Numerical Investigation of Heat Transfer Characteristics Using Different Discrete Rib Arrangements

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Abstract

Numerical study was carried out to study effect of different discrete rib arrangements on heat transfer performance. These rib arrangements include variously angled ribs, different. Spacing and also different angle combinations. Standard k-ε model is used for turbulence modelling. This study is carried out at Reynolds no. 22384 which was selected from previous studies. Different rib arrangements like inline and staggered for 30º, 45 and 60, discrete V's arrangement with different spacing, diamond like discrete rib arrangement with half and complete obstruction, inline and staggered connected V's and angle combination rib arrangements are studied numerically in this study. It is observed that 45 inline V-shape rib arrangement shows better Thermal Enhancement Factor (TEF) as compared to other arrangements and value of TEF is observed to be 1.0011. Ribs arranged in discrete V's shape with different spacing shows comparatively less heat transfer enhancement.

Keywords- Rib Arrangement, Heat Transfer Enhancement, Rib Spacing, Rib Angle

I. INTRODUCTION

A heat exchanger is a device that provides the transfer of thermal energy between two or more fluids, which are at different temperatures and are in thermal contact with each other. Heat exchangers are used for different applications as power generation, refrigeration, ventilating and air-conditioning systems, process, manufacturing, aerospace industries, electronic chip cooling etc. The traditional methods of reducing the air-side thermal resistance are by increasing the surface area of the heat exchanger, or by reducing the thermal boundary layer thickness on the surface of the heat exchanger. Increasing the surface area is effective but it results in the increase in material cost and increase in mass of the heat exchanger. Heat transfer rate is directly proportional to the turbulence of the fluid flowing over the heated surface. The active heat transfer enhancement methods are as cost of increased pumping power (stirring the fluid, vibrating the surface etc.) while in passive heat transfer enhancement methods no external power source is required (dimples, pin-fins, perforated baffles, twisted tapes, vortex generators etc.). The subject of heat transfer enhancement is of serious interest in the design of compact heat exchangers. The emphasis is given to minimizing the space occupied by the equipment for the desired rate of heat transfer. The Artificial roughness is used as turbulence promoters on a surface. It is also the technique to enhance rate of heat transfer to the flowing fluid in a testing duct. The surface roughness can be created by number of methods such as welding, fixing small ribs, fixing small diameter of wires, machining, and sand blasting, casting and forming.

The use of artificial roughness rib elements on the absorber plate is one of the effective ways which enhances the heat transfer coefficient of the air, thus increasing the heat transfer rate. These roughness rib elements break up the boundary layers and induces turbulence which results in heat transfer enhancement. These roughness elements being smaller in height as compared to duct size causes turbulence in the laminar sub layer adjacent to the wall without affecting the main turbulent zone in the flow. Efforts for enhancing heat transfer have been directed towards artificially destroying laminar sub-layer. Artificial roughness creates turbulence near wall and breaks laminar sub-layer. However artificial roughness results in high frictional losses leading to more power requirement for fluid flow. Hence turbulence has to be created in a region very close to heat transferring surface. Core fluid flow should not be unduly disturbed to limit pumping power requirement. This is achieved by keeping height of roughness element small in comparison to duct dimensions. The performance of the heat transfer surface with ribs depends significantly on the parameters of the flow structure, such as reattachment length of the separated streamline and turbulence intensities, as well as the area of the surface.

The V-shaped ribs have better thermal and hydraulic characteristics than transverse rib. For Reynolds number of 4000-40,000 it has found that the use of in-line ribs provides considerable heat transfer augmentations, the thermal efficiency of the roughened duct air heater being 6–26% higher than that of a smooth duct air heater the highest advantage is at the lowest flow rate (Karwa, et al, 2013) [1]. The study is done V-discrete ribs with different flow orientations like forward flow, backward flow and transverse flow. Backward flow orientation shows better heat transfer performance. This study is made for Reynolds no. range of 7000 to 30000. About 31% enhancement is observed as compared to smooth plate. (Jadhav, et al, 2016) [2]. Numerical investigation was carried out for internal v ribs carried out. Reynolds no. range was 4000 to 40000. It has been observed that V ribs arranged inline gives better results (Kumar, et al 2014) [3]. Investigation with turbulent flow and V-shaped rib configurations with three different inclinations (60, 45 and 30 degree) were studied and compared to the perpendicular (90 degree) rib case. The better results are obtained with inclination of 45(Fang, et al, 2015) [4]. A comparative experimental study on the heat transfer characteristics of steam and air flow in rectangular channels roughened with parallel ribs was conducted. Reynolds number for both coolants ranges from 3000 to 15,000, the rib spacing ratios were 8, 10 and 12, and rib angles were 90, 75, 60, and 45 degrees respectively. The heat transfer enhancement of both steam and air increased with decreasing the rib angle from 90 to 45 degree (Chao Ma, et al,2015) [5]. The performance of V and W ribs studied for different pitch to rib height ratios. The Reynolds number was varied from 5000 to 35,000 The rib height to mean duct hydraulic diameter ratio (e/Dh, m) was kept constant at 0.08, Study for three pitch to height ratios (P/e) equal to 6, 10 and 17.5 were reported for straight and V ribs. The optimum P/e ratio based on constant pumping power thermal performance criterion was observed to be equal to 10 (Abraham, et al,2016) [6]. The effect of duct aspect ratio on heat transfer and friction characteristics was investigated experimentally and numerically. Four different duct aspect ratios (AR) were studied: AR = 1/4, 1/2, 1/1 and 2/1, the corresponding hydraulic diameter (D) was 32.0 mm, 53.33 mm, 40.0 mm and 53.33 mm, and the rib height-to-hydraulic diameter ratio (e/D) was 0.078, 0.047, 0.0475 and 0.047, respectively. The rib pitch-to-rib height ratio (p/e) was kept 10 for all the ducts. The investigated Re number ranges from 10,000 to 80,000 The averaged heat transfer coefficient ratio of steam was higher by 12–25% than air at the same test conditions (Shui, et al, 2013) [7].

II. RIB GEOMETRIES

Different rib geometries to be studied numerically are explained below. Fig.1 to Fig.17 shows different rib geometries like inline and staggered for different angles, discrete V's with different spacing, diamond like discrete arrangement with half and complete obstruction, discrete connected V's, ribs arranged in angle combinations. These rib arrangements are to be studied numerically and results to be compared with each other for heat transfer characteristics.

