A Study of Heat Transfer Enhancement using V Shaped Dimples on a Flat Plate with Experimentation & CFD

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Abstract

Heat transfer augmentation without considerable pressure drop is one of the most important issues now a days. The different active and passive methods are used for the same. One of the passive technique is use of dimples. A depression forming recesses on a flat surface is considered as a dimple. Variation in dimple geometry results in various heat transfer and friction characteristics. Introducing the dimples on the surface not only increases the surface area available for heat transfer but also reduce the hydrodynamic resistance for the fluid flow over the surface, resulting in less pressure drop. The present work is based on experimental investigation of the forced convection heat transfer over the V-shaped dimpled surface. The experiments were performed on 150mm × 150mm × 15mm aluminium plate with V- shaped dimples with pitch 3.2D, where D = diameter of dimple. V-dimples with δ/D = 0.2, 0.3, 0.5 are investigated, where δ= dimple depth. The experimentation was carried over a rectangular duct with aspect ratio 3:2. Heat transfer coefficient, Nusselt Number, friction factor and overall thermal performance of dimpled surface are compared with those from flat surface under same conditions. It is found that dimpled surface show 30- 45% enhancement over a flat plate. The maximum enhancement is observed in δ=0.3D in inline pattern. The dimpled surface with δ=0.5D shows minimum enhancement. Maximum frictional losses are observed in δ=0.5D dimpled surface and minimum in δ=0.2D dimpled surface. The overall thermal efficiency goes on increasing with increasing Reynolds number for all cases. The enhancement was verified using CFD analysis using ANSYSIS-15 ICEM software using k-ε turbulence model.

Keywords- Forced convection heat transfer, Nusselt Number, heat transfer enhancement, V-shaped dimples

I. INTRODUCTION

Heat exchangers have several engineering and industrial application. The important considerations in design of a heat exchanger are exact analysis of heat transfer rate and pressure drop estimations apart from issues such as long term performance and the economic aspect of the equipment. The major challenge in designing a heat exchanger is to make the equipment compact and achieve a high heat transfer rate using minimum pumping power. Techniques used for heat transfer augmentation depends on the engineering applications. The high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. In general, some kind of inserts are placed in the flow passage to augment the heat transfer rate which reduces the hydraulic diameter of the flow passage. Heat transfer enhancement in a tube flow by inserts such as twisted tapes, wire coils, ribs and dimples is mainly due to flow blockage, partitioning of the flow and secondary flow. Flow blockage increases the pressure drop and leads to increased viscous effects because of reduced flow area. Blockage also increases the flow velocity and in some situations leads to a significant secondary flow. Secondary flow further provides a better thermal contact between the surface and the fluid because secondary flow creates swirl and the resulting mixing of fluid improves the temperature gradient, which ultimately leads to a high heat transfer coefficient.

Use of dimples on the surface can significantly intensify the heat transfer enhancement. Dimples are arrays of indentations along surfaces. Different dimple geometries like triangular, spherical, tear drop shape have been employed to achieve heat transfer enhancement. These are attractive method for internal cooling because they produce multiple vortex pairs that augment local Nusselt number distributions as they advert downstream. Since they do not protrude into the flow they produce less form drag resulting in less pressure drop penalty. With this benefit, dimples offer advantages for cooling later turbine stages where lower pressure cooling air is employed. They are also advantageous because the pressure drop that they produce through an airfoil passage is relatively low, which allows favorable pressure margins to be maintained in other parts of the airfoil interior.

II. OBJECTIVE

Previous research work has been done for V-shaped dimples with $\frac{\delta}{D}$ = 0.3 in inline pattern. The present study is aimed to make a comparative study of V- shaped dimples $\frac{\delta}{D}$ = 0.2, 0.3 and 0.5 in inline pattern. The main objective of the experimental work is to

determine the heat transfer characteristics such as heat transfer coefficient, Nusselt number, friction factor, overall thermal efficiency of the V- shape dimpled surfaces and to compare it with flat surface. Experimental work will be compared with CFD analysis under same boundary conditions.
Test Matrix for experimentation is given below –

