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# Comparison of heat transfer performance of different types of ribbed surfaces

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***Abstract*—Jet impingement is a wide research field nowadays and lot of new work is going on in this field. Jet impingement achieves a high heat transfer rate because of which it has been used as an effective cooling technique in many industries. If a surface is provided with the ribs, heat transfer rate from a flat plate increases considerably. In this work, different rib designs are studied for their effect on heat transfer rate. CFD simulation has been conducted on flat surface without ribs and flat surface with V-shaped, X-shaped and Y-shaped ribs. Reynolds numbers selected are 7000, 9000 & 11000 to study its effect on heat transfer. Different Z/d ratios such as 6, 8, 10 were analyzed. Among all these geometries, best heat transfer coefficient is achieved by flat plate with Y-shaped ribs. As Reynolds number increases, heat transfer rate increases and as Z/d increases, heat transfer rate decreases for all three geometries. The best heat transfer rate is achieved with Y-shaped ribs.***

***Index Terms*— heat transfer coefficient, Jet impingement, ribs, Z/d ratio.**

## I. INTRODUCTION

Impingement heat transfer is considered as a promising heat transfer enhancement technique. Among all convection heat transfer enhancement methods, it provides significantly high local heat transfer coefficient. At the surface where a large amount of heat has to be removed, this technique can be employed directly through a very simple design involving a plenum chamber and orifices. Due to its widespread applications ranging from electronics equipment and turbine blade cooling to drying of textiles and glass tempering, impinging jets have been studied extensively in the literature.

Caliskan et al [2] have studied detailed heat transfer over a surface with V-shaped ribs (V-SR) and convergent-divergent shaped ribs (CD-SR) Best heat transfer performance was obtained with the V-SR arrangements as compared to flat plate and convergent-divergent rib arrangement. The average Nusselt number values for the V-SR plate showed an increase ranging from 4% to 26.6% over those for the smooth plate.

Lei Tan et al. [3], studied the convective heat transfer on the rib-roughened wall impinged by a row of air jets inside semi-confined channel. Three typical rib configurations, including orthogonal ribs, V-shaped ribs and inverted V-shaped ribs, were considered under different non-dimensional jet-to-target distances ranging from 1 to 3 diameters and impinging jet Reynolds numbers ranging from 6000 to 30,000. The results show that the rib-roughened wall enhances the convective heat transfer up to 30% in the ribbed region by comparison with the smooth wall under the same jet Reynolds number. Inverted V-shaped rib seems to be advantageous on the convective heat transfer enhancement, especially at lower jet-to-target spacing. At the jet-to-target spacing ratio of 1, the flow coefficient of the rib-roughened channel is decreased 5%-10% in related to the smooth channel. S. Caliskan [4] studied the flow field & heat transfer characteristics of smooth surfaces and surfaces with V-shaped ribs (V-SR) experimentally with a Laser-Doppler Anemometry (LDA) system. For low jet-to-plate spacing's, the production of turbulence kinetic energy is higher for the V-SR surfaces as compared to smooth surfaces. The Nusselt numbers are higher for the V-SR surfaces as compared to smooth surfaces.

Gau et al. [5], performed experiments to study slot air jet impingement cooling flow and the heat transfer along triangular rib-roughened walls. The effect of different rib protrusions (heights) on the impinging flow and heat transfer along the wall is studied, which is achieved by using different sizes of nozzles. The geometric shape of the triangular ribs is more effective in rebounding the wall jet away from the wall than in the case with rectangular ribs. The rebound of the jet away from the wall causes a significant reduction in the heat transfer. During the experiments, the Reynolds number varies from 2500 to 11,000, the slot width-to-rib height ratio from 1.17 to 6.67, and nozzle-to-plate spacing from 2 to 16.

Katti et al. [6] investigated the heat transfer enhancement from a flat surface with axisymmetric detached rib-roughеners due to normal impingement of circular air jet. The ratio of average Nusselt numbers of ribbed and smooth surface is seen to increase with Reynolds number. Correlation is developed for Nusselt numbers averaged upto an r/d of 1.5. Enhancements in heat transfer decrease for higher Z/d . Chougule et. al. [7] studied CFD multi-jet air impingement on flat plate. It is observed that in multi-jet impingement, spacing between the air jets play important role. A value of nozzle pitch of 3 to 5 was recommended to reduce adjacent jet interference.