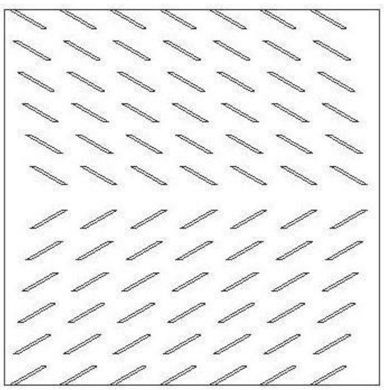

Fig. 1: 30 Inline

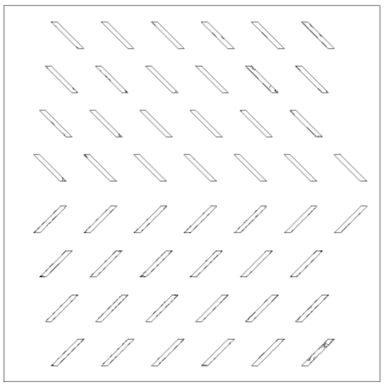

Fig. 2: 45 Inline

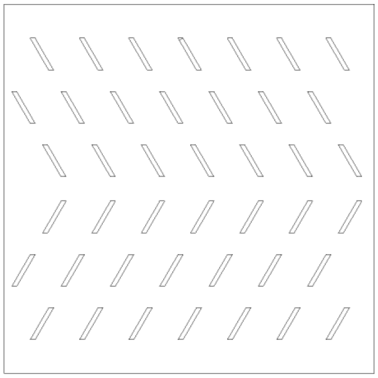

Fig. 3: 60 Inline

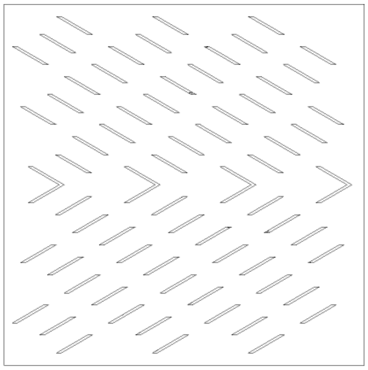

Fig. 4: 30 Staggered

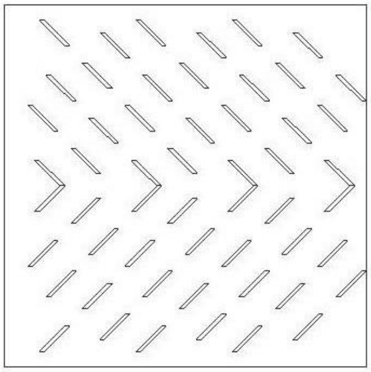

Fig. 5: 45 Staggered

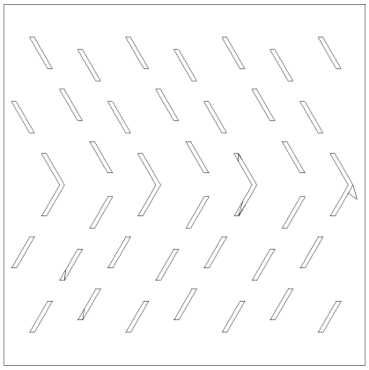

Fig. 6: 60 Staggered

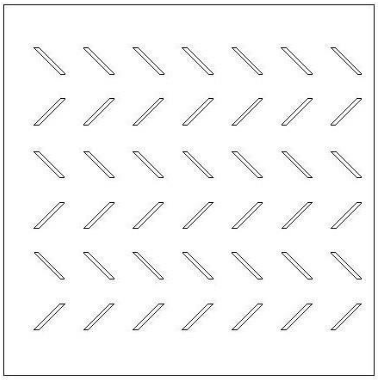

Fig. 7: Discrete V's

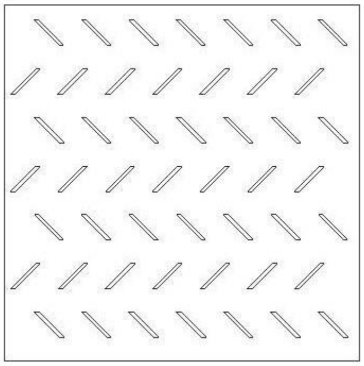

Fig. 8: Discrete V's 5mm down 10mm Distance

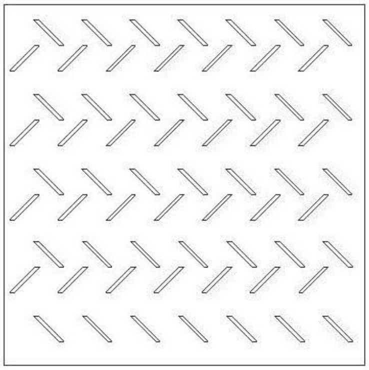

Fig. 9: Discrete V's 5mm down 0mm Distance

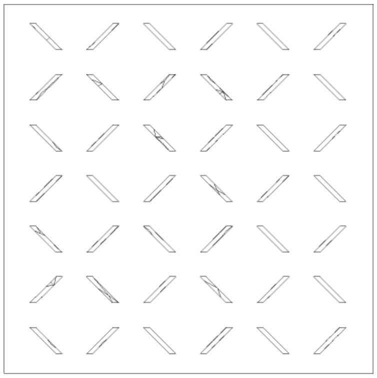

Fig. 10: Diamond

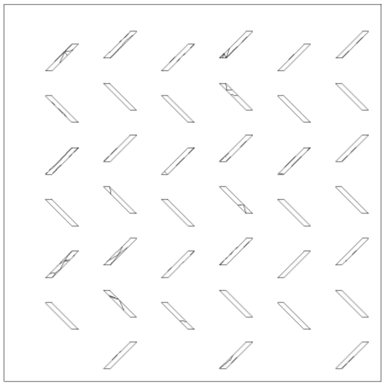

Fig. 11: Diamond Rib with Half Obstruction

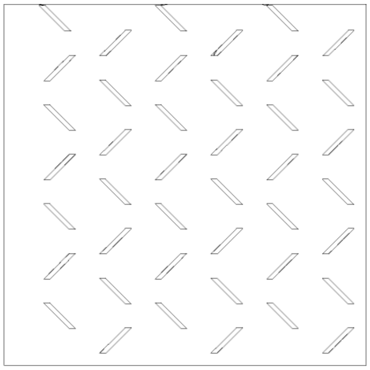

Fig. 12: Diamond Rib with Complete Obstruction

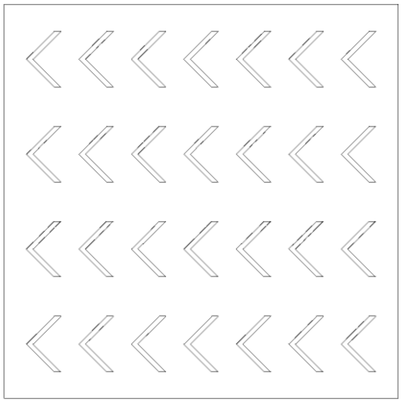

Fig. 13: Discrete Connected Inline V's

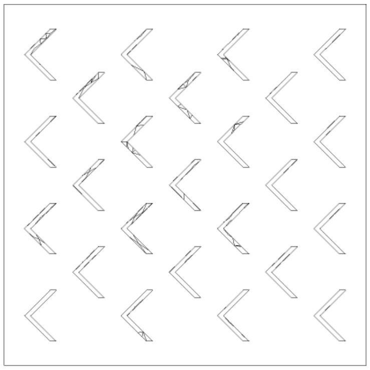

Fig. 14: Discrete Connected Staggered V's

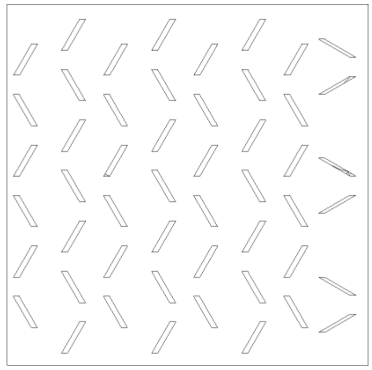

Fig. 15: 60-30

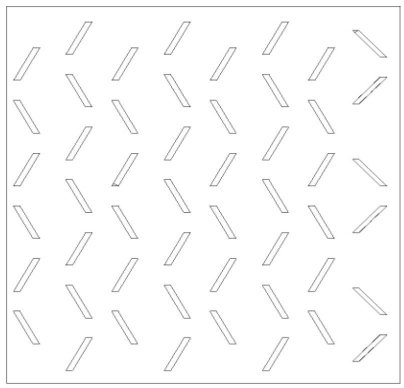

Fig. 16: 60-45

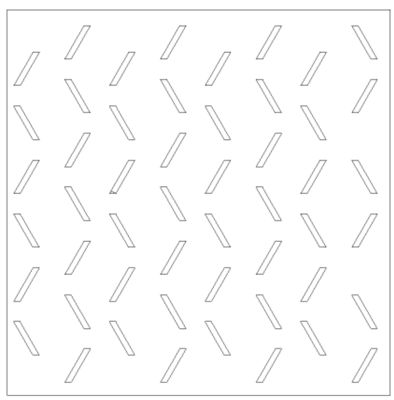

Fig. 17: 60-60

III. TURBULENCE MODELLING AND COMPUTATIONAL PROCEDURE

Standard k-ε model is one of the most widely used models in various applications. It is two equation semi empirical model with one transport equation for turbulent kinetic energy (k) and other for dissipation of turbulent kinetic energy (ε). [6]

Parameters considered in this study are Reynolds no.(Re), Nusselt no. (Nu) and Thermal Enhancement Factor (TEF). These parameters can be defined as below,

Reynolds no. (Re):

$$Re = \frac{\rho v D}{\mu}$$

Nusselt no. (Nu):

$$Nu = \frac{h D_h}{k}$$

Thermal Enhancement Factor (TEF):

$$TEF = \frac{Nu/Nu_0}{(f/f_0)^{1/3}}$$

Where Nu_0 and f_0 indicates Nusselt no. and friction factor for smooth plate respectively.

A. Meshing and Grid Independent Study

The tetra/mixed type of mesh is used for this computational model. The mesh density is selected at rib element on test plate. For tetra/mixed mesh there are three methods available as robust, quick (Delaunay) and smooth. Delaunay method is robust and fast for meshing of complex meshing. The grid independence study is done over the different no. of cells. The characteristics of grids 99,128, 1, 38,263 and 1, 81,028 and 2, 14,155 cells are used for simulation. The variation in nusselt number and friction factor values is negligible when increasing no. of cells from 1, 81,028 to 2, 14,155.

IV. RESULT AND DISCUSSION

Numerical study was carried out for all rib arrangements as explained above. This study was carried out at constant heat flux condition and at Re 22384. This Reynolds no. was selected from previous study as Nusselt no. (Nu) ratio was maximum in previous study for this Reynolds no. (Re). Flow structure and temperature profiles are as presented below for each arrangements.

A. Temperature Contours

Fig. 18 to Fig. 34 shows the temperature contours for all the rib arrangements. In Fig.18 to Fig. 23 temperature contours for ribs arranged in 30^0, 45^0 and 60^0 with inline and staggered arrangements are presented. Here it is observed that thickness of thermal boundary layer is increased with angle as obstruction to flow increases increasing turbulence. In Fig.24, Fig.25 and Fig. 26 temperature contours for discrete V's arrangements with different spacing is shown. Fig. 27, fig. 28 and Fig. 29 shows temperature contour for diamond like rib arrangements with providing half and complete obstruction. From these contours it is observed that thermal boundary layer has more thickness than remaining. This due to more obstruction is provided to the flow creating more turbulence and hence allowing more mixing of air with more air coming in contact with heated surface.

In Fig.30 and Fig. 31 ribs are arranged in discrete connected V's arrangements inline and staggered respectively. Here for inline arrangements thickness of thermal boundary layer is observed to be more indicating more heat transfer from surface to air as more air comes in contact with heated surface. Fig. 32, Fig 33 and Fig.34 presents the temperature contours for ribs arranged in angle combinations. Here only initial line of ribs has different angle and remaining ribs have same rib angle providing complete obstruction to flow. Here pattern of thermal boundary layer is observed to same for all these three arrangements. Thus, temperature contours for all rib arrangements are explained indicating mixing of hot air with cool air. Thickness of the thermal boundary layer is more when ribs provide obstruction to flow and creates more turbulence when air passes over ribs.

