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Numerical Study of Heat Transfer and Flow Characteristics in a Rectangular Channel with Rib Type Turbulent Promoters

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ABSTRACT
Numerical simulation is carried out to study turbulent forced convection heat transfer and friction loss for flow through a rectangular channel having a constant heat flux with different rib type turbulent promoters. The rib shapes used in this study are rectangular, triangular, trapezoidal, wedge rib upstream and wedge rib downstream. Two rib configurations: (i) ribs mounted on bottom wall and (ii) ribs mounted on both wall (staggered rib array) are simulated. Standard k- ε turbulence model and enhanced wall treatment are used to perform the simulation. The inflow Reynolds number is varied from 5000 to 10000. Simulation is performed for a rectangular channel with aspect ratio 13, pitch to height ratio of 4 for configuration (i) and 8 for configuration (ii), rib width to rib height ratio of 2, blockage ratio of 1/3 for configuration (i) and 1/6 for configuration (ii). The results indicate that the heat transfer performance and the friction loss are strongly affected by different rib shapes. The highest heat transfer performance is achieved for wedge rib upstream for configuration (i). For configuration (ii) highest value of heat transfer performance is obtained for triangular rib. The lowest value of heat transfer performance is obtained for wedge rib downstream for both configurations. For ribs mounted in bottom surface wedge rib downstream has the highest frictional loss and for staggered rib arrays highest frictional loss is found for wedge rib downstream. Triangular ribs mounted on both wall with staggered rib array configuration shows the highest thermal performance among the rib shapes.

Keywords: Numerical simulation, Heat transfer, Friction loss, Reynolds Number, Rib type turbulent promoter.

1. Introduction

The usage of ribs in the cooling channels or heat exchangers is one of the commonly used passive heat transfer enhancement technique. Heat transfer augmentation with turbulent promoters has become significantly important in many aspects of engineering. Different shaped ribs are used so far and their effects are analyzed. Laminar flow is comparatively easy to analyze with governing equations but turbulent flow is beyond analytical analysis depends massively on physical intuition and dimensional arguments. It is also necessary to consider their relations that describe the manner in which they influence the velocity distributions which in turn affects temperature fields. Numerical simulations are used to analyze turbulent flow and their behavior while flowing through a channel with ribs. Enhancing heat transfer on a surface is done by making the surface rough and by the use of repeated ribs to act as turbulent promoters. These turbulent promoters break the laminar sub layer and or buffer layer and create local wall turbulence due to flow separation and reattachment between the ribs, which greatly enhance the heat transfer and increase the pressure drop. Liu Pingan, Gao Ye et al. [1] get highest heat transfer for square ducts with triangular ribs applying high Reynolds no.. In the work of Kamali R et al. [6] the highest heat transfer occurs in the case of trapezoidal ribs with decreasing height in the flow direction. Francesca, Danielle et al. [2] used 45^{o} ribs in a rectangular channel for their simulation. In our present work, the effect of turbulence promoters on fluid flows and heat transfer in a square duct for Reynolds numbers varying from 5000 to 10000 will be investigated. While traditional approaches use high Reynolds no. but using flow of low Reynolds no. is becoming very important in non-circular ducts flow and practical usages like micro cooler, automotive cooler and cold plates. In conversion and reclamation devices, also there is significant usage of low Reynolds no. flow. A two-dimensional domain of a square channel is considered and a constant heat flux boundary condition is applied to the ribs section of the channel. Air is considered as the medium flowing in the channel with varying velocity. Inlet air temperature is 298K. A constant heat flux of 100 W/m^2 is provided in the whole channel length section. It is assumed that the flow is a steady, turbulent, incompressible and fully developed flow. Numerical simulations are performed using the finite volume method. Standard k-ε turbulence model and enhanced wall treatment are used to produce the numerical simulations. Archarya et al. [3] applied *k*-ε models to straight and inclined ribs for two-dimensional rectangular ribs and found similar performance for these two cases. Iacovides and Raisee [7] examined the capabilities of low-Reynolds number versions of Launder & Sharma *k*-ε model [8] in predicting convective heat transfer for pipes and ducts. They obtained a more realistic heat transfer variation in the separation region and reasonable Nusselt number levels. *k*-ε model was also selected due to its good convergence rate and relatively low memory requirements. Another advantage of the model is it does perform well for external flow problems around complex geometries. It is the most common model used in Computational Fluid Dynamics (CFD) to simulate mean flow characteristics. Also, it provides good initial guess as well.