Table 1: Test Matrix

Sr. No.	Type of Square Plate	Variable Parameters*	
		Heat Input (W)	Air velocity(m/s)
1	Flat plate	10 20	2.6 3.2 4.2
2	V-shaped Dimpled plate (δ =0.2D inline)		
3	V-shaped Dimpled plate (δ =0.3D inline)		
4	V-shaped Dimpled plate (δ =0.5D inline)		

III. EXPERIMENTATION

Experimental set up is shown in below figure-

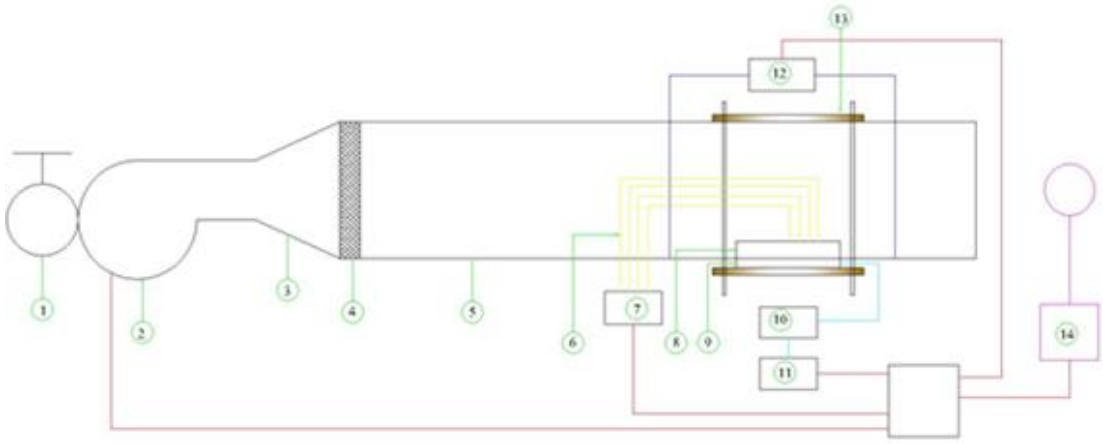

Fig. 1: Experimental Set Up

1) Flow Control valve
2) Blower Metallic Diffuser Section
3) Metallic Diffuser Section,
4) Honeycomb Flow Rectifier
5) Acrylic Duct
6) Thermocouple set
7) Universal Data Logger
8) Test plate
9) Plate Heater
10) Wattmeter
11) Dimmerstat
12) Pressure Differential Sensor
13) Heater Assembly
14) Flow Anemometer

In the present study of V-shaped Dimples following assumptions are made

1) The convection heat transfer coefficient ‘h’ is uniform and constant over the entire dimpled surface.
2) Heat transfer through the dimpled plate is at steady state, and there is no physical thermal energy source in the dimple.
3) The temperature of the fluid is uniform over the entire dimpled surface.
4) The thermal conductivity of the dimpled plate ‘K’ is uniform and constant.
5) Heat losses from radiation are neglected.
6) Complete contact between heater plate and test plate.
7) Constant air velocity from blower.
8) Losses through the fine gaps of acrylic duct are neglected.

Aluminium test specimens of industrial grades of size 150mm x150mm x 15mm are used having thermal conductivity 402 W/mK.

Fig. 2: Test Specimens

IV. RESULTS

In present work a study of heat transfer enhancement using V shaped dimples on aluminium plate placed in a rectangular channel with aspect ratio 3:2 was done. Three plates were arranged in inline pattern with δ = 0.2D, δ = 0.3D, δ = 0.5D configurations respectively. All results obtained were compared with flat plate without dimples. The experiments were performed for Reynolds number based on hydraulic diameter ranging from 19000 to 32000 for all five plates. All calculations and computations were carried out for the heat inputs at 10W and 20W. Steady state temperatures of the plates and pressure difference on the plates were measured. Heat transfer coefficients, average Nusselt number, friction losses and overall thermal efficiency were compared at different power inputs and different V shaped dimple configurations.

