Three Dimensional CFD Analysis of Rectangular Roughened Channel

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***Abstract*— In this paper three-dimensional CFD simulations are carried out to investigate heat transfer and fluid flow characteristics of artificially roughened duct using Ansys- Fluent. Heat transfer and pressure drop characteristics of the rectangular duct are investigated for Reynolds numbers ranging from 8070 to 18480. Model geometry is designed, meshed, analysed, and post-processed using Ansys-fluent software. Fluid flow and heat transfer of different roughness configurations are simulated and results compared using turbulent flow model (k-ω, with SST*), with steady-state solvers to calculate pressure drop, flow, and temperature fields. Rectangular duct has an aspect ratio of 5, while the domain length for numerical analysis is kept 450 mm long. The bottom wall of the duct is roughened by two different geometries of discrete ribs. Reasonable difference is found between the heat transfer simulation data for different roughness configurations, and the Ansys-Fluent software has been sufficient for simulating the flow fields in rectangular duct.**

***Keywords—Heat Transfer Enhancement*; Turbulent Flow; Artificial Roughness; Ribs; Forced Convection**

INTRODUCTION

In order to improve performances, in terms of higher thermodynamic efficiency and power output for heat exchanging devices like solar air heaters, heat exchangers and machines like gas turbines, the study of heat transfer mechanism and fluid flow play a vital role. Thermal assessment of heat exchanging devices, computational fluid dynamics (CFD) approach has advantages over the cost effecting practical experiments. To understand the complete phenomenon of heat transfer, heat flow as well as fluid flow must be considered in numerical analysis. Hence, accurate numerical method turbulence model and equation solving approach are required to predict heat transfer behaviour. [1]

In the direction of disturbing the sub-laminar layer adjacent to solid wall many types of roughness have been investigated by many authors in the last few decades.

Artificial roughness applied to break the laminar sub-layer to enhance heat transfer coefficient is used in various applications like heat exchangers, blade cooling channels in gas turbine, nuclear reactors and solar air heaters [2-3]. The use repeated ribs as an artificial roughness has been found to be as an efficient method of enhancing the heat transfer of fluid flowing in the channel by several investigators [4-5]. The ribs applying to the wall of flow passage of fluid break the laminar sub-layer and create local wall turbulence due to flow separation and reattachment of flow between consecutive ribs, which decreases the thermal resistance and highly augment the heat transfer. The enhancement of heat transfer by flow separation and reattachment, caused by ribs, is significantly higher than compared to the fin-effect, linked to the increased heat transfer area [6]. However, the use of artificial roughness results in higher friction losses leading to excessive power requirement for the air flow through the duct. Therefore, it is desirable that the turbulence must be created only in the region very close to the heat transfer surface, i.e. in the viscous sub-layer where the heat transfer takes place and the core flow should not be disturbed so as to avoid excessive friction losses. This can be achieved by keeping small the height of roughness elements in comparison with the duct dimensions. [7] It has been found through experimental investigations that the enhancement in Nusselt number for air flow in a roughened channel with angled ribs is on the average higher than that roughened with transverse (90 deg) ribs of the same geometry. Secondary flows generated by the angled ribs are considered to be reasoned for these higher heat transfer coefficients [8]. [9] Studied both the thermal and fluid-dynamic behaviours of turbulent flows with two-dimensional ribs and three-dimensional blocks in the context of surface roughness effects. They solved the Reynolds-averaged Navier-Stokes equations, coupled with the k-ω turbulence model with near-wall treatment, by a finite-volume method. They analysed different three dimensional configurations with ribs, both in line and staggered. For the in-line cases, they found that Nusselt number was about 20% lower than the results obtained from the 2-D simulations.

Literature [10] experimentally investigated the flow and heat transfer characteristics in a rectangular channel, with a large width-to-height ratio and 45° angled ribs arranged parallel on one or two opposite walls. In a rectangular channel (aspect ratio AR = 5) with angled rib turbulators, inclined at an angle of 45°, have been investigated with the Reynolds number Re = 29,000, The secondary flow brings relatively cold core fluid towards the rib leading-end regions and warm fluid towards the opposite rib trailing-end regions, causing marked span wise variations in the local heat transfer coefficient.