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The main objective is to study the effect on the heat transfer of a square channel with different ribs with pitch (P) to rib height (k) ratio as 4 and width (b) to rib height (k) ratio as 2 for various Reynolds number with dimensions as shown below. The two-dimensional detailed view of the channel is shown below:

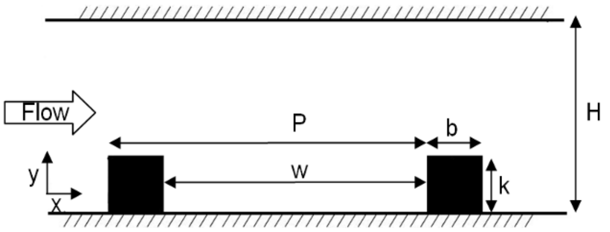

Figure 1: Detailed view of the rectangular channel with rectangular ribs from the work of R. Kamli et aI,. [6]

Various parameters of the channel are as follows: P/k is 4, b/e is 2, H/e is 3, the length of channel (L) is 39cm. In all these simulations, k is considered as 1cm. Hence, H is 3cm, b is 2cm, w is 2 cm and p is 4cm. The inlet velocity of the air is uniform and is calculated as per the required inflow Reynolds number. Similar simulation is carried out for staggered rib arrays. In this arrangement, ribs are also present in upper surface and each rib at upper surface is positioned in the middle of two ribs at lower surface. Reynolds number range is kept same for the experiment and a constant heat flux of 100 W/m^2 is also provided in the upper surface. The purpose of the study is to reconnoiter numerically the effect of heat transfer through a rectangular channel with different type of ribs. The shapes of ribs that are used in the numerical simulation are: rectangular, triangular, trapezoidal, wedge rib upstream and wedge rib downstream. Air is considered as the medium flowing in the channel and inlet air temperature is 298K. Hence the specific objectives of present study are as follows: 1) To measure average Nusselt number for the channel with different shaped ribs mounted on the bottom side of the channel. 2) With analyzed data graph will be plotted for average Nusselt number vs Reynolds number. 3) Simulations will be carried out to study the effect of ribs in both sides of the rectangular channel and a similar graph will be plotted.4) For each cases, variation of Nusselt number, Friction factor and Enhancement factor with Reynolds number will be shown in graph. 5) The results obtained from this investigation are to be compared with the published data available in the literature. 6) Conclusions will be drawn and recommendation will be made based on the discussion of the results.

2. Computational framework

In this work, Commercial CFD Software for Numerical Solution. Finite Element Method (FEM) was used for solving Incompressible Navier-Stokes. Momentum equations was solved (with or without pressure/either implicit or explicit) to obtain a predicted velocity. Then to solve pressure poison equation to obtain a pressure correction that can be used to update velocity so that continuity is satisfied. The grid independency test was performed. It was done for computing the solution on successively finer grids. The difference between two refinements was taken as a measure of the accuracy of the coarser of the two.

2.1 Governing Equation

The cornerstone of computational fluid dynamics is the fundamental governing equation of fluid dynamics, the continuity, momentum and energy equations. They are the mathematical statements of three fundamental physical principles upon which all of the fluid dynamics is based. 1) Mass is conserved 2) F=ma (Newton's second law) 3) Energy is conserved. Our study deals with all three of the above mentioned equations as it is concerned with both heat transfer and fluid flow.

Continuity conservation:

$$\frac{\partial \rho}{\partial t}+\nabla.\left(\rho \mathbf{v}\right)=0 \tag{1}$$

Momentum conservation:

$$\frac{\partial\left(\rho \mathbf{v}\right)}{\partial t}+\nabla.\left(\rho \mathbf{v}\,\mathbf{v}\right)=\rho g-\nabla P+\nabla.\left(\boldsymbol{\tau}\right) \tag{2}$$

Energy conservation:

$$\frac{\partial\left(\rho E\right)}{\partial t}+\nabla.\left(\mathbf{v}\left(\rho E+p\right)\right)=\nabla.\left(k_{eff}\nabla T+\left(\boldsymbol{\tau}_{eff}.\mathbf{v}\right)\right) \tag{3}$$

Where,

$$\boldsymbol{\tau}=\mu\left(\left(\nabla\mathbf{v}+\nabla\mathbf{v}^{T}\right)-\frac{2}{3}\nabla.\mathbf{v}I\right) \quad \text{and} \quad E=h-\frac{P}{\rho}+\frac{v^{2}}{2}$$

Here, v is the velocity vector of the flow, ρ is the density of the flow.