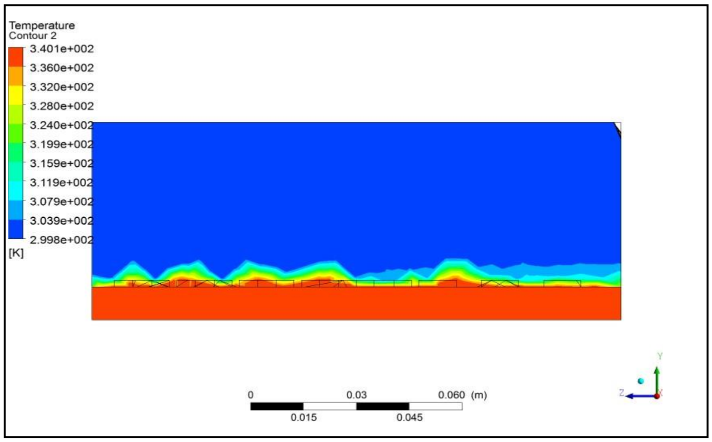

Fig. 18: 30 Inline

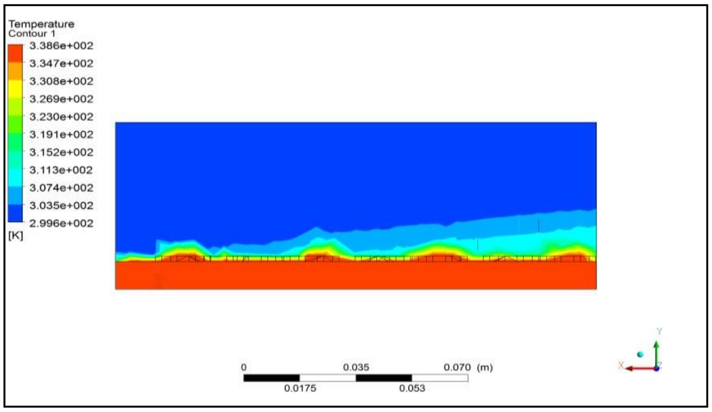

Fig. 19: 45 Inline

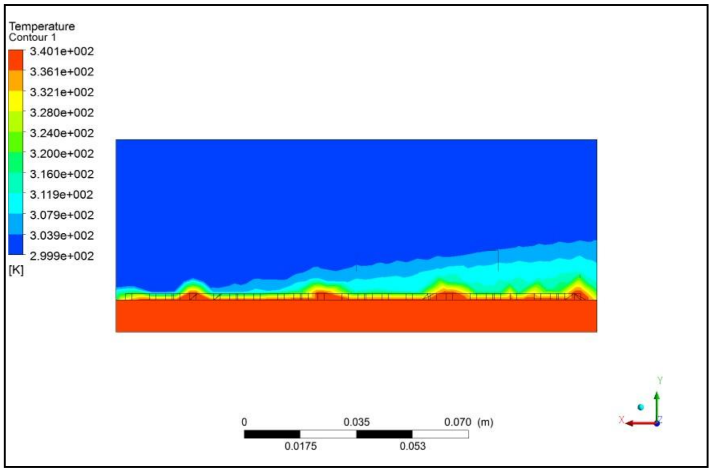

Fig. 20: 60 Inline

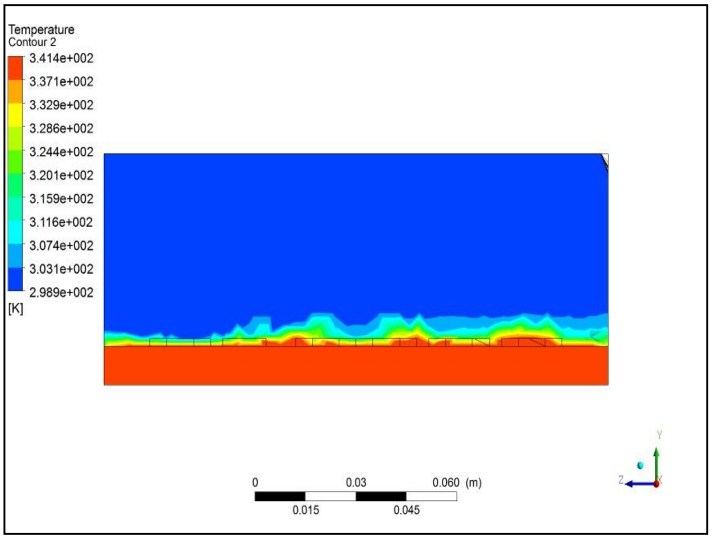

Fig. 21: 30 Staggered

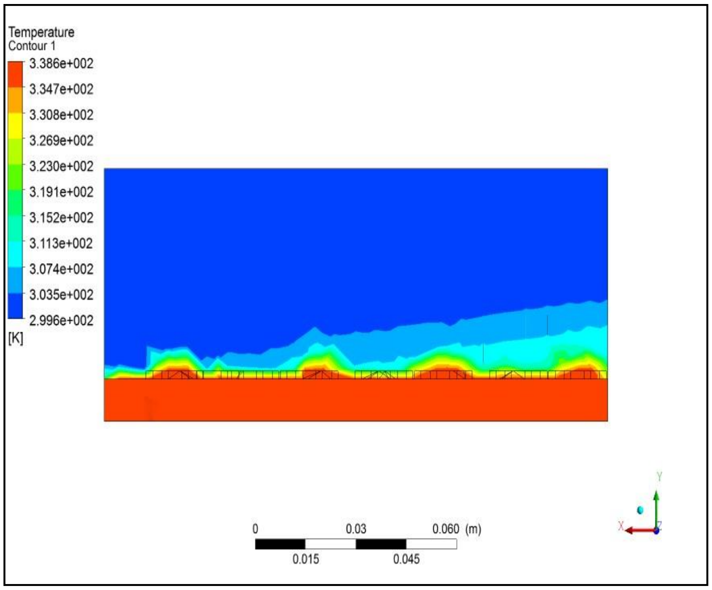

Fig. 22: 45 Staggered

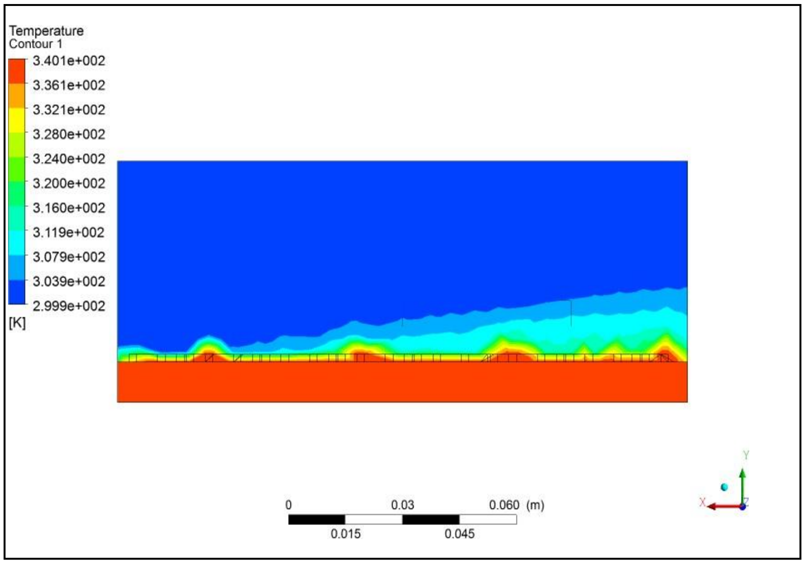

Fig. 23: 60 Staggered

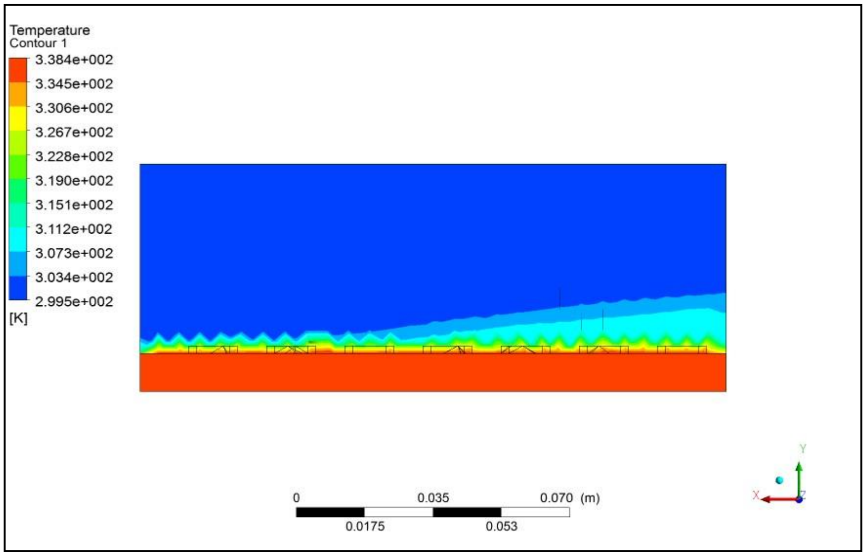

Fig. 24: Disc V's

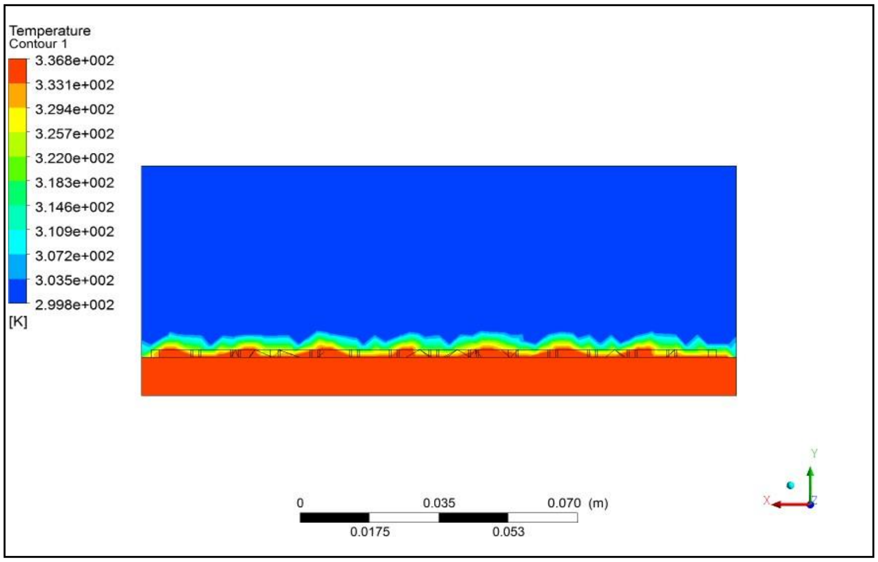

Fig. 25: Disc V's 5, 10

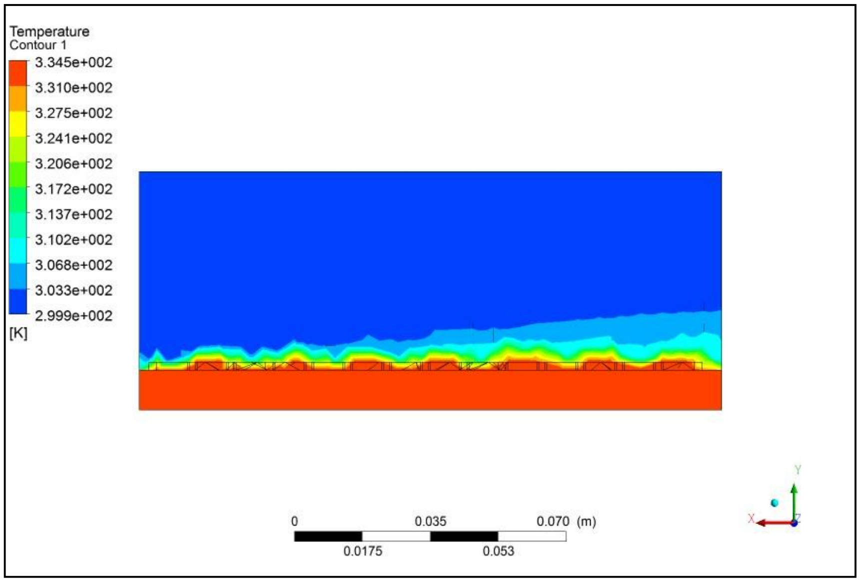