Anwarullah et al. [9] conducted an experimental investigation to study the effect of nozzle-to-surface spacing of the electronic components and Reynolds number on the heat transfer in cooling of electronic components by an impinging submerged air jet. Local heat transfer rates at a fixed radial location are measured and the stagnation Nusselt numbers for different H/d ratios calculated. They are correlated and compared with the data of earlier investigators.

Lakshmi Prasad et al. [10] conducted numerical simulation for studying heat transfer in multi jet impingement on sparse and dense pin fin heat sinks. It is observed that as the Re increases the heat transfer rate also increases. It is also observed that the denser design produces higher rate of heat transfer than the sparse design in all the cases. It is noticed that the symmetry of heat transfer patterns occur in the minimum cross flow condition. More et al. [11] conducted experimental study of high heat removal by aluminum pin fin heat sink using multi-jet air impingement. The thermal performance of 4×4 array circular pin fin heat sink is studied with the thermal parameters like average Nusselt number, total thermal resistance, average base temperature, and Reynolds numbers.

From the literature survey it has been observed that, the heat transfer enhancement is achieved with the help different designs of ribs on a flat surface. Objective of the thesis is to enhance the heat transfer rate through flat plate by providing it with the different designs of ribs. Till now, different rib designs are tested to understand the heat transfer enhancement with ribbed surfaces as compared to the flat plate. In this paper, V-shaped ribs, X-shaped ribs and Y-shaped ribs are tested for heat transfer rate for different Re and Z/d ratio. Aim is to understand the heat transfer performance of different types of ribbed surfaces as compared to flat plate. Most efficient rib geometry is identified at specific Re and Z/d ratio. Temperature distribution and velocity contours are also studied in the thesis.

## II. NUMERICAL SIMULATION

Simulation of the different rib designs was done with the help of CFD simulation software. Following steps are followed in the procedure:

### *A. Computational Domain*

Initially, the geometry that is to be used for the CFD analysis is created in Ansys Workbench. Three types of geometries namely, flat plate, flat plate with V-shaped ribs, flat plate with X-shaped ribs were created. As the optimization process started, different alterations were done with X-shaped ribs. Also, a new type of rib design was invented which gave a better heat transfer performance i.e. flat surface with Y-shaped ribs. In the model creation, along with the geometry, domain is also created. Dimensions of the domain are decided by the Z/d ratio. “Z” is the distance between the flat surface and the nozzle of air. Value of Z/d is 6, 8 or 10 where, d is the diameter of the nozzle that is 5mm. So, the height of the domain is calculated accordingly. The size of the domain is 500mm X 300mm. It is decided so that it will cover entire fluid in the simulation. The computational domain is bounded by the solid domain (heat sink) and fluid domain (Air). The solution domain is filled with stagnant air. It is assumed that the heat is generated inside the heat sink at a uniform rate and can be represented by a constant heat flux from the bottom. If this flux coming out of the heat sink is not dissipated properly, the temperature will go up. The air jet is discharged through the round nozzle with length l and diameter d is directed normally towards the pin-fin heat sink with base 100 x 100 x 6mm. subjected to a constant heat flux from below (bottom) and except top surface all other walls are adiabatic. The jet after impingement will exit from opening with minimum cross flow conditions.

### *B. Mesh Generation*

After the creation of the model, next step in the analysis is to create meshing. Meshing is a process in which geometry is divided into a number of small parts. A model created in Ansys workbench was imported to Ansys ICEM for meshing. Mesh was generated for both plate and domain. The type of mesh element used is “tetrahedron” for solid part and domain both. Only in the boundary layer prism type mesh was used. The grid dependency study was carried out. The errors obtained from the study are as shown below:

**Table 1: Grid independence study**

| No. of elements | Error with respect initial case |
|---|---|
| 0.5 Millon | |
| 0.75 Million | 15% |
| 1 Million | 6% |
| 1.2 Million | 1% |

The error should come within 2%. So, approximately no. of elements chosen for the meshing is 1.2 Millions. Finer the mesh more accurate are the results but the operation time also increases significantly. Mesh generated as shown in figure:

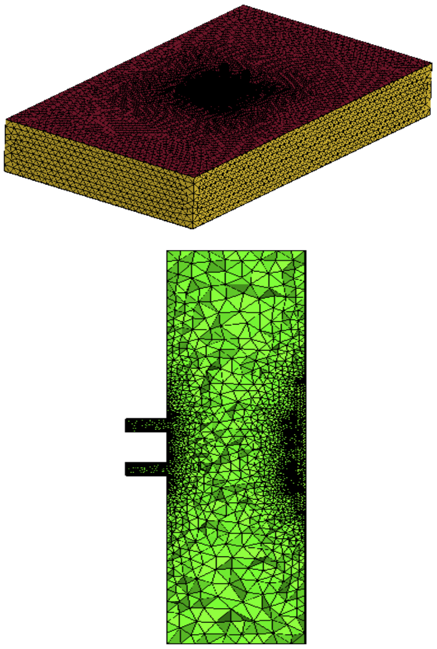


**Fig 1: Meshing**

### *C. Turbulence Model*

From the literature survey [7], it has been studied that, the suitable turbulence model for this case is K-w (SST) model since it solves the equations near boundary layer very efficiently as compared to remaining models. Also, the literature survey shows that, the error with this model is within 5%.

### *D. Application of Boundary Conditions*

Variables chosen for the analysis are: velocity and Z/d ratio. The effect of changing velocity of air jet and the effect of changing the distance between the jet and the plate are studied in this thesis. Pressure boundary condition is applied to the side faces of the computational domain. At nozzle inlet velocity boundary condition is applied. The top and bottom faces of the computational domain represent the two confining plates and adiabatic wall condition is applied to them. At the bottom of the heat sink, wall boundary condition is imposed with heat flux 5000W/m2 (50 watts for 100mmx100mm base plate). The side walls heat sink is assumed adiabatic thermal boundary condition state. The flow is assumed to be steady, incompressible and three-dimensional. The buoyancy and radiation heat transfer effects are neglected and thermo physical properties of the fluid such as density, specific heat and thermal conductivity are assumed to be constant. It is assumed that the heat is generated inside the heat sink at a uniform rate and can be represented by a constant heat flux from the bottom. In the fluid domain the inlet boundary condition is specified the measured velocity and static temperature (298K) of the flow were specified at the inlet of the nozzle. No-slip condition was applied to the wall surface. In fluid domain there is also opening boundary condition in which flow regime is subsonic, relative pressure is 0 Pa with the details of operating temperature (298 K) and the turbulence intensity of 5%.

The values of Z/d ratio selected are 6,8 and 10 where ‘d’ is a nozzle diameter which is 5mm (0.005m) and Z is the distance between the flat surface and the nozzle.

For Z/d=6, Z=30mm
For Z/d=8, Z= 40mm
For Z/d=10, Z=50mm

From the previous studies it is seen that, the nozzle plate-to-heated surface separation distance (Z) significantly affects thermal transport due to adjacent jet interference (A. M. Huber (1994), J. Y. San (2001)). Nusselt number value ($Nu_{avg}$) reaches maximum at about Z/d equal to 6. It is also found that the length of potential core is about 5 times the jet nozzle diameter. Beyond the potential core, with further increase in axial distance, the interaction between the attenuation of approaching jet velocity and the continuous increase in centerline turbulence intensity brings about a maximum heat transfer coefficient at Z/d = 6.

### *E. Results and Discussions*

Initially CFD simulations were run on three types of geometries, namely; flat plate, flat plate with V-shaped ribs, and flat plate with X-shaped ribs. Earlier the estimation was that, we would get the better heat transfer performance with X-shaped ribs as compared to smooth flat plate and V-shaped ribs since the heat transfer area is more for X-shaped ribs. For the comparison purpose, one case of each was run with Re=7000 and Z/d=6. Heat transfer coefficients were obtained from CFD for all these cases as shown below:

**Table 2: Heat transfer rate at Z/d=6 and Re=7000**

| Flat plate (W/m2K) | Flat plate with V-shaped ribs (W/m2K) | Flat plate with X-shaped ribs (W/m2K) |
|---|---|---|
| 86 | 158 | 118 |