2.2 Performance Parameter

The main dimensionless parameters governing this problem are Reynolds number and Nusselt number.

$$\mathrm{Re}=\rho\mathbf{u}D/\mu \tag{4}$$

Where, ρ is the density, μ is the dynamic viscosity and the velocity of the air.

$$Nu_{x}=\frac{h_{x}D}{k} \tag{5}$$

And the average Nusselt number is obtained by,

$$Nu = \frac{1}{A}\int Nu_x \partial A \tag{6}$$

Friction factor is calculated by,

$$f = \frac{2}{\left(L/D_h\right)}\frac{\Delta P}{\rho U^2} \tag{7}$$

Where ΔP is the pressure drop across the test section u is the mean air velocity in the channel. All of thermo-physical properties of the air are determined at the overall bulk air temperature. The thermal enhancement factor (TEF) is defined as the ratio of the heat transfer coefficient of an augmented surface (h) to that of a smooth surface (h_0) at an equal pumping power and given by,

$$TEF = \frac{h}{h_0}|_{pp} = \frac{Nu}{Nu_0}|_{pp} = \left(Nu/Nu_o\right)/\left(f\ /\ f_o\right)^{\frac{1}{3}} \tag{8}$$

Where, Nu_0 and f_0 stand for Nusselt number and friction factor for the smooth duct respectively.

2.3 Physical Model

Fig.1 and Fig.2 shows the geometry which has been considered in this simulation to study effect of ribs on heat transfer and friction factor. The total length of channel is 39cm and 3cm in height. The space between the ribs is 4cm. The height of the rib is considered to be 1cm for ribs mounted in bottom surface and 0.5 cm for staggered rib arrays. A two-dimensional detailed view of the channel for both rib configurations is shown if Fig.1 and Fig.2.

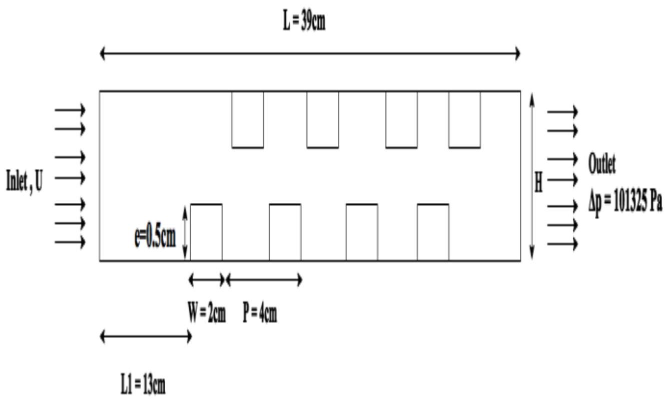

Figure2: Schematic Diagram of Rectangular channel with staggered rib arrays

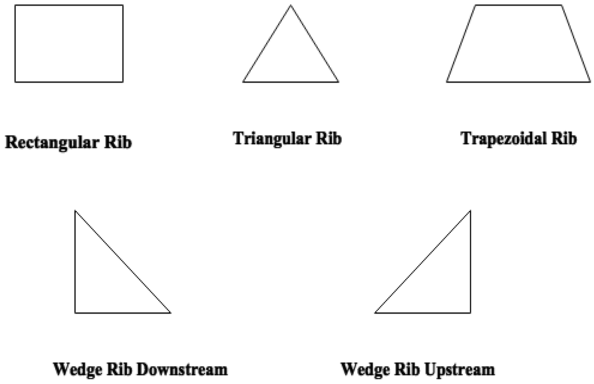

Figure 3: Different rib shapes used which are used in study

2.4 Boundary Condition

The Flow enters the domain from the left side with a velocity of U, which varies with the Reynolds number used in the study. An outlet pressure condition is imposed for the outflow at the right side of the domain. The inlet temperature of the air is 298K. A constant heat flux of $100W/m^2$ is supplied to the bottom for ribs mounted on the bottom surface and for on both side of the channel for staggered rib array configuration.