Fig. 26: Disc V's 5, 0

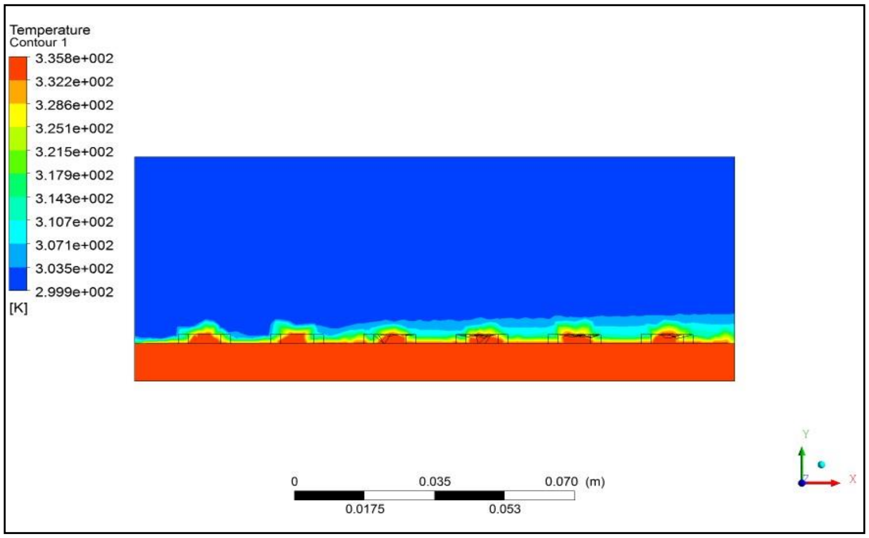

Fig. 27: Diamond

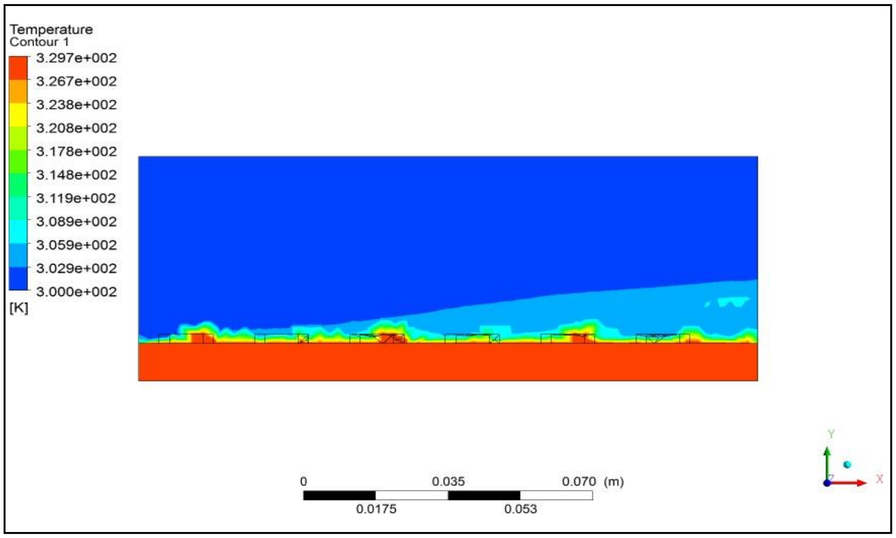

Fig. 28: Diamond Half Obstruction

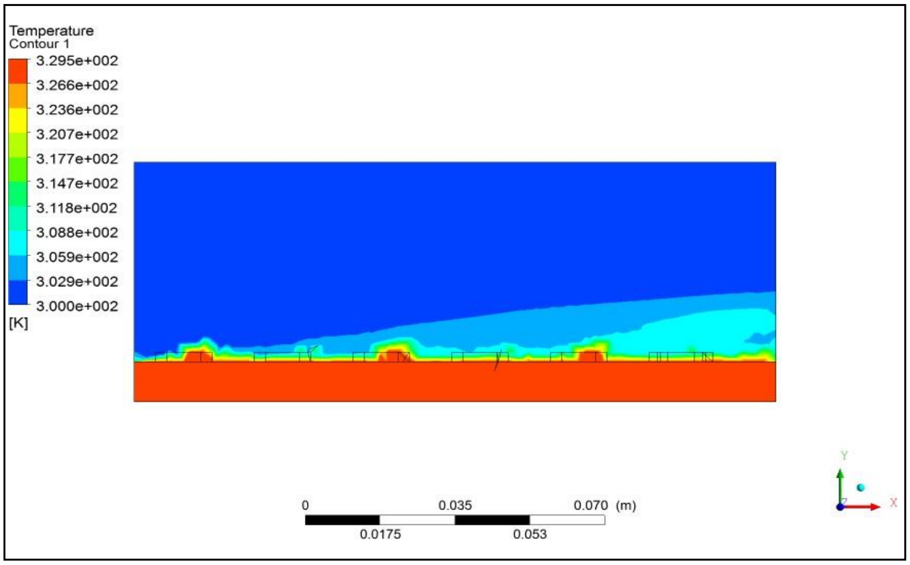

Fig. 29: Diamond Complete Obstruction

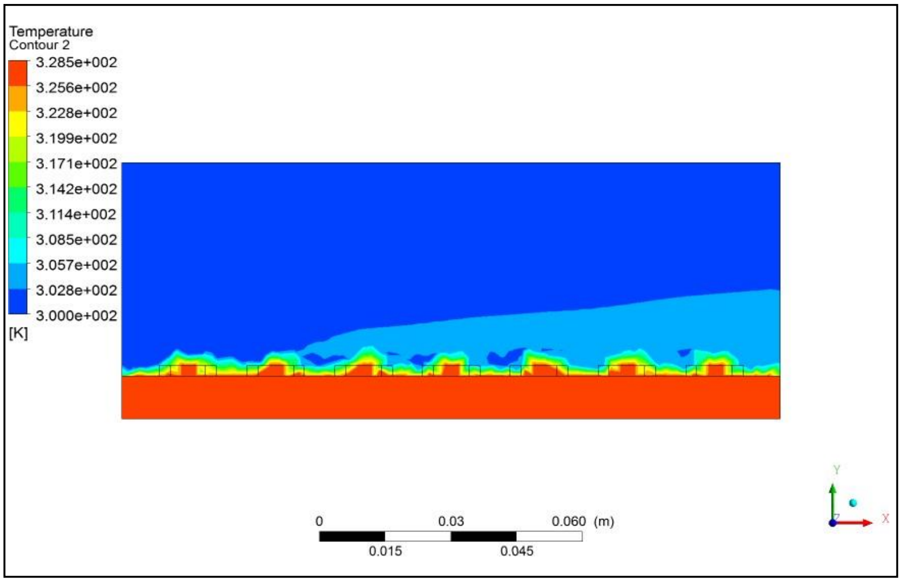

Fig. 30: Discrete Connected V's Inline

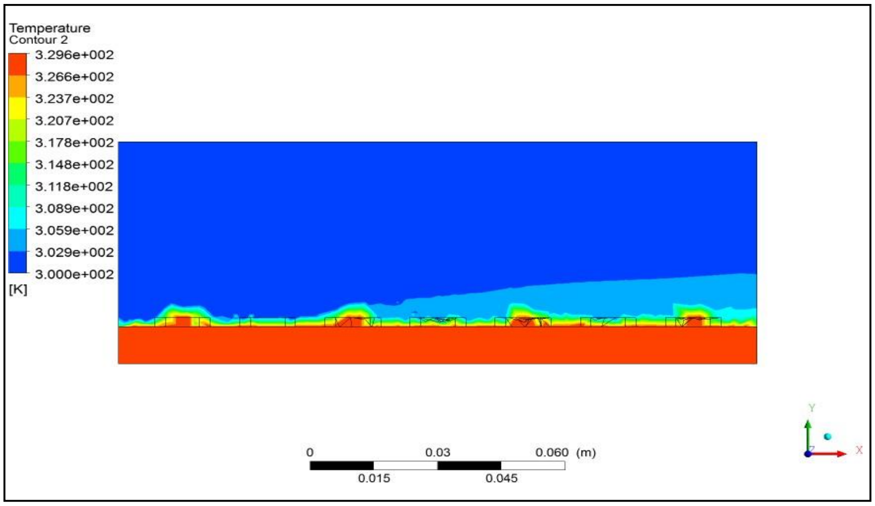

Fig. 31: Discrete Connected V's Staggered

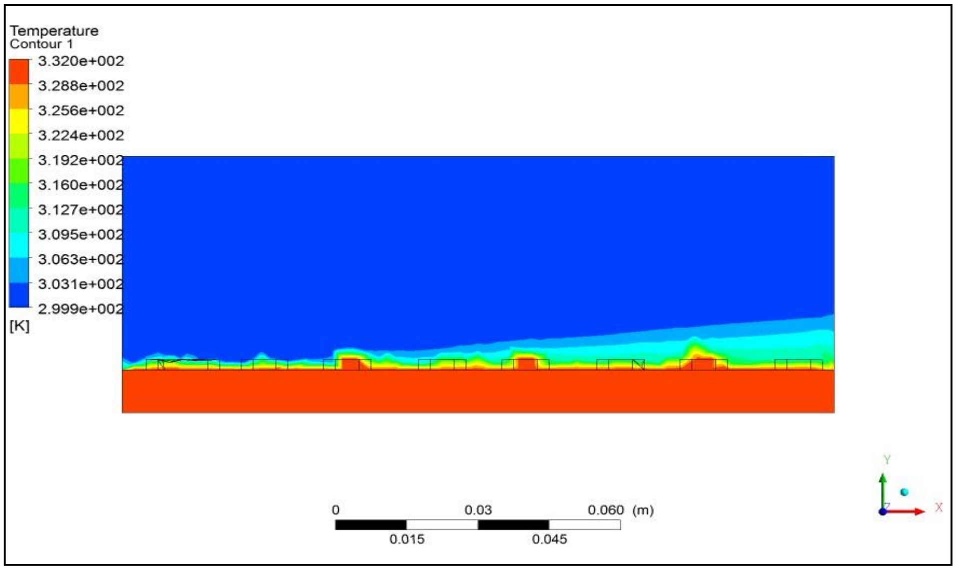

Fig. 32: 60-30

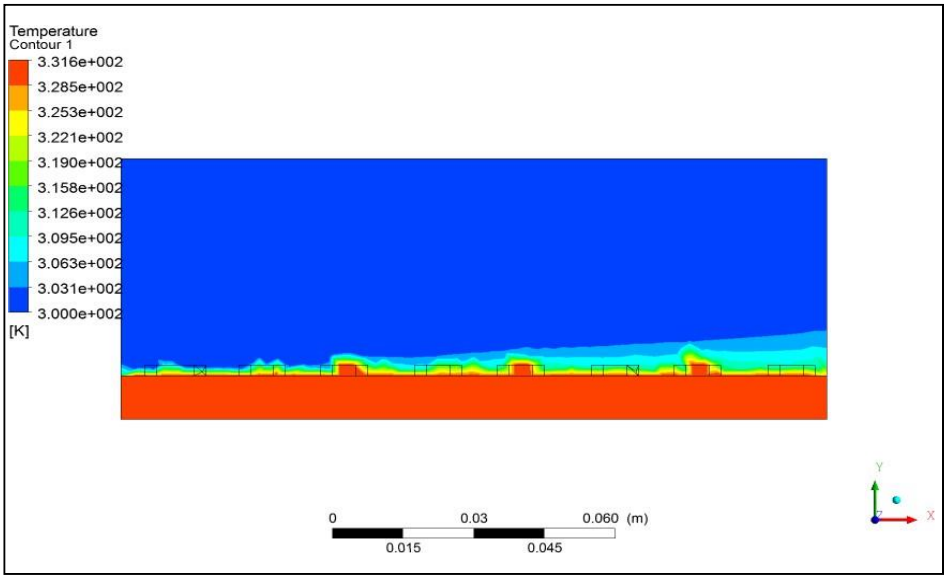

Fig. 33: 60-45

