International Journal of Engineering Research and Reviews ISSN 2348-697X (Online)
Vol. 5, Issue 2, pp: (36-44), Month: April - June 2017, Available at: www.researchpublish.com

Experimental and Numerical Investigation of Heat Transfer Enhancement through Inline and Offset Elliptical Dimples for Trapezoidal Channel

Mangesh Pohare[1], Dr. Laxman Kamble[2]

[1]ME Mechanical Heat Power Engineering, Dr. D Y Patil School of Engineering Academy, Ambi, Talegaon Dabhade, Pune, Maharashtra, India

[2]Department of Mechanical Engineering, Dr. D Y Patil School of Engineering Academy, Ambi, Talegaon Dabhade, Pune, Maharashtra, India

Abstract: **An exploratory examination has been completed to study heat transfer by dimpled surface. Reynolds number in view of hydraulic diameter and velocity is differed from 800 to 4000. The proportion of dimple depth to dimple diameter is 0.25.The proportion of channel height to diameter is 0.625. The heat transfer information acquired is contrasted and the information gotten from smooth plate under comparative geometric and stream conditions. It is watched that at all Reynolds number as velocity builds, the standardized Nusselt number and thermal performance increments.**

Keywords: **Dimple, heat transfer, Velocity, Nusselt Number, Reynolds Number, thermal.**

I. INTRODUCTION

Heat transfer inside stream entries can be upgraded by utilizing inactive surface alterations, for example, dimples. These heat transfer enhancement methods have practical application for inner cooling of turbine airfoils, ignition chamber liners and electronic cooling devices, biomedical devices and heat exchangers. As of late, dimples have drawn more consideration in view of the huge enhancement in heat transfer with a lower punishment in the pressure drop. Utilizing dimpled surfaces in these circumstances requires learning of the impacts of various dimple geometry qualities; in any case, at present, the authentic writing is insufficient in giving such information to numerous critical geometric parameters. Various heat transfer examines from Russia use dimples. These reviews utilize streams over level dividers with customary varieties of circular pits [2], streams in annular entries with a stunned exhibit of curved dimples on the inside tube shaped surface [3], streams in single hemispherical depressions [4, 5], streams in diffuser and focalized channels each with a solitary hemispherical cavity [6], and streams in a tight channel with roundly formed dimples put in relative positions on two inverse surfaces [7]. Heat transfer increases as high as 150 percent, contrasted with smooth surfaces are accounted for at times with obvious weight misfortunes [3]. Other late information demonstrate that the improvement of the general warmth exchange rate is around 2.5 times cover surface values up a scope of Reynolds numbers and weight misfortunes are about a large portion of the qualities delivered by traditional rib turbulators. Moon et al. give information that demonstrate that enhancements in warmth exchange strengthening and weight misfortunes stay at roughly steady levels for various Reynolds numbers and channel statures. Mahmood et al. [8] depict the components in charge of neighborhood and spatially arrived at the midpoint of warmth exchange growthes on level channel surfaces with a variety of dimples on one divider for one channel stature, equivalent to half of the dimple print distance across. Different examinations consider stream and warmth move in single circular cavities [9]; impacts of dimples and projections on inverse channel dividers [4, 5]; the impacts of dimple profundity on vortex structure and surface warmth exchange [11]; the impacts of profound dimples on neighborhood surface Nusselt number appropriations [2]; the consolidated impacts of perspective proportion, temperature proportion, Reynolds number and stream structure [3]; and the stream structure because of dimple despondencies on a channel surface [10]. The present review is not quite the same as existing examinations on the grounds that the impacts of dimple profundity δδ/D values extending from 0.2 to 0.4 are researched. The exploratory outcomes are given for a proportion of channel stature to dimple print distance across 0.5 and Reynolds number in view of pressure driven width 10,000 to 40,000. The information displayed in this paper is normal Nusselt number and erosion variables.

II. EXPERIMENTAL INVESTIGATIONS

A. Experimental Set Up:

An exploratory set-up has been outlined and created to concentrate the impact of dimpled surface on heat transfer and fluid flow qualities in trapezoidal duct. A schematic diagram of the exploratory set-up is shown in Figure1. The test assembly is an outside stream circle that comprises of a variable velocity blower, stream control valve, settling chamber, the test segment, flat plate heater, control panel, thermocouples and at exit section. The cross area of channel at inlet is 13mm x 13mm and at outlet 23mm x 23mm.The length of the duct is 500mm. The test section is created from Aluminum sheet. The Test area is of length 500 mm. To give uniform heat flux, flat plate heater of size of width at inlet 13 mm and at outlet 23mm.The length of the heater is 500mm and thickness is 2mm. The heater is situated below the test plate. The velocity of air is measured by an Anemometer. For temperature estimation aligned thermocouples are utilized. The pressure drop over the test segment is measured by a manometer tube, filled with water.