Two dimensional Particle Image Velocimetry (PIV) system and the heat transfer results were correlated with the flow structure [11]. It was found that inclined rib with a gap (inclined discrete rib) had a better heat transfer performance compared to the continuous inclined rib arrangement. Investigations have been carried out for the relative gap width of 0.5 to 2.0 and the Reynolds number range of 3000-18000, the inclined discrete rib with relative gap width (g/e) of 1.0 gives the higher heat transfer performance compared to the other relative gap width. [12-13] compared with the numerical CFD and experimental results for various types of rib roughness. The numerical simulation results agreed well with the experimental data by using the SST k-ω turbulence model and found the differences between the numerically calculated Nusselt number and the experimental data of less than ±10%.

Based on the above literature, it seems that inclined discrete ribs roughness geometry shows better heat transfer enhancement as compared to the continuous rib roughness arrangement. Researchers are continuously putting their effort to improve the heat transfer performance from heat transferring surface. In the present study 45° inclined discrete rib roughness arrangement with in line gap and zig zag gap patterns have been considered and heat transfer and friction factor analysis has been carried out.

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CFD ANALYSIS

An approach for concern with turbulent flow over a rib configured channel the selecting an appropriate grid configuration and corresponding turbulent models are investigated using CFD package Ansys- Fluent.

The CFD analysis follows the major steps: geometry creation in three dimensional domain, grid generation, defining boundary condition, computing and post processing.

COMPUTATIONAL DOMAIN AND GRID GENERATION

In this numerical investigation, three- dimensional turbulent flow through the rib configured channel is simulated. Domain section is of a rectangular channel of length L= 550mm long, height 40 mm, and width W= 200 mm long makes aspect ratio 5. One wall (Bottom) of the domain was roughened, while three others were smooth. Square shape, rib configuration was selected for creating the roughness of the wall. The ribs were arranged in discrete fashion on the bottom wall in a well-defined pattern. The transverse position of rib makes an angle of α = 45 degrees, with stream flow direction. The ribs, with square cross-sections ($e = t = 2$ mm) are spaced with rib pitch-to-height ratios p/e = 10 to yield rib pitch p = 16 mm.

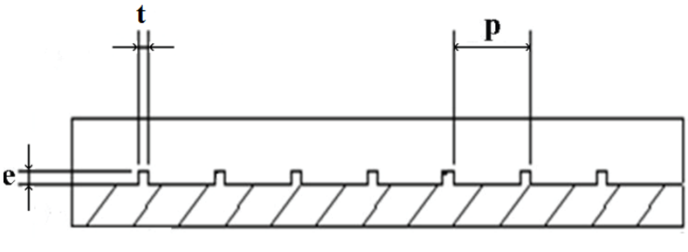

Figure 1 (a) Rectangular channel with rib- roughened wall.

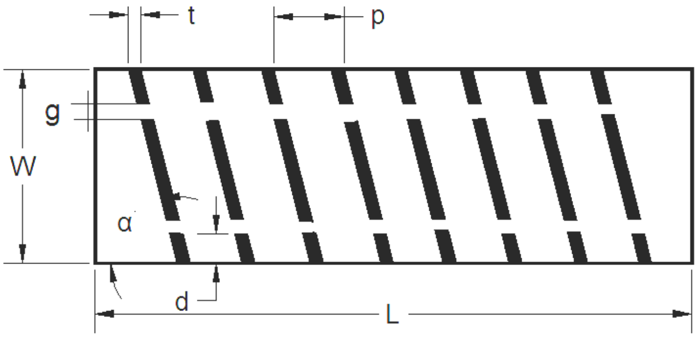

Figure 1 (b) In-line gapping rib configuration.

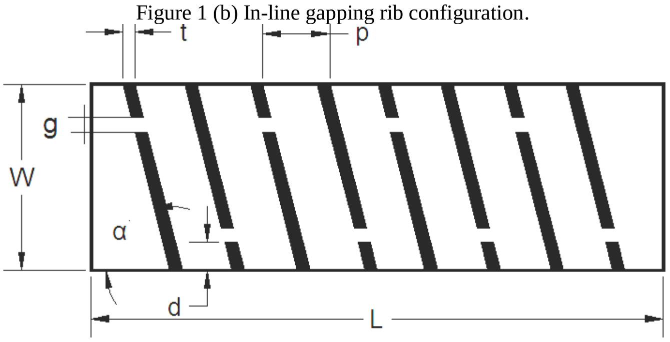

Figure 1 (c) Zig-Zag Gapping rib configuration.