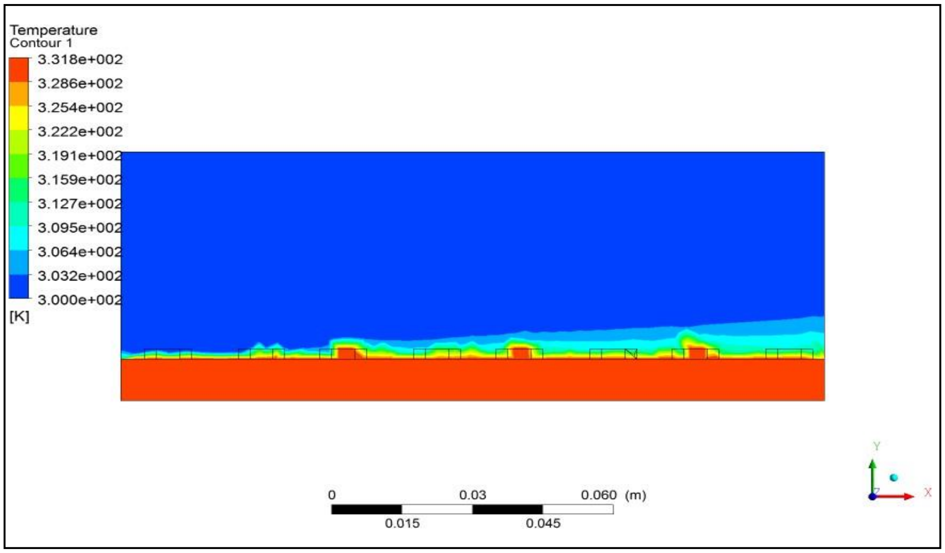

Fig. 34: 60-60

B. Velocity (Vector) Contours

Fig. 35 to Fig 51 shows velocity vector contours (top view) for all rib arrangements. Rib arrangement and the rib angle has effect on the turbulence created by ribs when air passes over it. Also it can be observed that turbulence is observed more when more obstruction is provided to flow by the ribs, when more turbulence is created more air gets mixed with hot air near surface and cool air. This increases rate of heat transfer from heated surface to air flowing over it.