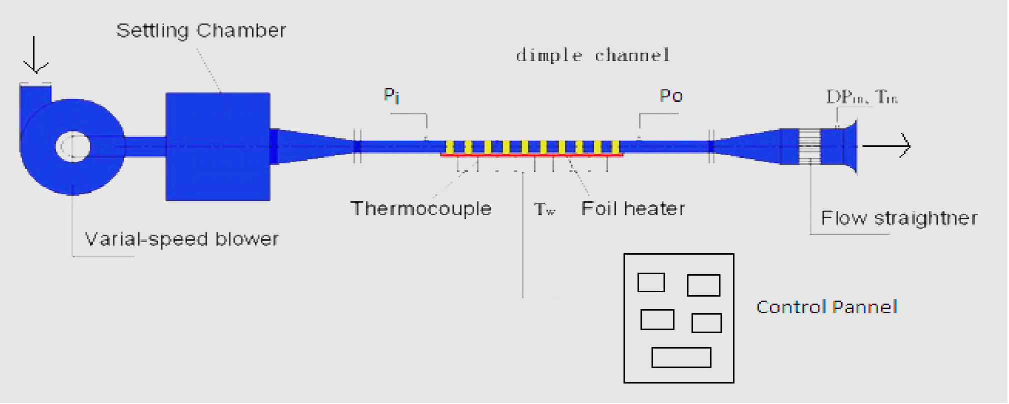

Fig.1 Experimental Setup Diagram

B. Experimental Procedure:

The test plates are assembled in duct one by one and checked for air leakage. The blower was switched on to let a predetermined rate of airflow through the duct. The velocity of air is measured by using Anemometer. A constant heat flux is applied to the heater. Seven thermocouples are attached to measure the inlet, outlet and surface temperature of test plate at different points. Seven values of velocity in the range from 1 m/s to 4 m/s were used for each set at same or fixed uniform heat flux. At each value of velocity and the corresponding heat flux, system was allowed to attain a steady state before the temperature data were recorded. The pressure drops were measured by using manometer tube when steady state is reached.

During experimentation the following parameters were measured

1) Pressure difference across the manometer tube.

2) Temperature of the heated surface of test plate and temperatures of air at inlet and outlet from the duct

C. Geometric Parameters:

a) Test Plates:

The material for test channel is Aluminum. For experimental study three plates are manufactured, one plate is kept smooth for comparison and on two plates dimples are produced with inline and offset configurations.

Table 1 Test Plate Dimensions

Sr. No.	Parameter	Dimensions
1	Length of test channel (L)	500mm
2	Width at inlet section (W_1)	13mm
3	Width at outlet section (W_2)	23mm
4	Thickness of test channel (t)	5mm
5	Hydraulic diameter (D_h)	15mm