There are two new types of rib configuration are selected for numerical analysis shown in figure 1 (b) and in figure 2 (c), In-line gapping and zig-zag gapping respectively. Heat transfer and pressure drop analysis has been done by author [8] for the inline gapping condition in discrete rib configuration. In both the patterns the relative gap width (g/e) kept 1.0.

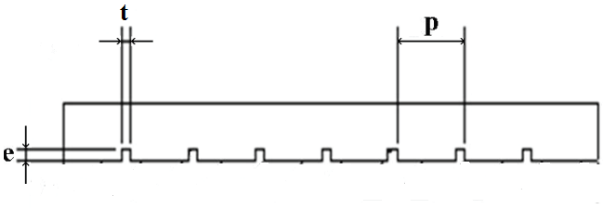

Figure 2 computational domain for CFD analysis.

Figure 2 shows the computational fluid domain, consider in the CFD analysis in which uniform heat flux (1000 watt) is supplied through the bottom wall included the rib face.

TURBULENCE MODELING AND VALIDATION

In this numerical analysis the fully turbulent conditions are assumed at different uniform velocities. The Reynolds numbers are ranges from 8070 to 18480 at ambient temperature (Ta= 300K) and pressure. At the solid wall no slip condition is employed for the flow. The Inlet velocity and outlet flow condition are given for the permeable walls. Air is used as the working fluid, while the bottom wall material is aluminium. Thermo– physical properties of working material are shown in table 1.

Table 1. Thermo-physical properties

Properties	*Air*	*Aluminium bottom Plate*
Density kg/m3	1.225	2719
Specific heat j/kg-k	1006.43	871
Viscosity kg/m-s	1.7894 e-05	-
Thermal conductivity kg/m-k	0.0242	202.4

The two turbulent model k-ε, RNG and k-ω SST were chosen for model validation. Simulation and Nusselt numbers calculation were conducted for smooth channel for both the models. The figure shows the variations of Nusselt numbers for different models with respect to change of Reynolds number. Results are validated by comparing the obtained numerically predicted data with the Nusselt numbers calculated using the following correlation given by Dittus Boelter.

$$Nu_s = 0.024\, Re^{0.8}\, Pr^{0.4} \qquad (1)$$

Figure1 shows the trend of variation of the Nusselt number with respect to Reynolds number is same in the k-ω SST model and number derived from Dittus-Boelter Equation.

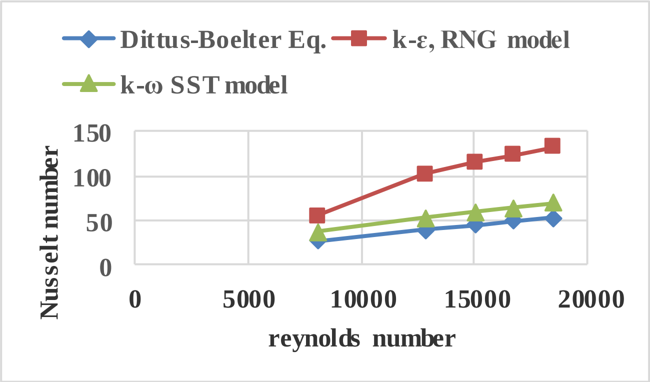

Figure 3 Nusselt Numbers variation of smooth surface.

GOVERNING EQUATION AND HEAT TRANSFER ANALYSIS

Finite volume steady state approach was used for solving continuity, momentum and energy equation in the steady-state regime. A second order upwind scheme was selected for solving the energy and momentum equations. The Semi-Implicit pressure linked equation (SIMPLE) was chosen as coupling scheme to couple pressure and velocity. The convergence criterion for velocity component is about 10e-3 while 10e-6 is considered for energy equation.

RESULT AND DISCUSSION

Figure 4 presents the normalized stream wise velocity distributions inside the channel. The velocity profile at the bottom is not similar because of the arrangement of ribs at the bottom wall, whereas the velocity profiles at the top is uniform because the top surface is smooth. At the bottom surface for *p/e*=10, the backward flows at *x/e*=1, 5, 8 are followed by the reattachment point *x/e*=10 at this point the velocity gradient is zero.