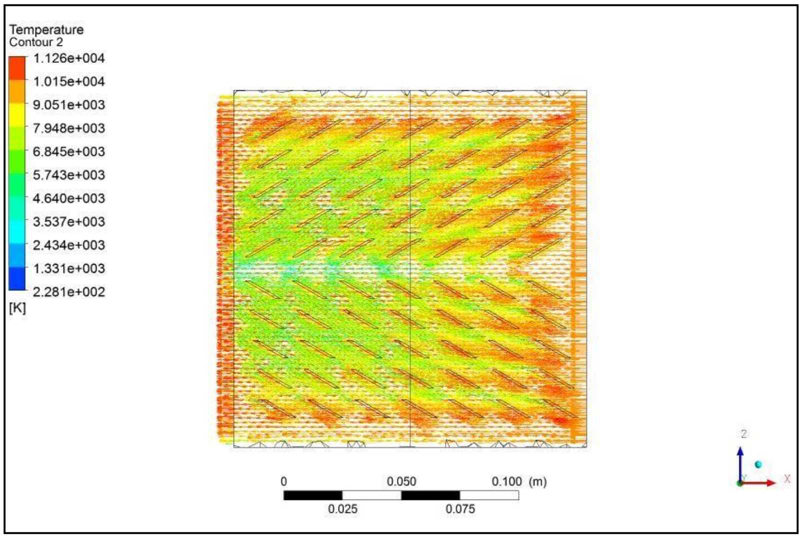

Fig. 35: 30 Inline

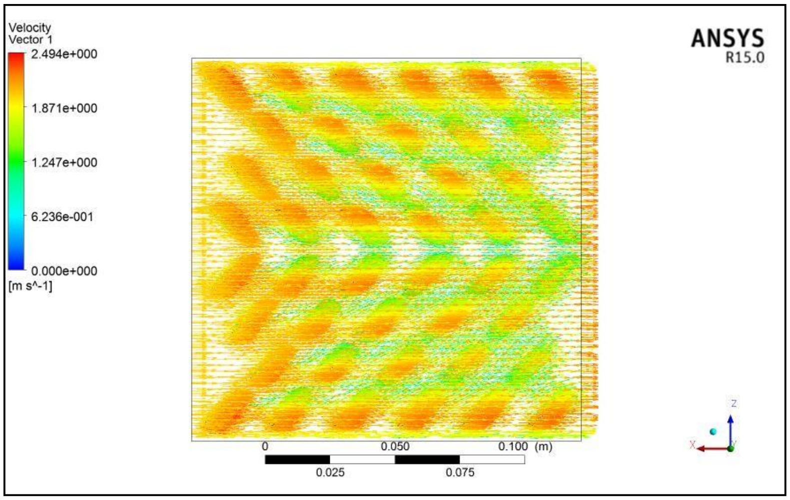

Fig. 36: 45 Inline

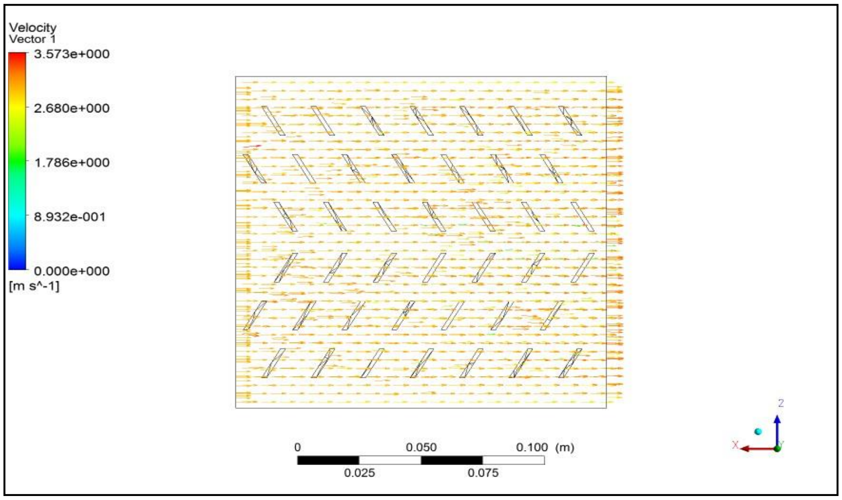

Fig. 37: 60 Inline

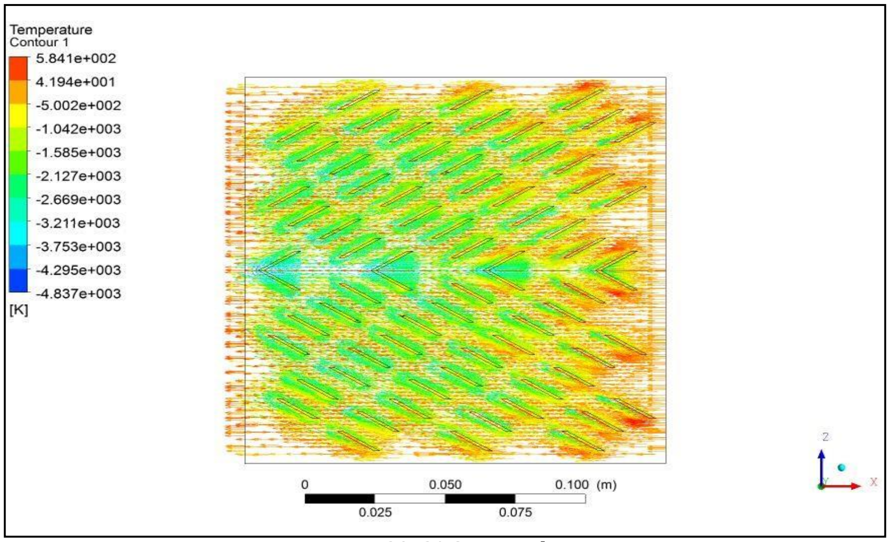

Fig. 38: 30 Staggered

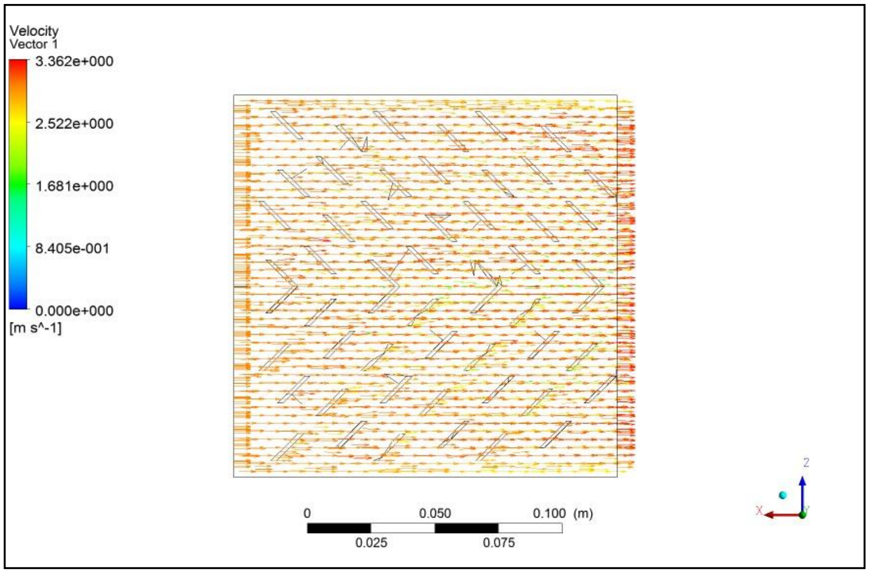

Fig. 39: 45 Staggered

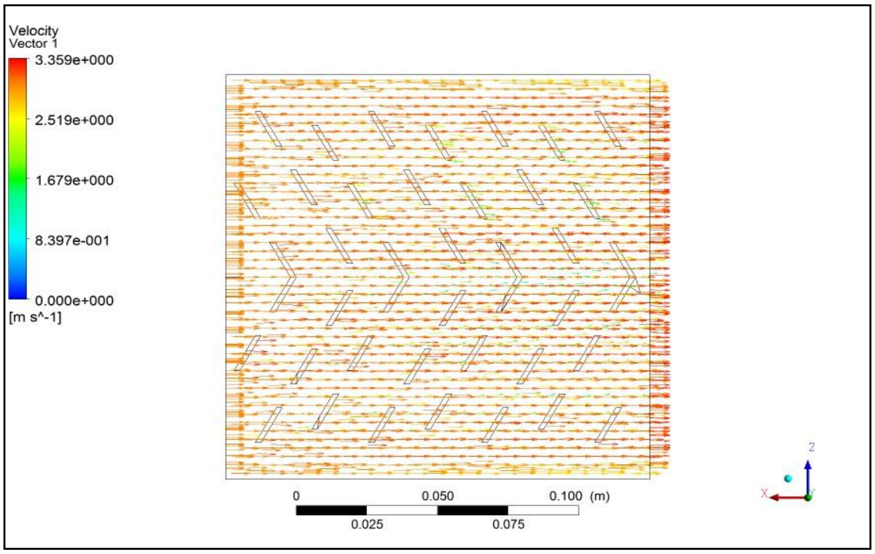

Fig. 40: 60 Staggered

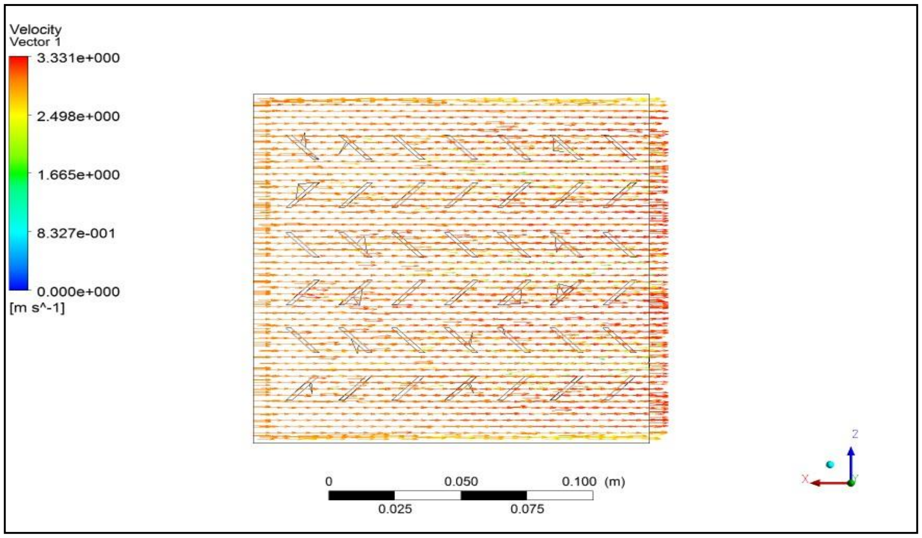

Fig. 41: Discrete V's

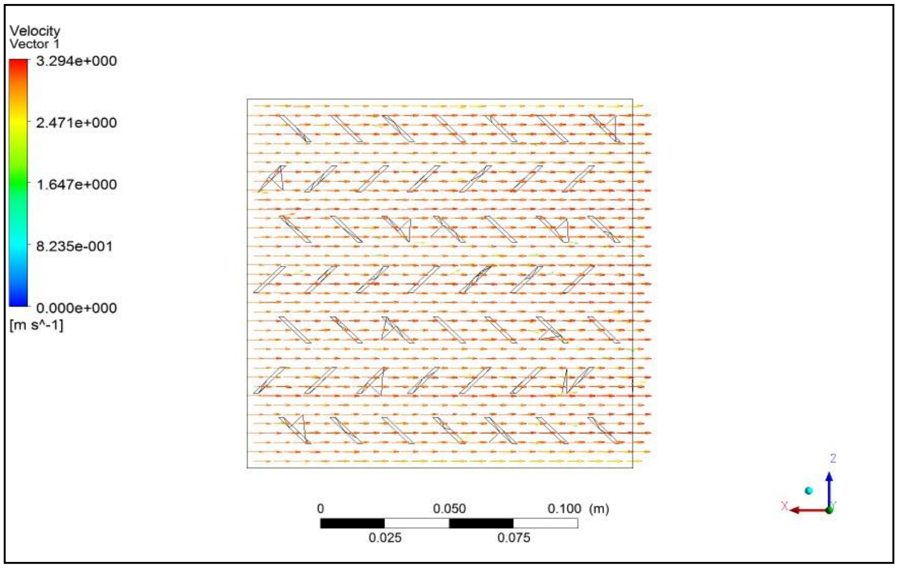

Fig. 42: Discrete V's 5mm down 10mm Distance

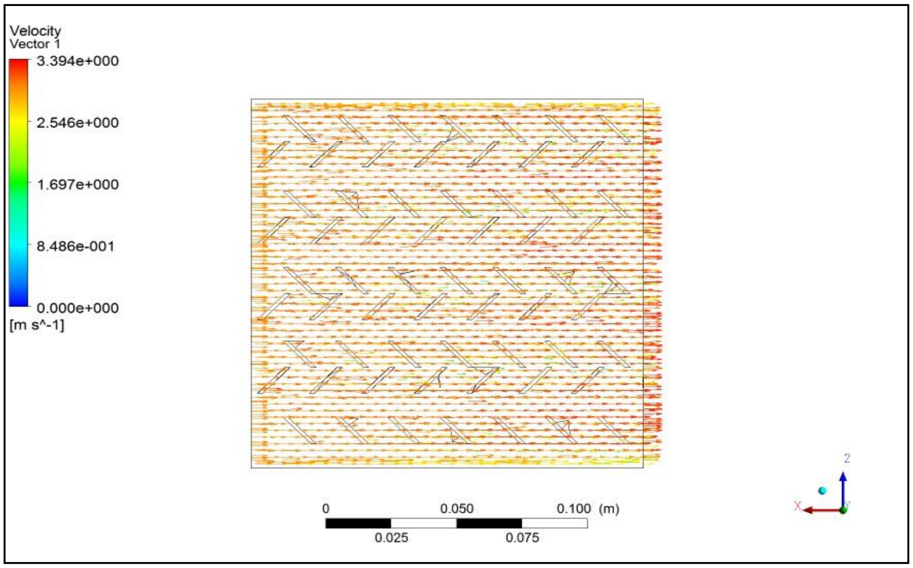

Fig. 43: Discrete V's 5mm down 0mm Distance

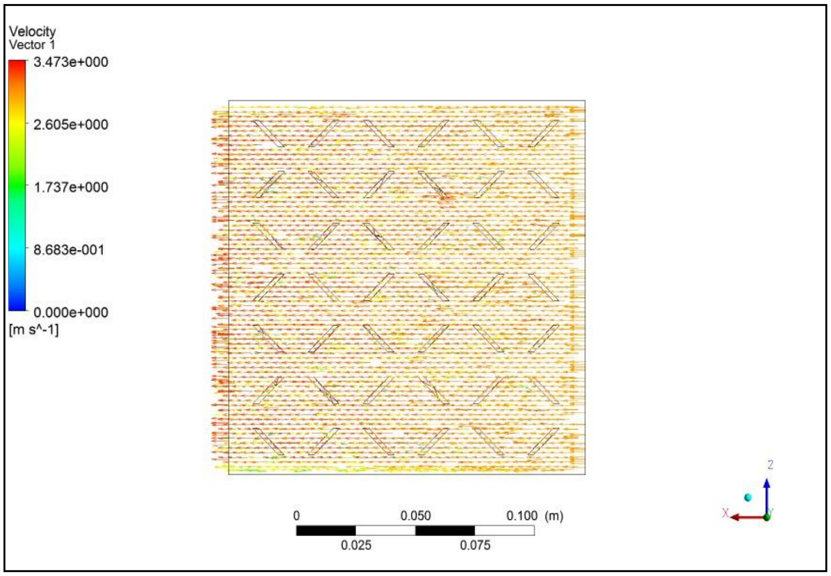

Fig. 44: Diamond

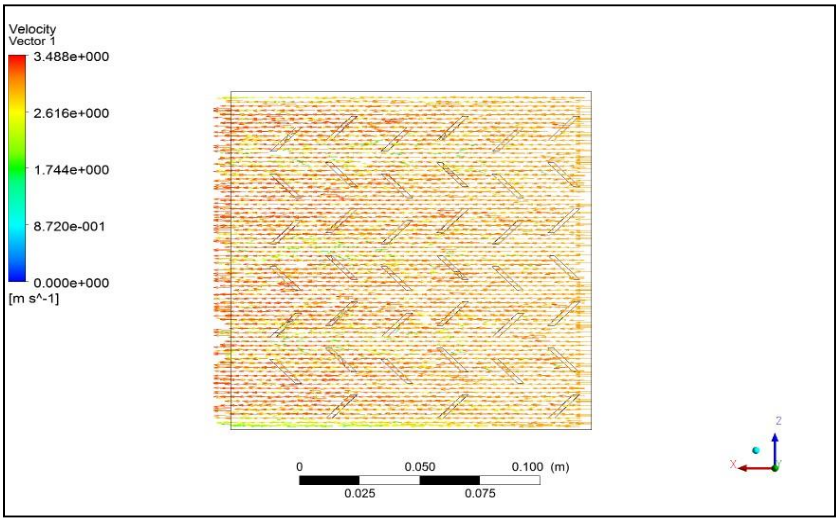

Fig. 45: Diamond Half Obstruction

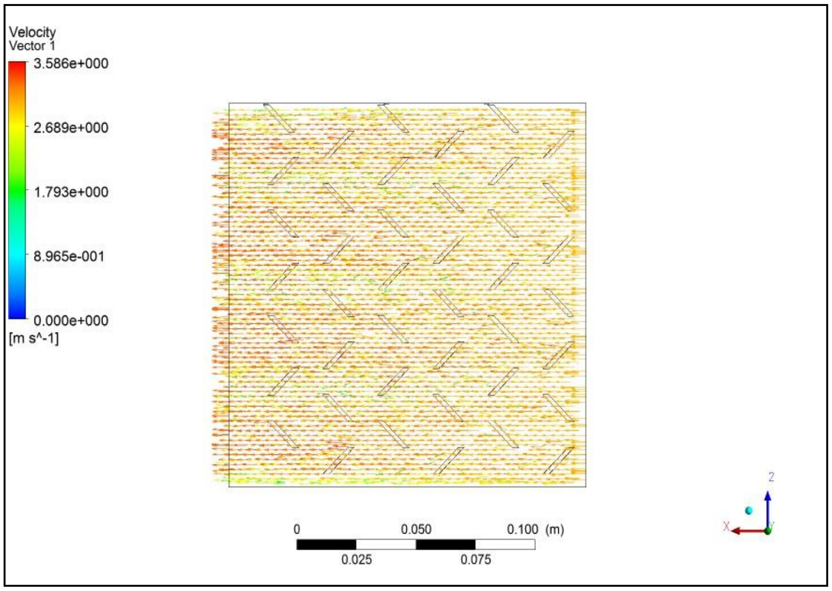

Fig. 46: Diamond Complete Obstruction

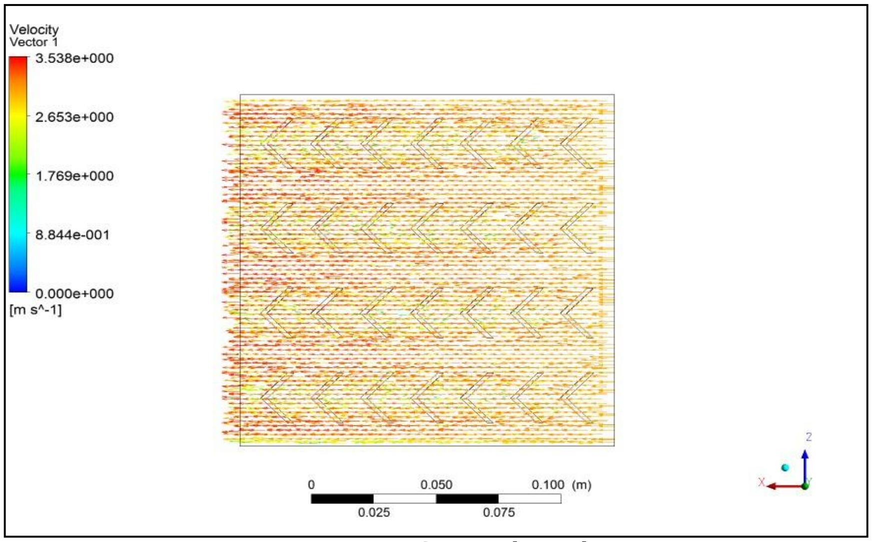

Fig. 47: Discrete Connected V's Inline

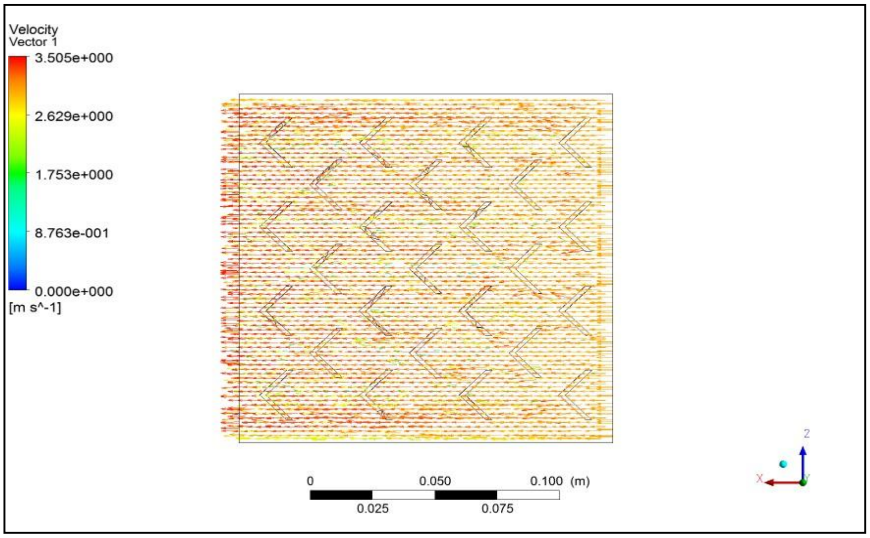

Fig. 48: Discrete Connected V's Staggered

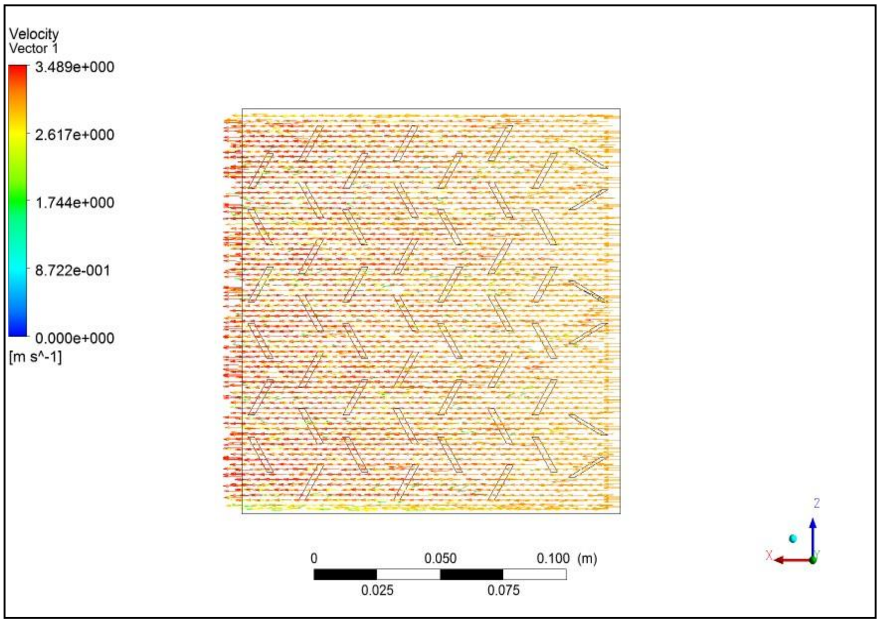

Fig. 49: 60-30

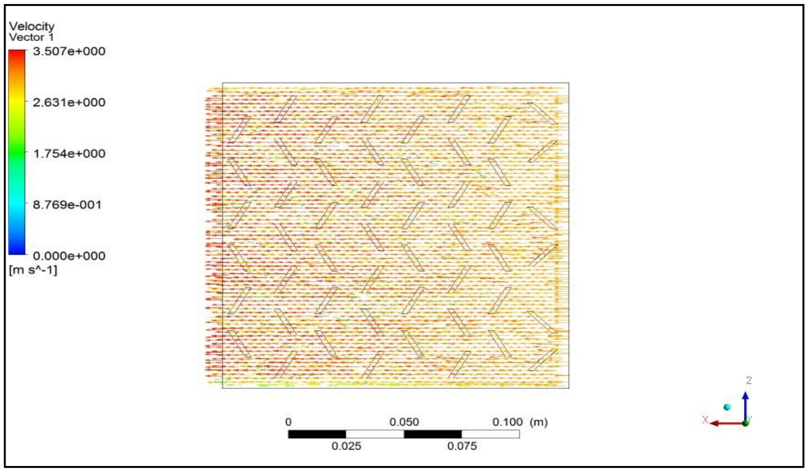

Fig. 50: 60-45

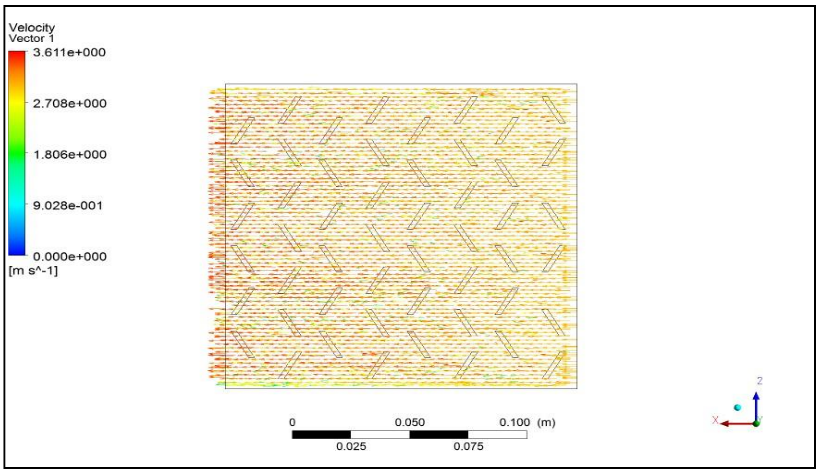

