflow. The axially averaged heat transfer coefficient along the tube length is provided by the air heat transfer statistics, which were collected with a constant wall temperature. In laminar and turbulent zones, the improved tubes had larger friction factors than the smooth tubes. Nonetheless, the friction factors tended to converge and get closer to the smooth tube value as the Reynolds number dropped in the purely laminar area. Additionally, when the friction factor grew, the Reynolds number for the laminar-turbulent transition dropped. Colburn j factors tended to converge at low Reynolds numbers and approximated the smooth tube value, much like the friction behavior.W. M. Rohsenow, L. R. Glicksman, and J. C. Han[6]- based on the use of an analogy for heat-momentum transmission and the rule of wall similarity. The influence of geometrically non-similar characteristics has been taken into account to derive broad correlations between friction and heat transport.

The relationships between friction and heat transport accord well with those of other researchers. Despite this, disparities become apparent when correlations are compared to dams that have been lowered using the Hall-type transformation. According to a performance comparison, sand-grain roughness or repeated-rib roughness with a 900-flow attack angle or both provide less heat transfer for the same fraction of friction power. Murata Akira and Mochizuki Sadanari[7]- Using the large eddy simulation with a Lagrangian dynamic sub-grid-scale model, the effect of rib orientation and channel rotation on heat transport in a two-pass square channel with 1800 abrupt turns was numerically explored. The sharp-turn-produced flow field was the dominant factor in the enhanced heat transfer that occurred both within and after the turn in the stationary state. The straight pass's pressure surface experienced a decline in heat transmission while the rotation was at its greatest speed. M. Schnieder, R. Poser, B. Weigand, and M. Amro [8]- As a model of a leading-edge cooling channel for a gas turbine blade, the current study examines the internal cooling in a triangle channel with a rounded edge. Heat transport is measured using a transient liquid crystal technique. A variety of innovative 3D rib topologies are disclosed with experimental results for Reynolds numbers ranging from 50 000 to 200 000. According to the testing findings, 600 ribs often offer more heat transfer increases than 45 deg. ribs. But for the 600 ribs, this leads to incredibly high friction coefficients. Chang, G.F. Hong, K.F. Chiang, and T.-M. Liou [9]- A unique roughness for heat transfer enhancement (HTE) with deeper scales and V-shaped ribs is developed. In this particular investigation, triangular (isosceles), wedge (right-triangular), and rectangular rib cross-sections were employed. We propose two types of rib arrangements: in-line and staggered arrays. A rectangular channel with an aspect ratio of AR=15, a height of H=20 mm, a single rib height of e=6 mm, and a rib pitch of P=40 mm is measured. The flow rate according to the Reynolds numbers is based on the inlet hydraulic diameter of the channel in a range of 4000 to 16,000.

As stated earlier, over the past few decades, a large number of researchers have examined interior cooling channels with various ribs. Many alternative rib forms' heat transmission capabilities, however, have not yet been documented. The current study employed three-dimensional RANS analysis to assess the flow structures, heat transfer properties, and thermal performances of rib-roughened rectangular channels. The rib shapes included two cases: Case I involved reveres pentagonal shapes at the same height and reveres pentagonal shapes at uniformly varying heights, while Case II involved reveres triangular shapes at the same height and Triangular shapes at uniformly varying heights.

COMPUTATIONAL ANALYSIS

The computational analysis is predicated on the following premises.

1) The flow is two-dimensional, turbulent, fully developed, and stable.

2) The roughness material, absorber plate, and duct wall's thermal conductivity are temperature-independent.

3) The roughness material, absorber plate, and duct wall are all isotropic and homogenous. Since there is very little change in density,

4) It is believed that the working fluid, air, is incompressible within the operational range of solar air heaters.

5) The walls in the model that come into contact with the fluid are given a no-slip boundary condition.

6) Other heat losses and little radiation heat transfer.

Fig. 1 and Fig. 2 depict the computational domain utilised for CFD analysis and Fig. 3 and Fig. 4 shows the modeling of the reverse pentagonal rib.