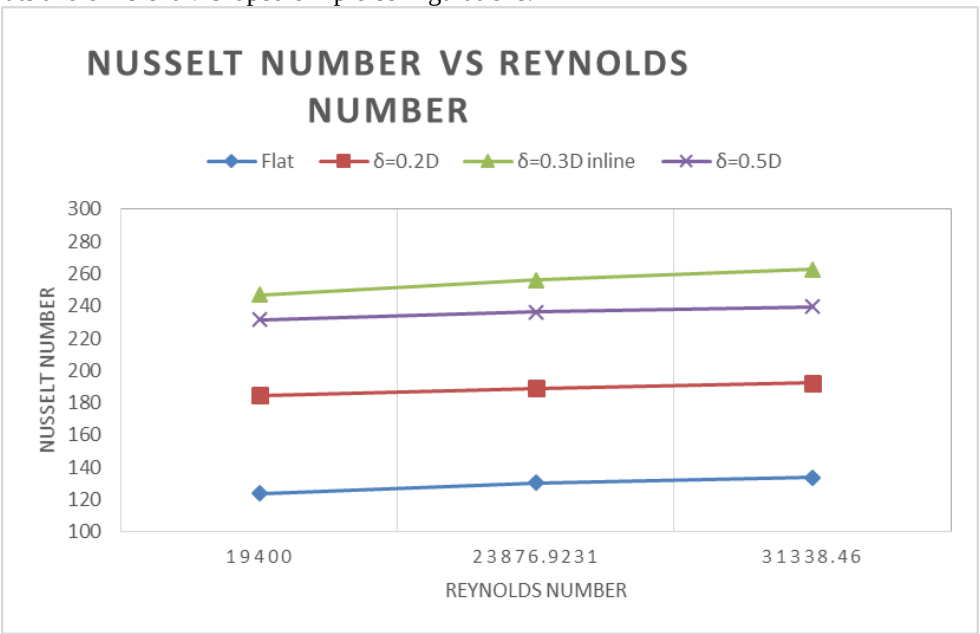

Fig. 3: Variation in Nusselt Number with Reynolds number at Q= 10 W

As observed from figure 3 Nusselt number goes on increasing with increasing Reynolds number, for different V shaped dimpled configurations .The heat transfer coefficients are greater in all three inline dimpled configurations (i.e. δ = 0.2D, δ = 0.3D, δ = 0.5D) as compared with the flat plate without dimples. The V- shaped dimple with δ = 0.3D shows maximum heat transfer rates at all Reynold's numbers, for δ = 0.5D it is intermediate and δ = 0.2D shows the minimum. Strongest secondary vortices are formed in case of δ = 0.3D, which leads to almost double heat transfer rates. In case of δ = 0.5D air pockets are thought to be formed in dimple cavities which hampers the heat transfer rate.

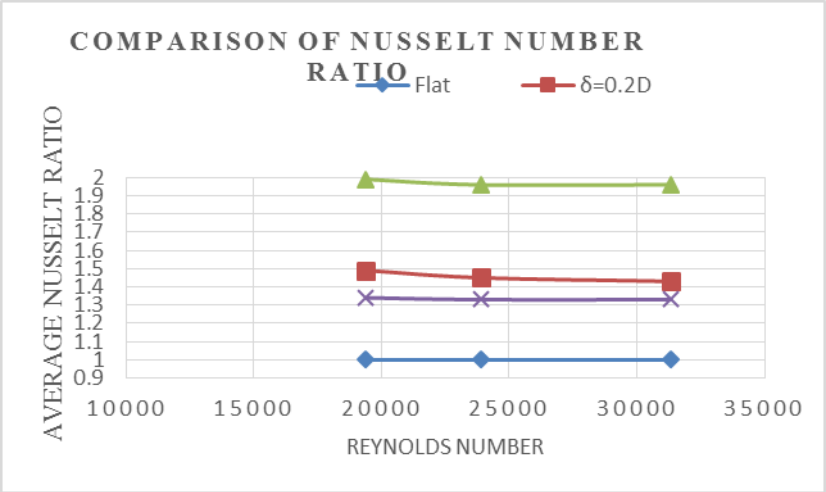

Fig. 4: Variation of Nusselt Number Ratio with Reynolds number at Q= 10 W

From figure 4 it is observed that Nusselt number ratio is almost twice for V-shaped with δ = 0.3D as compared with the flat plate. δ = 0.2D shows better Nusselt number ratio than δ = 0.5D but lower than that of δ = 0.3D.