Table 1: Inlet velocity as per the inflow Reynolds number

Reynolds Number	Inlet Velocity
5000	2.595
5500	2.854
6000	3.114
6500	3.373
7000	3.633
7500	3.892
8000	4.152
8500	4.411
9000	4.671
9500	4.930
10000	5.190

2.5 Mesh Generation

In our present study the whole computational domain is discretized into free triangular. The free triangular structure is most popularly used for mesh generation in numerical modeling. In Fig.4, meshing for the whole domain for rectangular rib is shown. Dense meshing is done near the wall where ribs are mounted. For staggered rib array configuration, no of elements generated is more than ribs on lower side and also dense meshing is done on both sides (in Fig.5)

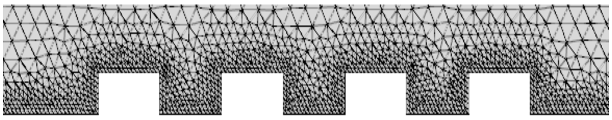

Figure 4: Triangular mesh structure of the domain for ribs mounted on lower side

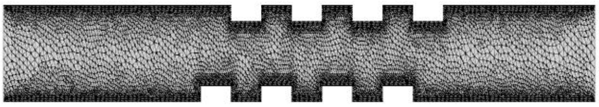

Figure 5: Triangular mesh structure of the domain for ribs mounted on lower side for staggered rib arrays

Table 2: Meshing Details for both rib configuration

	Ribs mounted on lower	Staggered rib array
Number of elements	4875	18508
Number of edge elements	465	1268
Number of vertex elements	20	36
Number of nodes	2671	9889

3. Result

The results are presented step by step, first the results of heat transfer and friction characteristics in a smooth channel are compared in terms of Nusselt number and friction factor. The data are post processed to find the variation of Nusselt number and friction factor for a range of Reynolds number.

3.1 Effect of Reynolds Number

Fig.6 shows variation of average Nusselt number with Reynolds numbers for rectangular channel and staggered rib arrays configuration. It is found that Nusselt number increases with the increase of Reynolds number which is similar to conclusion found by V.D.

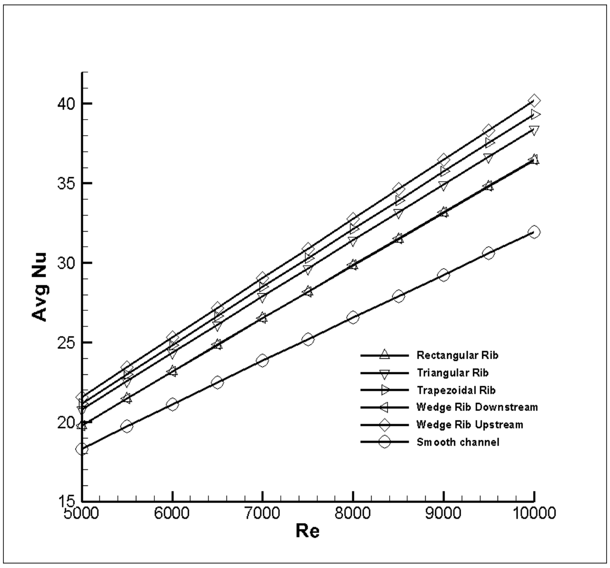

Figure 6: Average Nusselt number plot for ribs mounted on the bottom of square channel.

Boga and S.Jayave [4]. It happens because ribs interrupt the development of laminar boundary layer of the fluid flow and increase the turbulence and heat flow.

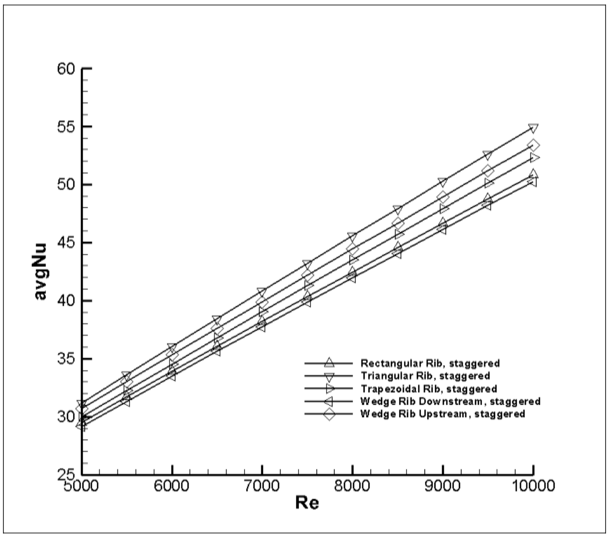

Figure 7: Average Nusselt number plot for channel with staggered rib arrays

The Nusselt number obtained under turbulent flow conditions for all five types of rib shapes are presented in the Fig.6 and Fig.7. The highest value of Nusselt number is found for wedge rib pointing upstream for ribs mounted in bottom wall and for ribs mounted on both wall it is found for triangular ribs. For both configurations wedge rib downstream has the lowest value of Nusselt number.