The computational domain is defined, and then a non-uniform mesh is produced. Since the turbulent boundary layer is so thin about the height of the flow field, it is preferable to have more cells close to the plate while designing this mesh. Boundary criteria have been established following mesh generation. First, let us clarify that the duct exit is located on the right edge, and the duct entrance is located on the left. The top surface is the top edge, while the inlet and outlet lengths are the bottom edges. Every interior border of the 3D rectangular duct is designated as a turbulator wall. ANSYS 14.5 is the program used for domain meshing. Due to the use of low Reynolds number turbulence models, very fine grids are produced. Several low Reynolds number models, including the Standard k-ω model, the Renormalization-group k-ε model,

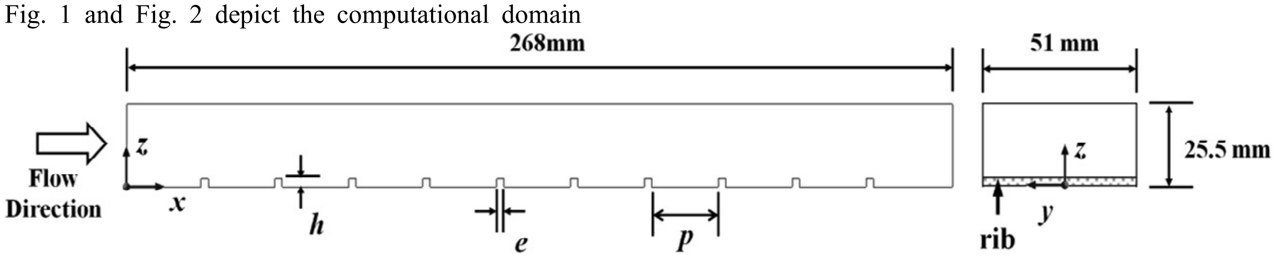
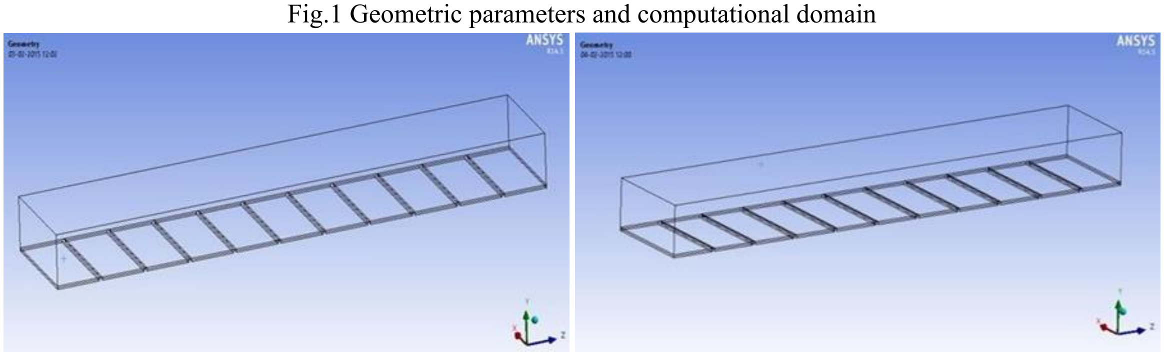


Fig.1 Geometric parameters and computational domain



Reveres pentagonal shape at same height   Reveres pentagonal shape at uniformly varying height

Fig. 2: Computational domain (Case I)