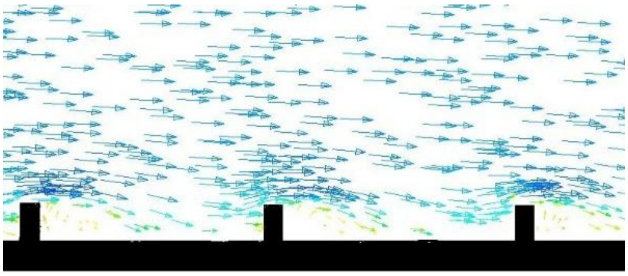

Figure 4 Velocity vector plot for ribbed channel.

Figure 5 shows the path line of main stream flow of the proposed roughness arrangement. It gives a clear visualization of eddy formation, reattachment point on the bottom wall. This analysis shows that the selection of roughness parameters is justified for the proposed study.

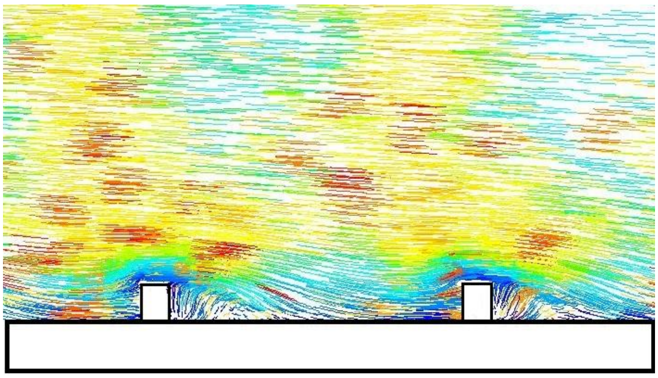

Figure 5 Path line of stream flow of air for ribbed channel.

HEAT TRANSFER AND PRESSURE DROP

INLINE GAPPING

Studies were conducted over the 45° inclined ribs with the inline gap arrangement and zig zag gap position in consecutive ribs. Figure 6 and figure 7 present the temperature- contour in x-y plane section view for Re= 18480 for in-line gap position of the leading edge and trailing edge respectively. Thermally effected zone near the ribs and inter-rib spacing have been examined because it is the large thermally effected zone. At the leading edge side rib surface cold fluid carries more heat as compared to trailing edge side surface, this difference occurs because of generation of secondary flow that travel along the ribs from leading edge to trailing edge. So, the hot spot is seen in the inter-rib region at trailing edge side.

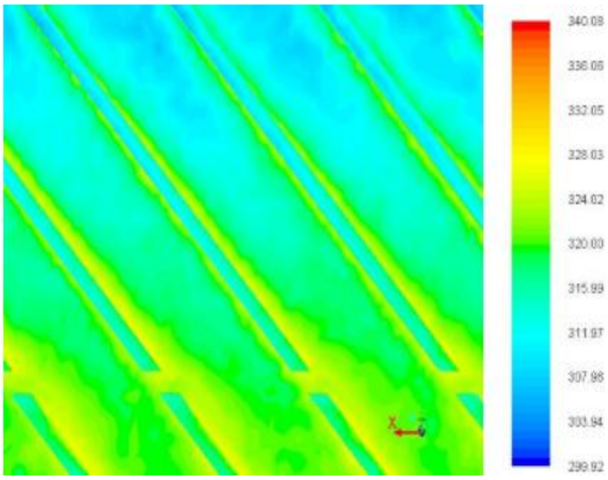

Figure 6 Temperature contour on the ribbed surface, case of in-line gapping at the leading edge side.

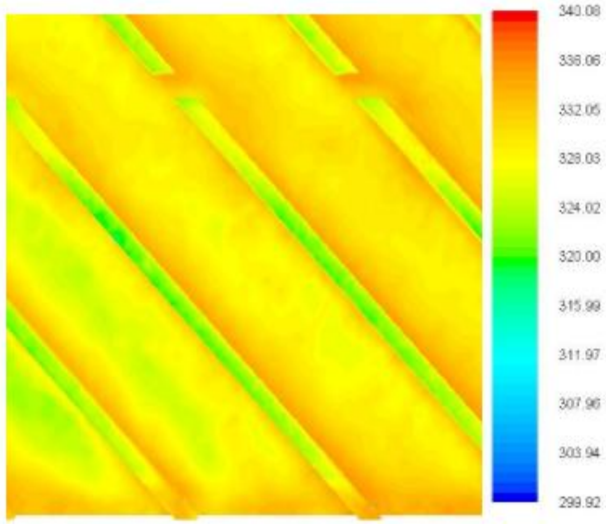

Figure 7 Temperature contour on the ribbed surface, case of In-line gapping at the trailing edge side.