The wedge rib upstream and triangular ribs yield higher value of Nusselt number than rectangular ribs for all value of Reynolds number similar to the conclusion by

Ahn [5], who found that the triangular shape is more efficient for heat transfer than rectangular shaped

3.3 Average heat transfer and friction characteristics

The present result of heat transfer and flow friction characteristics in a rectangular channel equipped with different shapes of ribs are presented in the form of Nusselt number ratio and friction factor ratio. Both Nusselt number and friction factor are normalized by following factors:

$Nu_0 = 0.023Re^{0.8}Pr^{0.4}$ and $f_0 = 0.0791Re^{0.25}$

For ribs mounted on bottom wall it is found that Nusselt number ratio increases with the increase of Reynolds number for all type of rib shapes. It shows highest value for wedge rib upstream and lowest for rectangular rib. Though rectangular rib has higher surface area for heat transfer than wedge rib upstream, but having lower recirculation zone results in lower heat transfer.

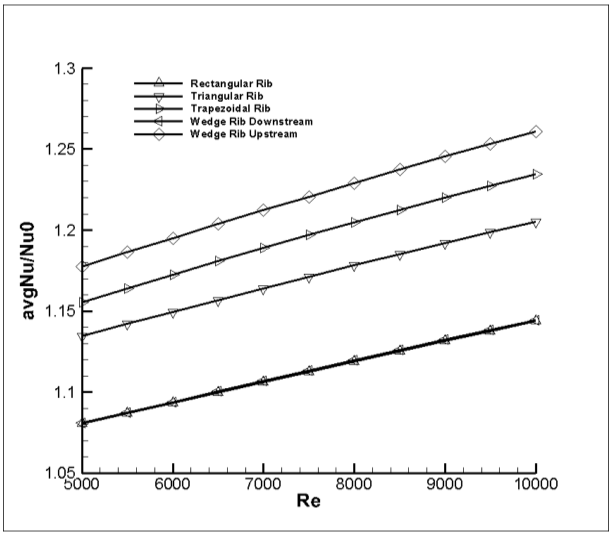

Figure 8: Variation of Nusselt number ratio with Reynolds number for ribs mounted on the bottom of rectangular channel.

Triangular rib has small recirculation zone compared to the other rib shapes except wedge rib downstream, it gives more heat transfer. Because when recirculation zone becomes larger, it reduces the flow velocity in that area which nullifies the effect of fluid mixing. Heat transfer enhancement decreases due to this phenomenon in those areas.

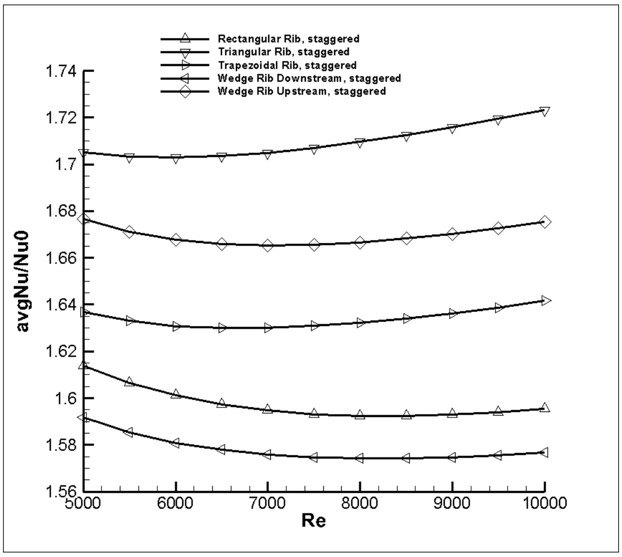

Figure 9: Variation of Nusselt number ratio with Reynolds number for channel with staggered rib arrays