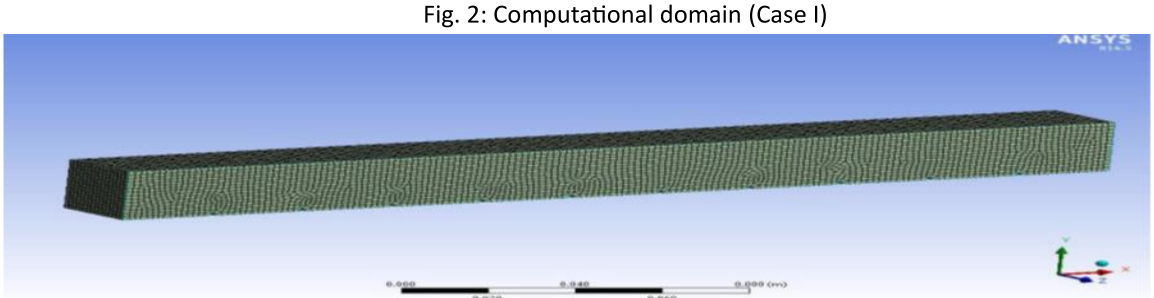


Fig.3 Modelling of Reverse pentagonal

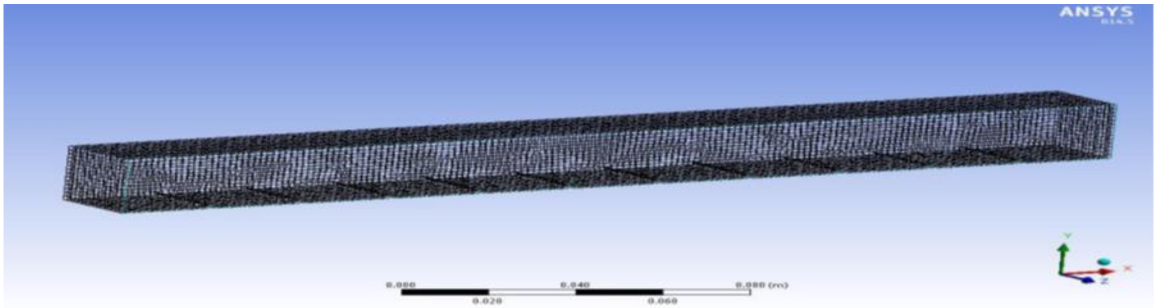
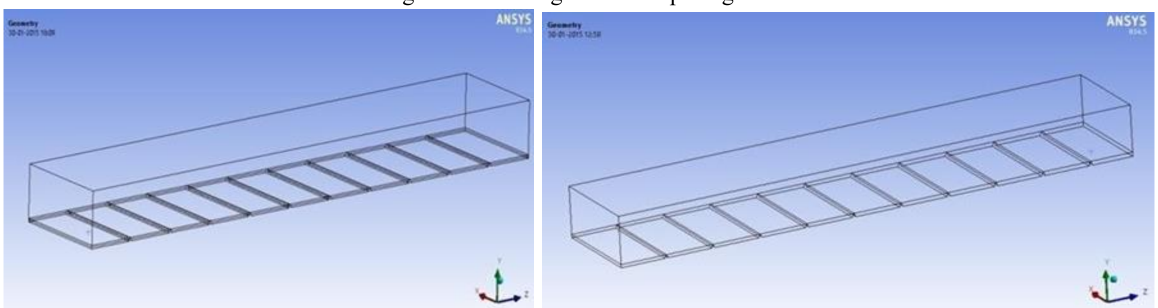


Fig 4 The meshing of reverse pentagonal



The triangular shape at the same height The Triangular shape at uniform height

Fig. 5: Computational Domain (Case II)

The realizable k-ε model, and the Shear stress transport k-ω model, are used to simulate the previous experimental investigation to choose the turbulence model. The outcomes of various models are contrasted with those of experiments. Based on its closer findings to the experimental data, the RNG k-ε model is chosen. Due to the extremely little variance in the operating range of the duct, it is believed that the working fluid, air, is incompressible. Using the Reynolds number, the mean inlet velocity of the flow was determined. Inlet boundary conditions and outlet boundary conditions for velocity have been identified. The governing equation was discretized with the use of the SIMPLE method and second-order upwind. The FLUENT software solves the following mathematical equations which govern fluid flow, heat transfer, and related phenomena for a given physical problem.