ZIG-ZAG GAPING

From the figure 8 and figure 9 it is observed that the same thermal effect arises in the case of a zig-zag order gap position as it is observed in case of in-line gap position arrangement in rib. The leading edge heat removal is higher as compared to trailing edge. From this temperature distribution it is clear that in case of zig-zag gap pattern the heat removal by the fluid is higher as compared to inline gap pattern because of generation of secondary flow that travel along the ribs from leading edge to trailing edge.

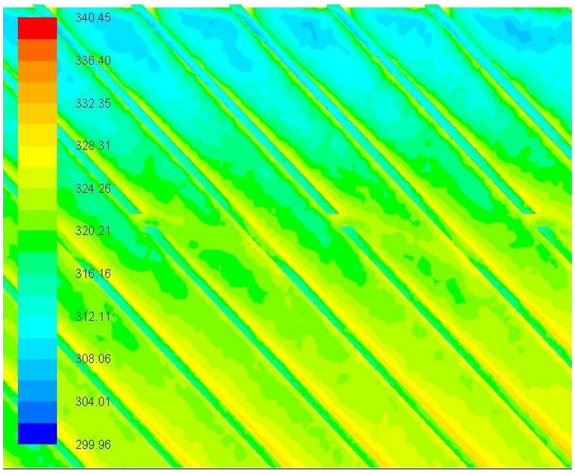

Figure 8 Temperature contour on the ribbed surface, case of Zig-zag gapping at the leading edge side.

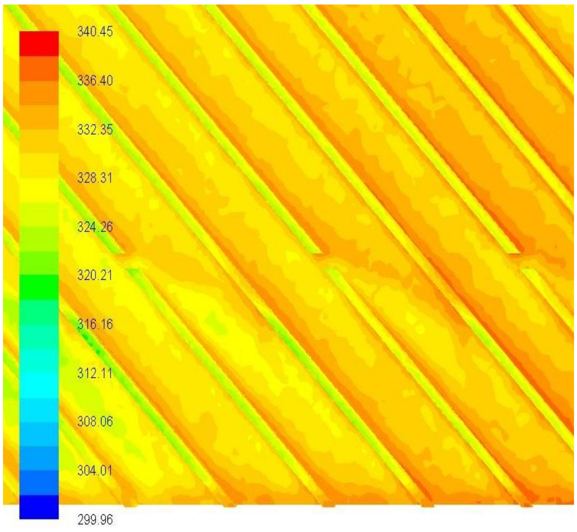

Figure 9 Temperature contour on the ribbed surface, case of Zig- zag gapping at the trailing edge side.

Figure 10 shows the distribution of heat transfer co-efficient as a function of Reynolds number for the In-line gapping arrangement and zig-zag gapping position in the ribs. It is observed that the value of heat transfer coefficient of zig-zag gap position is more as compared to the in-line gap position. Because the heat removal in case of zig-zag gap pattern is higher as compared to in-line gap position arrangement, as it has been confirmed from the temperature field arrangement as discussed above

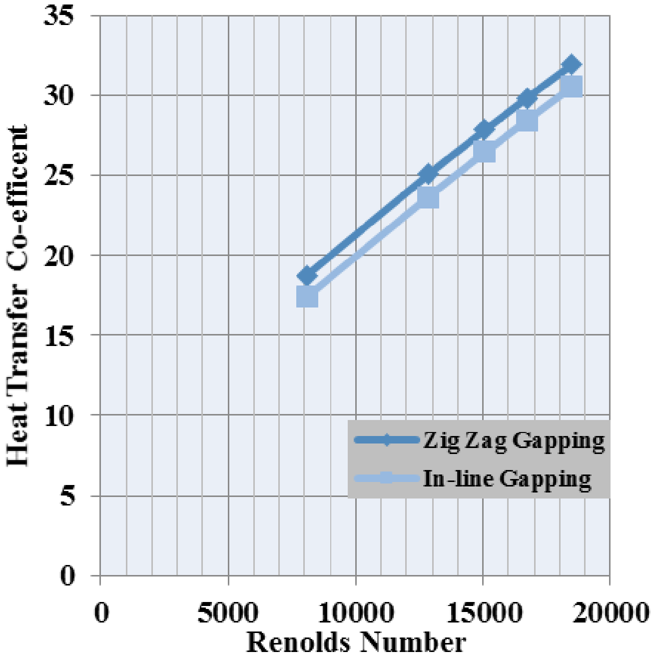

Figure 10 Effect of rib configuration on average number for different values of Reynolds number.