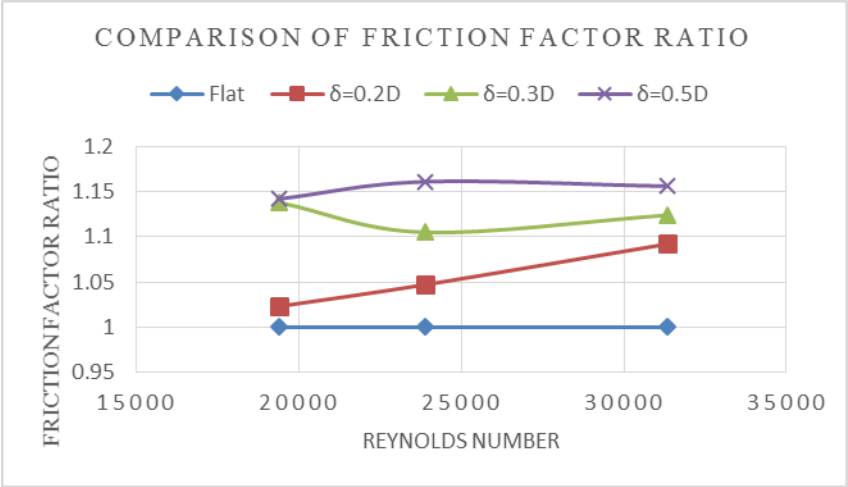

Fig. 5: Variation of Friction Factor Ratio with Reynolds number at Q= 10 W

As observed from figure 5 friction factor ratio goes on increasing with Reynolds number. Friction factor ratio is maximum for dimpled plate with δ = 0.5D at each Reynolds number and minimum for δ = 0.2D.

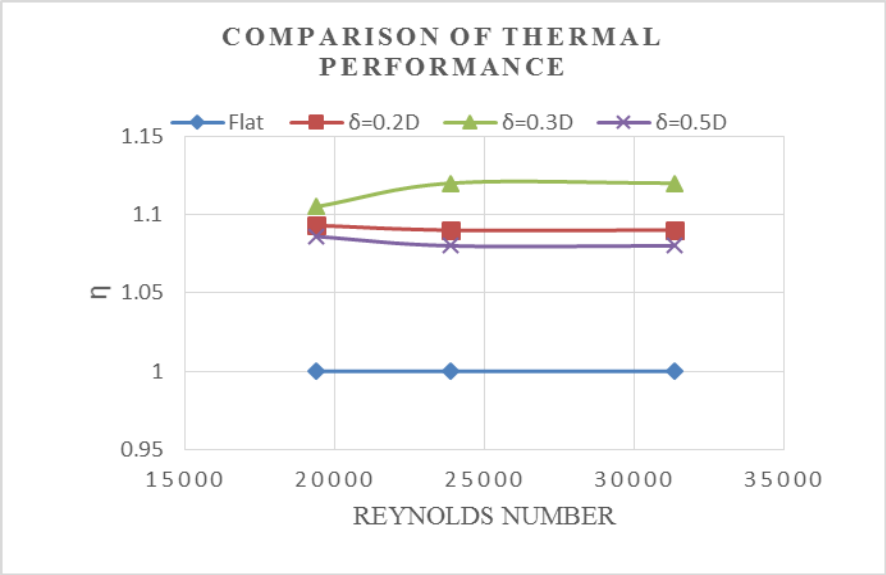

Fig. 6: Variation in Overall thermal Performance with Reynolds number at Q= 10 W

It was observed that with increase with Reynold's number both heat transfer coefficient and friction factor ratio are increasing but still it can be seen from figure 6, that the overall thermal efficiency either remains constant or increases slightly for a given arrangement. It is also observed that as the dimple depth increases from $\delta = 0.2D$ to $\delta = 0.3D$ the overall thermal performance is improved. But with further increase in dimple depth from $\delta = 0.3D$ to $\delta = 0.5D$ the overall thermal performance is observed to reduce.