Table-1 properties of the working fluid (air) and rectangular duct

Properties	*Working fluid (air)*	*Top plate(aluminum)*
Density (kg/m^{-3})	*1.225*	*2719*
Specific heat c_p	*1006.43*	*871*
Viscosity (Nsm^{-2})	*1.7894×10^{-5}*	*--*
Thermal conductivity k ($Wm^{-1}k^{-1}$)	*0.0242*	*202.4*

Table-2 Calculation Parameter of at same Height Case (I)

Re no	*Pressure drop Pa*	*Nusselt no.=hl/k*	*Friction factor f*	*Objective function Ff*
4000	*2.0551836*	*0.016179308*	*0.1271217841*	*2.307854636*
8000	*7.0929518*	*0.020529396*	*0.11010768*	*2.330483209*
12000	*14.062802*	*0.022676229*	*0.0976912718*	*2.3560458*

Table-3 Calculation Parameter of at Uniformly Varying Height Case (I)

Re no	*Pressure drop Pa*	*Nusselt no.=hl/k*	*Friction factor f*	*Objective function Ff*
4000	*1.916368*	*0.016181294*	*0.1273140007*	*2.30901726*
8000	*6.6133466*	*0.020527079*	*0.109774701*	*2.338101343*
12000	*13.244226*	*0.02267566*	*0.09772581173*	*2.356323477*

Table No.-4 Calculation Parameter of at same Height Case (II)

Re no	*Pressure drop Pa*	*Nusselt no.=hl/k*	*Friction factor f*	*Objective function Ff*
4000	*2.0551836*	*0.01627716*	*0.1365362218*	*2.3634801*
8000	*7.0929518*	*0.020430432*	*0.1177356503*	*2.4837801*
12000	*14.062802*	*0.022414996*	*0.10376587*	*2.4038947*

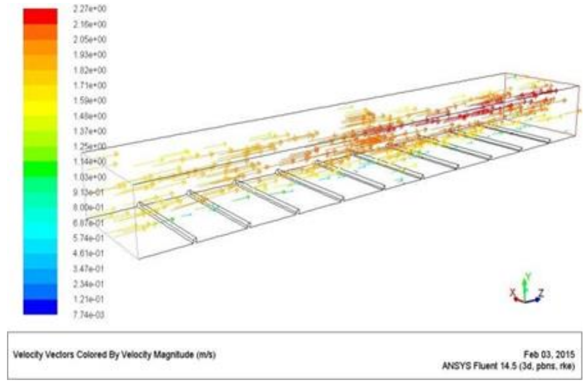
Table-5 Calculation Parameter of at Uniformly Varying Height Case (II)

Re no	*Pressure drop Pa*	*Nusselt no.=hl/k*	*Friction factor f*	*Objective function Ff*
4000	*2.2032502*	*0.016385512*	*0.146330335*	*2.4186931*
8000	*7.62783*	*0.020436946*	*0.1266140742*	*2.44299009*
12000	*15.105382*	*0.022387851*	*0.1114588136*	*2.46189742*

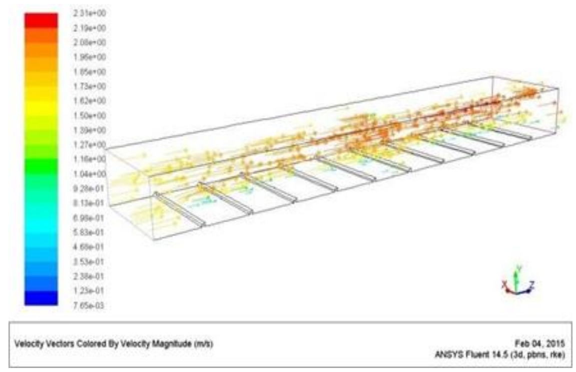
RESULT AND DISCUSSION

The current work presents CFD Analysis measurements of pressure loss and heat transfer in channels with various rib configurations. As previously noted, measurements were made in a channel with an aspect ratio of AR=1 for four rib configurations with every rib shape throughout a range of Reynolds numbers. To create a more realistic flow with an entrance effect for the test section, the inlet for the computational domain is also the inlet to the domain. It is presumed that the flow exits the test portion to ambient, with an outlet boundary condition of zero static pressure.