Fig. 51: 60-60

C. Nusslet No. (Nu) Comparison

Nusselt no.(Nu) provides way to compare the heat transfer characteristics of different arrangements for heat transfer devices.

In Fig. 52 Nusselt no. for all rib arrangements are presented. These Nusselt no. are calculated at constant Reynolds no.(Re). Here it can be observed that Nu for 45^0 inline rib arrangement is maximum as compared to other rib arrangements, while minimum value of Nu is observed for discrete V's arrangement. Then diamond like rib arrangement shows slight increase in value of Nu which is more with complete obstruction provided to flow. With different rib arrangement area of heat transfer changes as number of ribs changes which has significant effect on the nusselt no.(Nu). Ribs with angle combinations shows moderate Nusselt no. (Nu) as compared to all other rib arrangements. Thus it is observed from Fig. 52 that V-discrete ribs have better heat transfer characteristics than all other rib arrangements.

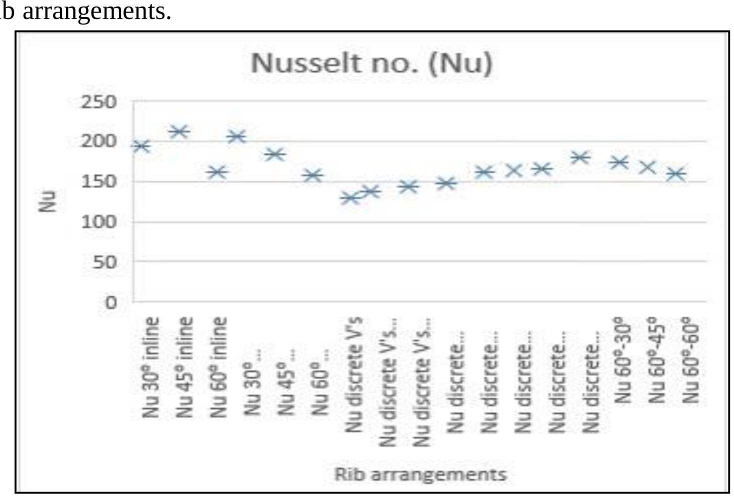

Fig. 52: Nusselt No. Comparison

D. Thermal Enhancement Factor (TEF)