Temperature profiles for these three types of surfaces

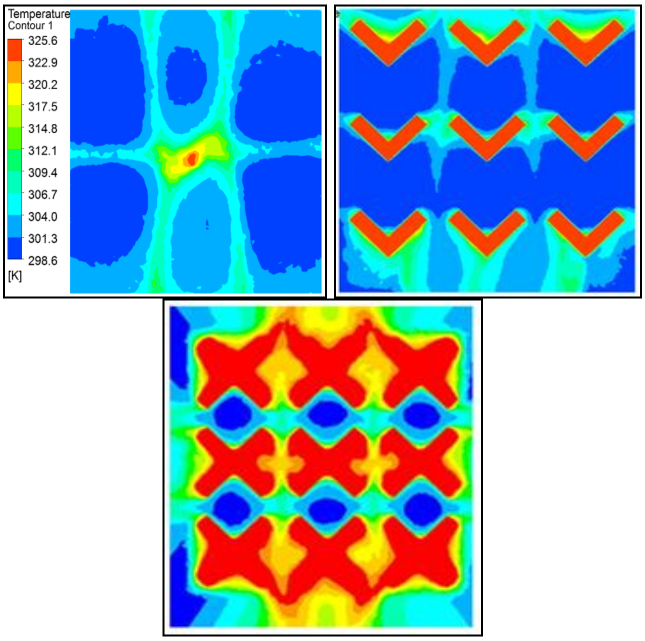


**Fig 2: Temperature profiles of flat plate and flat plate with V-shaped ribs and X-shaped ribs**

The temperature profiles show that, there is no uniform temperature distribution on the plate with X type of ribs. This happens because there is a restriction to the flow of fluid in X-type. Therefore, though X-type ribs are having higher surface area, heat transfer rate is not greater in case of X as compared to V-type ribs. To make the fluid distribution more uniform, X-shaped ribs are interrupted in the middle. It will allow the fluid to flow through the X-shaped ribs. It will lead to more uniform distribution and more uniform cooling of the plate. The velocity contours for all these V-type rib and X-type rib are as given below:

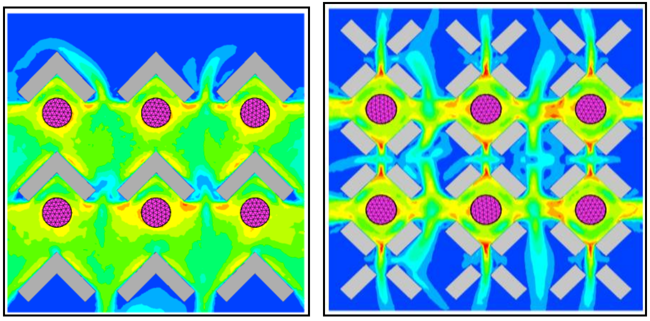


**Fig 3: Velocity contours for V-type ribs and interrupted X type ribs**

Above velocity contours reveal that, the flow distribution is improved in interrupted X-type rib as compared to earlier X-type rib. The heat transfer coefficient is also improved compared to X-type, but it is still less than V-type ribs. To improve the design further and to make the fluid distribution more uniform over the plate another type of design that is Y-type rib is tried. The velocity contour for Y-type of rib is as shown below. In this type, impingement point is at the center of three arms of Y-type ribs, so that the flow spreads more uniformly on the plate.

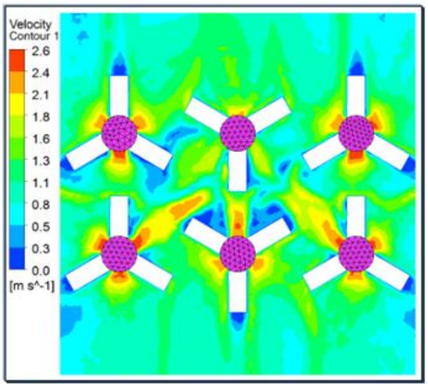


**Fig 4: Velocity contour in Y-type rib**

It can b observed in the above figure that, the stagnant fluid region is very small. Summary of values of heat transfer rates for all the above mentioned cases are as given below. These are the values for Z/d ratio of 6 and Re 7000:

**Table 3: Heat transfer rate at Z/d=6 and Re=7000**

| Flat Plate | V-type ribs | X-type ribs | Interrupted X-type | Y-type ribs |
|---|---|---|---|---|
| 86 | 158 | 118 | 134 | 188.5 |