The plot of the Reynolds number against the average Nusselt number for various rib height values is displayed in Fig. 2. Because of the rise in rib height, the average Nusselt number rises as the Reynolds number does as well. It is evident that when the Reynolds number rises, the roughened duct's improved heat transmission compared to the smooth duct likewise rises. Additionally, at a constant amount of roughness pitch (P), it is evident that Nusselt number values rise with an increase in relative roughness height (e/d). This is because the heat transfer coefficient is higher at the reverse pentagonal and lower at the leading edge of the Triangular ribs. The stronger secondary flow is created by increased reattachment of the free shear layer caused by higher relative roughness height. At a Reynolds number of 12000, the roughened duct with a relative roughness height of 1.598 yields the maximum Nusselt number (Nu= 0.02267566). At a Reynolds number of 4000, the roughened duct with the lowest Nusselt number is produced by a pitch-to-width ratio of 14.3366. For AR=1 at a Reynolds number of 18000, the maximum enhancement of the average Nusselt number is determined to be twice that of the smooth duct.



(a) Reveres pentagonal shape at the same height



(a) Reveres pentagonal shape at uniformly varying height

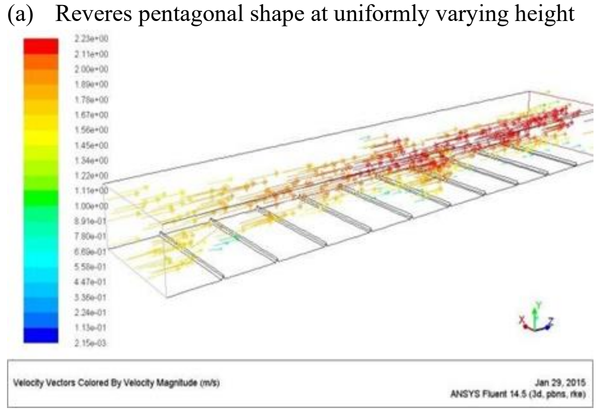


Fig. 6: Nusselt number vs Reynolds number (Case II)

(a) The triangular shape at the same height

(b) Triangular shape at uniformly varying height

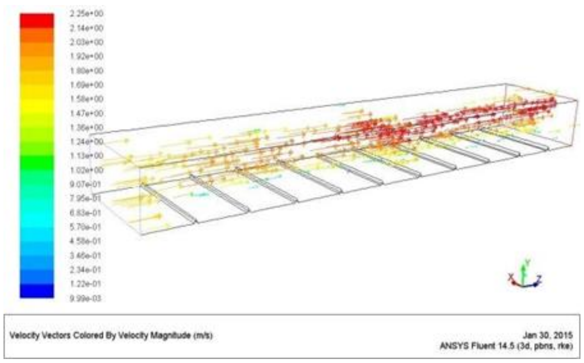


Fig. 7: Contour plot of turbulence intensity(Case I)

CONCLUSION

The fluid pressure decreases with increases in unite with the rise in heat transmission. The degree of friction factor augmentation for the one-ribbed wall channel in comparison to the smooth channel. The friction factor ratio f/f_0 increases significantly (from 2.41 to 2.46) when ribs are present at walls with uniformly variable heights, and it becomes about greater when compared to a wall channel with only one rib. The tendency is for f/f_0 values to somewhat rise as the Reynolds number increases. Coefficients of friction and heat transmission are in good agreement with findings from other researchers. Despite this, disparities become apparent when correlations are compared to dams that have been lowered using the Hall-type transformation. As the Reynolds number rises, so does the Nusselt number. At a higher Reynolds number of 18,000, the maximum value of the average Nusselt number is determined to be 0.02267566 for the pitch-to-width ratio of 14.33.

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