Thermal Enhancement Factor (TEF) allows to compare the heat transfer i.e. thermal properties and the hydraulic properties simultaneously for heat transfer device. Fig. 53 shows thermal enhancement factor for all rib arrangements studied numerically. It is observed from fig. that ribs arranged inline with 45^0 and in staggered with 30^0 shows better Thermal Enhancement Factor (TEF) hence better heat transfer performance than other rib arrangements. Lowest heat transfer performance is observed with ribs arranged in discrete V's arrangement and 60-60 angle combination.

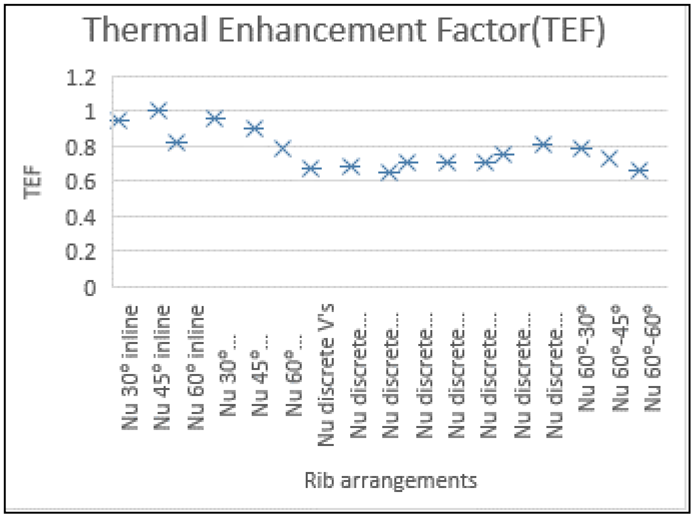

Fig. 53: Thermal Enhancement Factor

V. CONCLUSIONS

Numerical study has been carried out for different rib arrangements with discrete ribs. Results obtained from each arrangements are then compared with each other.

1) Ribs arranged inline 45 angle and in V-shape shows better heat transfer characteristics as compared to other rib arrangements.
2) Thermal Enhancement Factor (TEF) is maximum for 45 angled ribs arranged inline in V-shape which is slightly more than one and equal to 1.0011.
3) Heat transfer performance depends on rib angle and rib arrangement significantly.

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