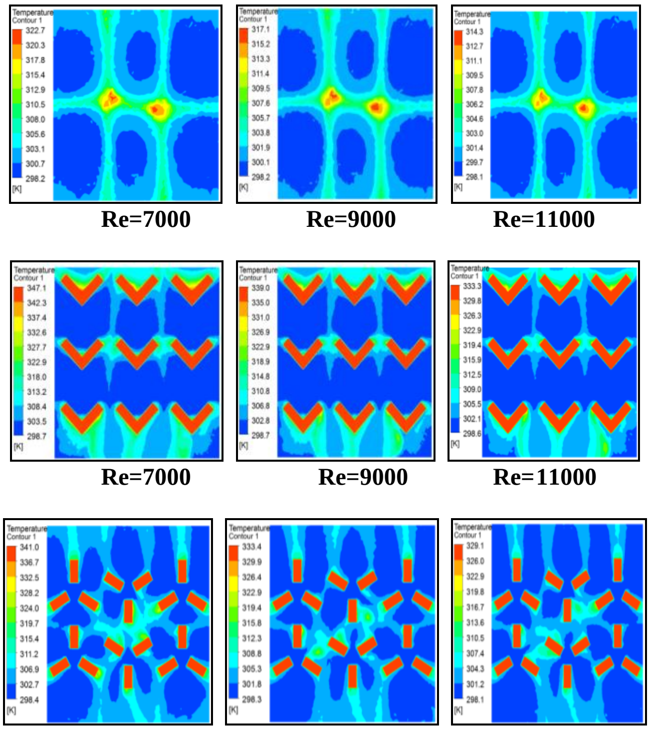
From above table it reveals that, the Y-type ribs are even better than V-type ribs.

Based on these results, the further investigation of ribs is done on flat plate, V-type ribs and Y-type ribs. The effect of Reynolds number and Z/d ratio is studied on these three geometries. Cases were run to know the effect of these parameter. For further investigation, V-shaped ribs and Y-shaped ribs were selected since they are having best heat transfer performance among all rib designs. V-shaped ribs and Y-shaped ribs were compared with the flat plate. Effect of Reynolds number and Z/d ratio was investigated. Three Reynolds numbers and three Z/d ratios were selected for the investigation. Following simulations were conducted:

**Table 4: CFD simulation cases**

| Geometry | Z/d ratio | Reynolds number |
|---|---|---|
| Flat plate | 6 | 7000 |
| | 6 | 9000 |
| | 6 | 11000 |
| | 8 | 7000 |
| | 8 | 9000 |
| | 8 | 11000 |
| | 10 | 7000 |
| | 10 | 9000 |
| | 10 | 11000 |
| V-shaped ribs | 6 | 7000 |
| | 6 | 9000 |
| | 6 | 11000 |
| | 8 | 7000 |
| | 8 | 9000 |
| | 8 | 11000 |
| | 10 | 7000 |
| | 10 | 9000 |
| | 10 | 11000 |
| Y-shaped ribs | 6 | 7000 |
| | 6 | 9000 |
| | 6 | 11000 |
| | 8 | 7000 |
| | 8 | 9000 |
| | 8 | 11000 |
| | 10 | 7000 |
| | 10 | 9000 |
| | 10 | 11000 |

Velocity and temperature contours are discussed below. The temperature contours are shown below for Z/d=6



Re=7000 Re=9000 Re=11000

Re=7000 Re=9000 Re=11000

Re=7000 Re=9000 Re=11000

**Fig 5: Temperature contours for Z/d=6 at different Reynolds numbers**

It is noticed that, as Reynolds number increases temperatures decrease for all three cases. If the geometries are compared with each other, the maximum temperatures are found for V-type ribs and Y-type ribs. But if the high temperature regions are observed, it is near rib areas for both cases, so that can be ignored. Whereas in flat plate it is on the flat surface only. Velocity contours are as shown below:

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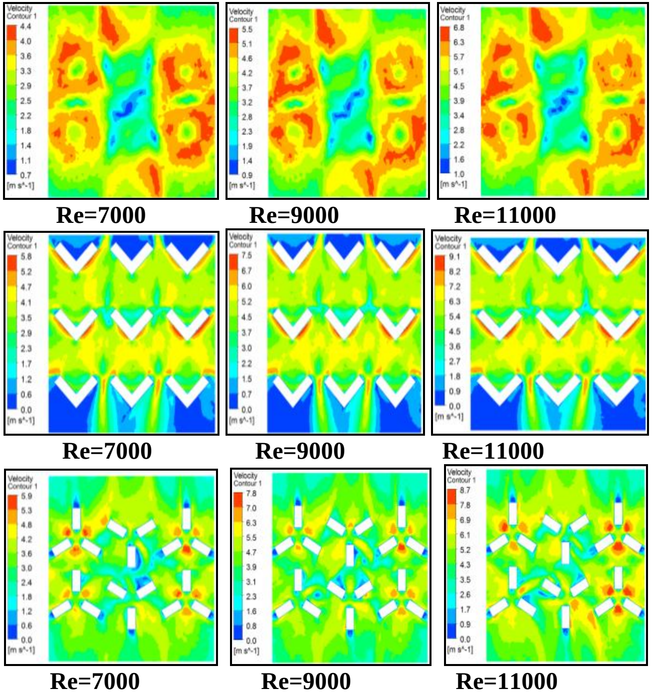
Re=7000 Re=9000 Re=11000
Re=7000 Re=9000 Re=11000
Re=7000 Re=9000 Re=11000