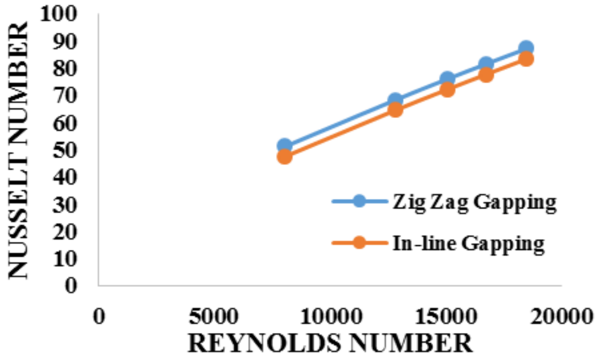

Figure 11 Effect of rib configuration on average Nusselt number for different values of Reynolds number.

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Figure 12 shows the pressure loss through the channel in considering cases of roughness pattern. Pressure loss is much higher in case of in-line gap pattern to zig-zag gap pattern at low Reynolds number.

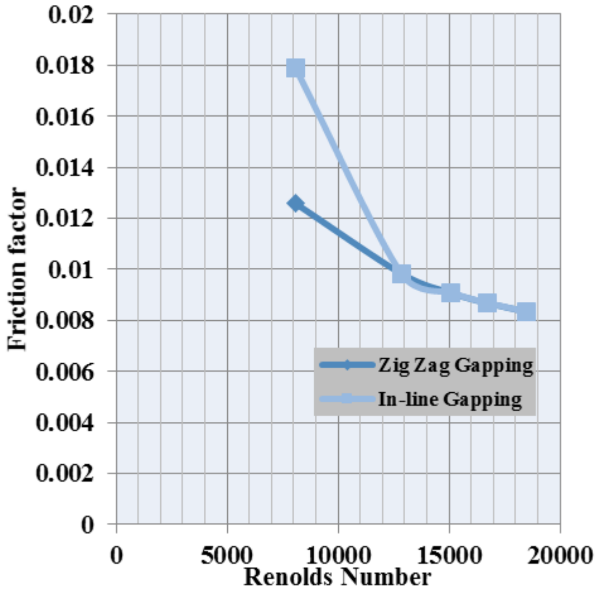

Figure 12 Effect of rib configuration on average friction factor for different values of Reynolds number.

Heat transfer performance

Heat transfer performance is characterized by the ratio of the Nusselt number of roughened channel with that of smooth channel. Figure 13 gives a comparative result on performance of studying roughness configuration in the form of a Nusselt number ratio varies with Reynolds number. It decreases with increasing Reynolds number.

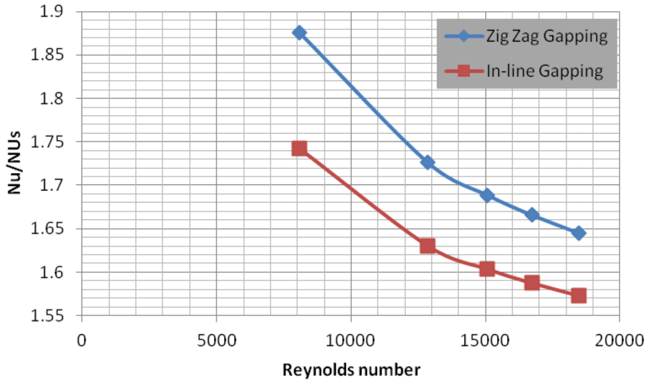

Figure 13 Nusselt number ratio for the rib configurations.

CONCLUSION

Heat transfer and friction loss in a rectangular channel with the two different types of discrete ribs were investigated and compared with the results. The main conclusions are described as follows:

- For the selected pitch to height ratio (p/e=10) the reattached point is occurring between the two consecutive ribs.
- For the zig- zag gape position configured gives a better heat transfer performance over the in-line gap arrangement of ribs inclined at 45 degree to the stream flow.

- Heat transfer performance decreases with an increment of Reynolds number for both the roughness pattern.
- Friction losses in the case of inline gap arrangement is higher as compared to zig-zag gap pattern at low (8000 Re), and almost remain equal as increases of the Reynolds number from 12500 Re.

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