V. CFD IMAGES & CONTOURS

The CFD analysis was carried out on ANSYS-15 ICEM software. The dimple geometries employed are exactly the same as those used in the experiments. A k-ε turbulence model is employed to investigate the three –dimensional turbulent flow field and heat transfer on a dimpled square plate in a rectangular channel. The turbulent flow structures and heat transfer characteristics are studied for four different V-shaped dimpled arrangements and a flat plate for Reynolds number ranging from 19000 to 33000. The grids for all test plates are generated using the software package ICEM.A structured grid with hexahedral mesh is employed. The grids near the dimples are very dense whereas grid density is relatively sparse on remaining surface.In the present study the inlet section and outlet section are assumed adiabatic. The constant surface heat flux corresponding to 10 watt and 20 watt are applied respectively. No – slip velocity conditions are applied at all channel surfaces. Uniform velocity and uniform temperature are used at the channel inlet. The inlet turbulence intensity level is 5 %. The fluid in this study is incompressible, dry air with constant thermal – physical properties.

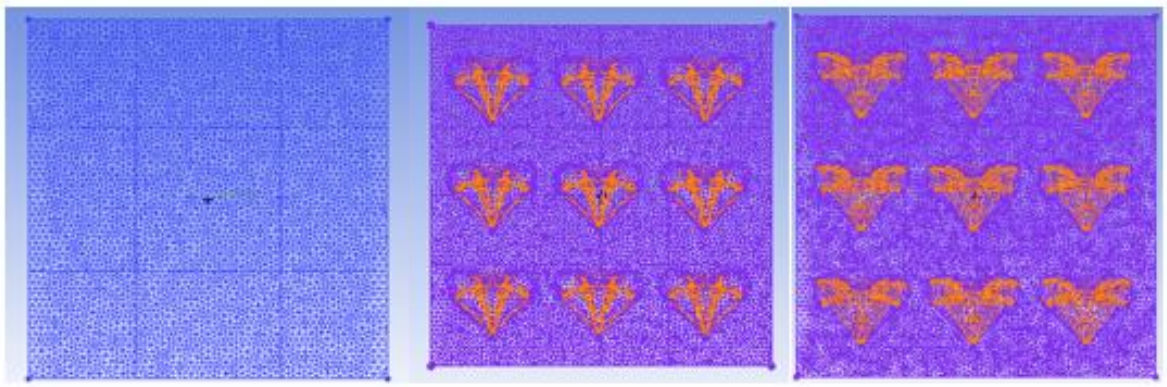

Fig. 7: Meshing

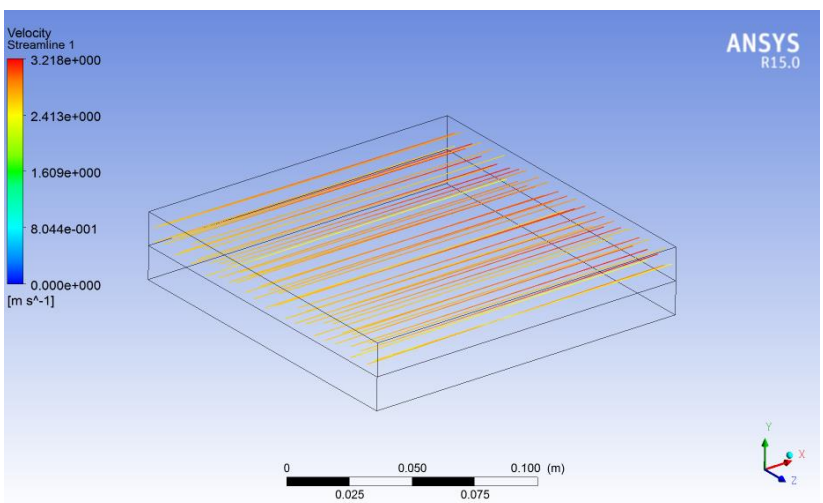

Fig. 8: Velocity streamlines on surface of Flat plate at Re=19400, Q= 10W

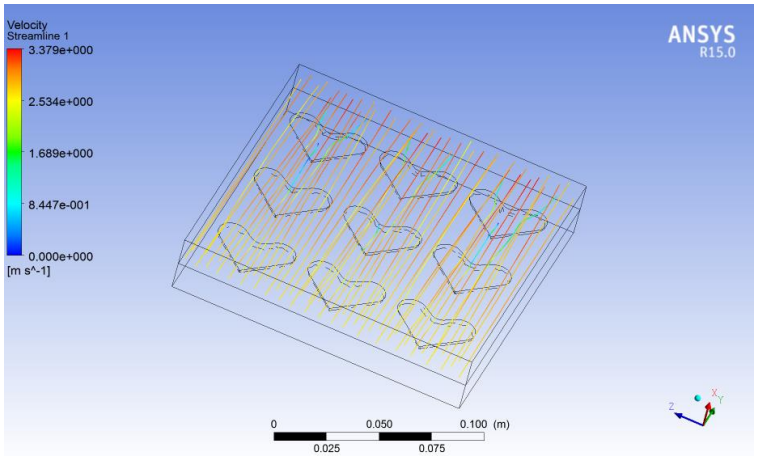

Fig. 9: Velocity streamlines on surface of V- Shape dimpled plate with δ=0.2D at Re = 19400, Q= 10W

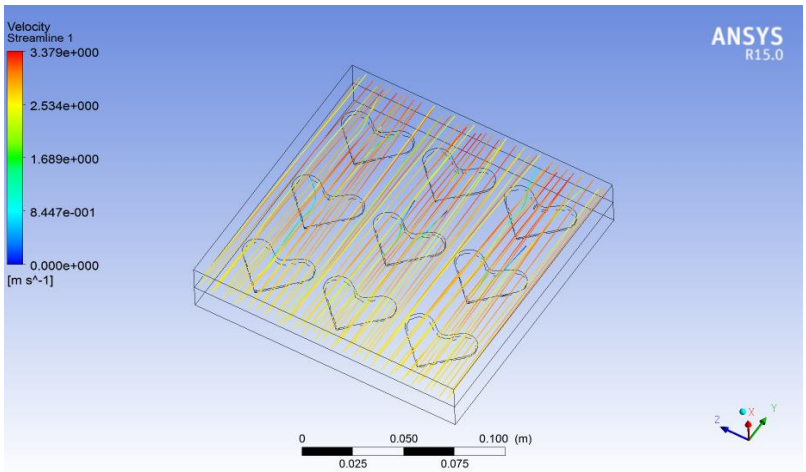

Fig. 10: Velocity streamlines on surface of V- Shape dimpled plate with δ=0.3D at Re = 19400, Q= 10W

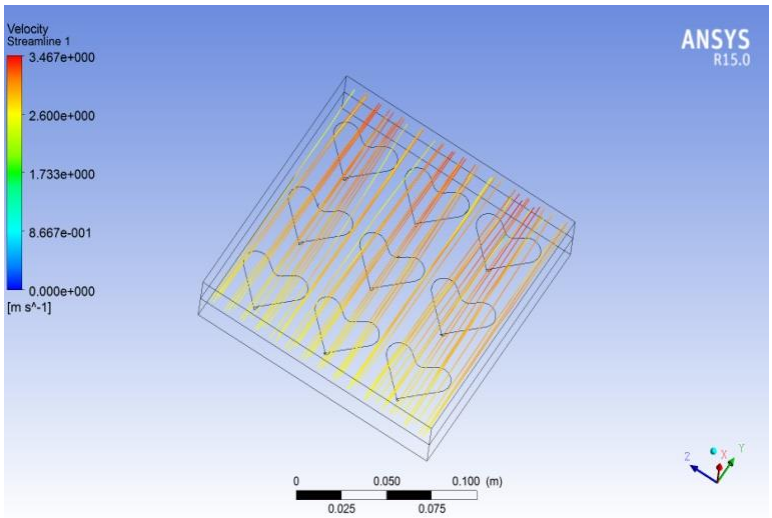

Fig. 11: Velocity streamlines on surface of V- Shape dimpled plate with δ=0.5D at Re = 19400, Q= 10W