**Fig 6: Velocity contours for Z/d=6 at different Reynolds numbers**

The velocity contours show that, fluid distribution is most uniform for the case of Y-shaped ribs. There is a very small area in which the fluid is stagnant. In V-type ribbed surface, fluid does not reach the edges. Blue area indicates the stagnant fluid. In Y-shaped ribbed surface, the fluid reaches almost every portion of the flat plate with some velocity. As much as the velocity is more, heat transfer rate increases.

From the simulations, it is revealed that, for Z/d=6, the temperatures are lowest as compared to Z/d=8 and Z/d=10.

At constant Z/d ratio, as Re increases, velocity of fluid over the flat plate increases.

***Heat Transfer Rates:***

Overall Heat transfer rates of flat plates are obtained for all cases from the simulation are as shown in table 5:

**Table 5: Heat transfer rates of flat plate & ribbed surfaces**

| Flat Plate | | | |
|---|---|---|---|
| Heat Transfer Coefficient, W/m2-K | | | |
| Z/d : Re | 7000 | 9000 | 11000 |
| 6 | 86.5 | 104.4 | 122.1 |
| 8 | 83.6 | 100.4 | 117.4 |
| 10 | 75.4 | 92 | 105 |

| V-Type Ribs | | | |
|---|---|---|---|
| Heat Transfer Coefficient, W/m2-K | | | |
| Z/d : Re | 7000 | 9000 | 11000 |
| 6 | 158.3 | 190.9 | 222.5 |
| 8 | 152 | 183.3 | 214.2 |
| 10 | 137.6 | 164.4 | 191.1 |

| Y Type Ribs | | | |
|---|---|---|---|
| Heat Transfer Coefficient, W/m2-K | | | |
| Z/d : Re | 7000 | 9000 | 11000 |
| 6 | 188.5 | 228.9 | 260.1 |
| 8 | 172.6 | 205.2 | 239.7 |
| 10 | 162.4 | 190.8 | 219.1 |

It is observed that, the heat transfer rates increase as Reynolds number increases. As Z/d ratio increases, distance between the nozzle and flat plate increases. As a result, heat transfer rate decreases. If the different types of surfaces are compared, Y-type ribs perform the best among all three cases.

## III. CONCLUSIONS

1. Heat transfer rate increases in ribbed surfaces as compared to smooth flat surface. In case of ribbed surfaces, heat transfer rate increases by 83% in case of V-shaped ribs as compared to smooth plate without ribs. In case of Y-shaped ribs, heat transfer rate increases by 118% compared to smooth flat plate.
2. As Reynolds number increases, heat transfer rate increases in all cases.
3. As Z/d ratio increases, distance between nozzle and flat surface increases. As a result, heat transfer rates decrease in all cases.
4. Y-shaped ribs are the best for heat transfer. In Y-shaped ribs, the distribution of fluid is uniform along the flat plate. Also the temperatures achieved are lower in ribbed surfaces.
5. V-shaped ribs and Y-shaped ribs have same surface areas. Y-shaped ribs give better heat transfer performance with same heat transfer area. So, cost saving can be achieved with Y-shaped ribs.

## IV. FUTURE ENHANCEMENT

In this paper, it has been proved that, the heat transfer rate improves significantly with Y-shaped ribs as compared flat plate without ribs. The effect of Z/d ratio and Reynolds number is studied in this thesis. Following study can be taken up for the further study:

1. In this work, the angle between rib arms in Y shape is 120 deg. There is a scope to change this angle.
2. Trials can be taken by changing other geometrical parameters of Y-shaped rib like height, width of the arms.
3. The effect of pitch variation on heat transfer can also be studied in future study.
4. No. of Y-shaped ribs can be varied and its effect on heat transfer rate can be studied.
5. The material of a plate used in this thesis is Aluminum. Effect of other materials can be studied.
6. Other rib designs can be studied to further increase the heat transfer rate by eliminating the discrepancies in the current rib design

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