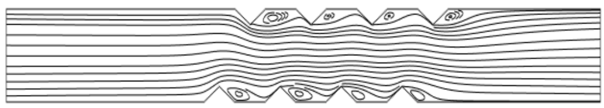

Figure 10: Small recirculation zone for triangular ribs in staggered rib arrays

The effect of ribs on pressure drop across the channel is presented in the Fig.11 and Fig.12 for both rib configurations in the form of friction factor ratio with Reynolds no. For both rib configurations wedge rib downstream has the highest frictional loss. For ribs mounted on bottom wall rectangular ribs gives the lowest frictional loss. And for ribs mounted on both wall at low Reynolds number wedge rib upstream gives the lowest frictional loss and at higher Reynolds number trapezoidal rib gives the lowest frictional loss. Friction loss across the wall section is found to increase with Reynolds number because the velocity of flow increasing with Reynolds number as well. It has a direct impact on the pressure drop which reaches higher value at higher Reynolds number. The loss is mainly due to dissipation of dynamic pressure caused by head loss due to viscosity near the wall region and blockage because of the ribs. Also, friction factor increases with Reynolds number because a high amount of fluid remains at low velocity in the recirculation region past the rib shapes. This implies high flow velocity at the center of channel, resulting increased pressure drop. For this reason, at higher Reynolds number wedge rib upstream shows more frictional loss than trapezoidal rib.

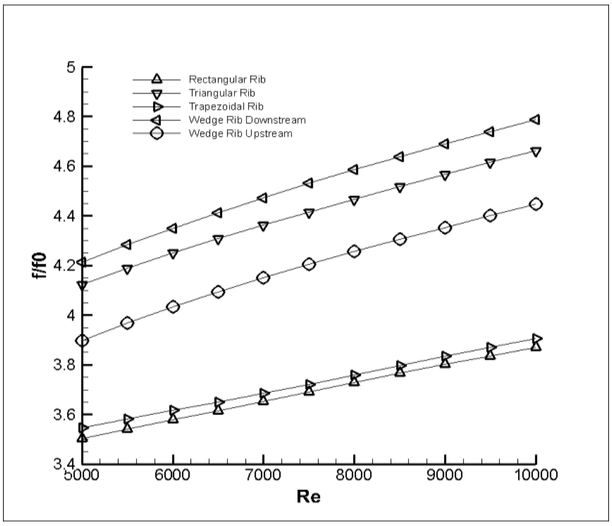

Figure 11: Variation of friction factor ratio with Reynolds number for ribs mounted on the bottom of rectangular channel

4. Conclusion

For both cases rectangular rib and wedge rib downstream gave the worst heat transfer performance and wedge rib downstream shows the highest friction loss among the investigated shapes. Rectangular ribs also show significant friction loss. Thus, rib shapes having slopes on the front proved to be a critical factor in determining heat transfer performance as it affects the recirculating zone. But friction loss doesn't depend only on the recirculation zone but mainly on the shape of the ribs. It is seen that both heat transfer performance and friction loss are dependent of Reynolds no.

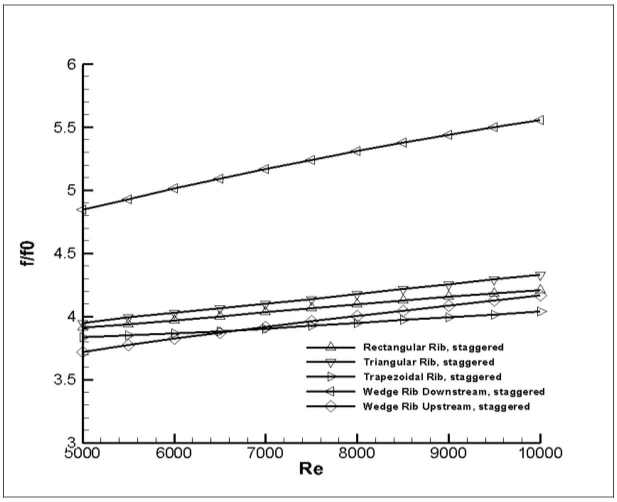

Figure 12: Variation of friction factor ratio with Reynolds number for channel with staggered rib arrays

For staggered rib arrays, triangular ribs show best heat transfer performance with average friction loss and for one sided rib arrays, wedge rib upstream ribs show best heat transfer performance. But friction loss doesn't depend only on the recirculation zone but mainly on the shape of the ribs. It is seen that both heat transfer performance and friction loss are dependent of Reynolds no

NOMENCLUTURE

D_h	Hydraulic diameter
C_P	Specific heat capacity of air
e	Rib height
f	Friction factor
H	Channel height
h	Average heat transfer co-efficient
K	Thermal conductivity of air
k	Turbulent kinetic energy
Nu	Nusselt number
P	Pitch
ΔP	Pressure drop
Pr	Prandtl number
Re	Reynolds number

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