From the velocity streamline images fig. 8 to 12, it is observed that the velocity streamlines run parallel to each other in case of flat surface. It can be observed that these lines deviate in case of V-shaped dimpled surface. Strong secondary flow can be observed in case of δ= 0.3D V-shaped dimpled surface. The heat transfer enhancement is observed not only in the dimple cavity but also in the region in vicinity of it. Heat is carried away by the scrubbing action of the secondary vortices inside the cavity. This secondary flow also breaks the thermal boundary layer in the adjacent region causing enhancement outside the dimples.

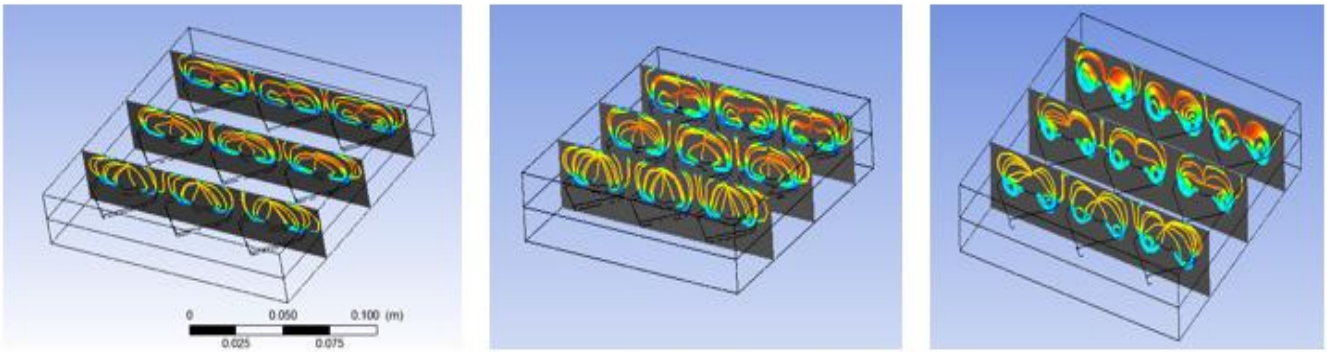

Fig. 12: Velocity streamlines along three vertical planes on surface of V-shaped dimpled plates δ=0.2D, 0.3D, 0.5D

VI. CONCLUSION

Experimental investigation of V-shaped dimples on a flat plate is carried out and heat transfer characteristics were studied for each case with different variable parameters with respect to input power and velocity.
The important findings of the experimental investigations are as follows-

1) With 4 types of arrangements such as flat , inline (δ = 0.2D, δ = 0.3D, δ = 0.5D) and with variable parameters such as input power and velocity, total 24 cases are studied and it has been found that the inline arrangement with δ = 0.3D gives optimum solution as compared to other 4 arrangements.
2) It is observed from the experimental results that the convective heat transfer coefficient is enhanced with increase in air velocity. Thus the primary advantage offered by the V-shaped dimples is that there is increase in heat transfer enhancement with increasing Reynolds number. As the Reynolds number rises, the heat transfer enhancement rises due to counter rotating vortices formed within the legs of the dimples.
3) When Reynolds number was varied from 19000 to 35000, it was observed that there is maximum increase in heat transfer coefficient in dimpled plate with δ=0.3D inline pattern, δ= 0.2Dinline and δ= 0.5D inline pattern.
4) The maximum friction factor ratio was observed for δ= 0.5D pattern and minimum for δ= 0.2D pattern. Based on constant pumping power conditions, the maximum overall thermal performance increases from δ= 0.2D to δ= 0.3D but reduces from δ= 0.3D to δ= 0.5D.
5) The CFD analysis shows that secondary vortices are created in the legs of V dimples which leads to heat transfer enhancement. The fluid exit temperature and heat transfer coefficient obtained from CFD results are in agreement with the experimental results.

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