The test channel with dimensions is shown in fig. 2 below

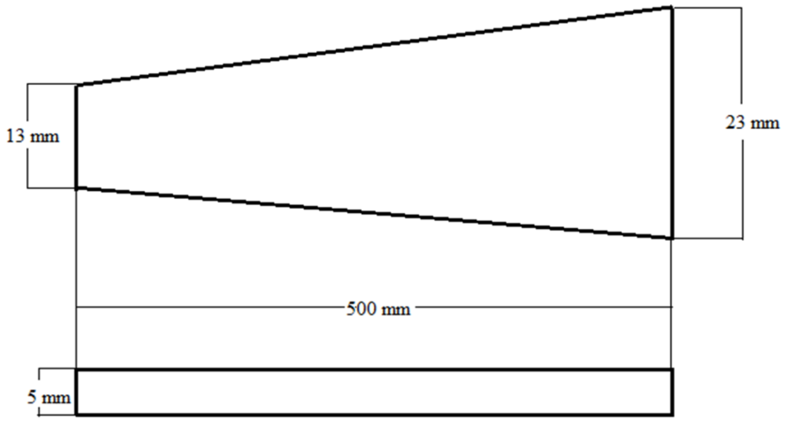

Fig.2 (a) Smooth Plate

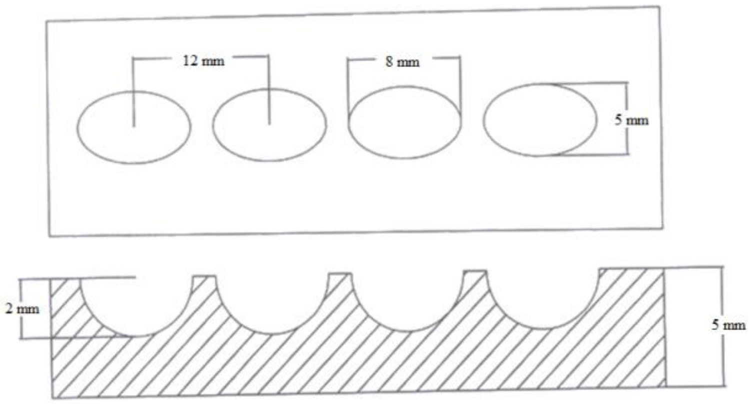

Fig.2 (b) Inline Dimple Plate Major Axis along Length

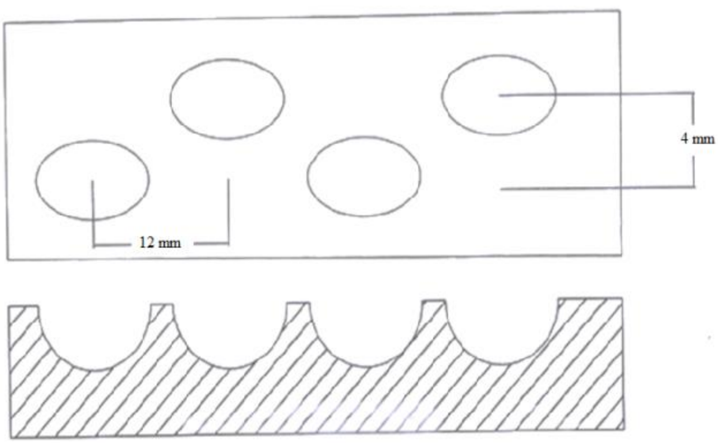

Fig.2(c) Offset Dimple Plate Major Axis along Length

b) Elliptical Dimple:

Table 2 Elliptical Dimple Dimensions

Sr. No.	Parameter	Dimensions
1	Major Diameter (D_{ma})	8mm
2	Miner Diameter (D_{mi})	5mm
3	Height (Depth) of the Dimple (H)	2mm
4	Horizontal Central Distance between two dimples (X)	12mm
5	Vertical Central Distance between two dimples for offset configurations	4mm

The elliptical dimple with dimensions is shown in fig.3 below

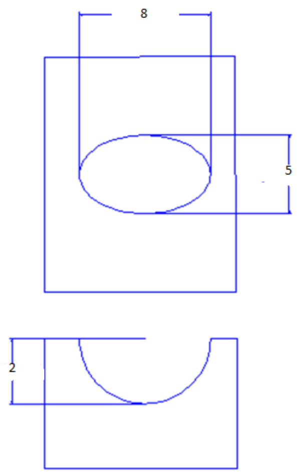

Fig.3 Elliptical Dimple Dimensions

D. Flow Parameters:

Steady state value of the plate and air temperatures in the channel, at various locations for a given heat flux and mass flow rate of air, is used to determine the values of performance parameters.

a) Reynolds Number:

The Reynolds number associated with the flow can be calculated by using following mathematical equation.

$$Re = \frac{\rho\, V\, Dh}{\mu} \text{ ………………………. } (1)$$

Where,

ρ is the density of the fluid in Kg/m^3

V is the velocity of the fluid in m/s

D_h is hydraulic diameter in m

μ is the viscosity of the fluid in Ns/ m

b) Hydraulic Diameter

The hydraulic diameter D_h is calculated by using mathematical relation.

$$Dh1 = \frac{2(W1\ X\ h1)}{(w1+h1)} \text{ …………….. } (2)$$

and

$$Dh2 = \frac{2(W2\ X\ h2)}{(w2+h2)} \text{ …………….. } (3)$$

where,

D_{h1} is hydraulic diameter at inlet section in m.

D_{h2} is hydraulic diameter at outlet section in m.

W_1 is width of test section at inlet in m.

W_2 is width of test section at outlet in m.

h_1 is the height of channel at inlet in m.

h_2 is the height of channel at outlet in m.

The average hydraulic diameter is then calculated as

$Dhavg. = \frac{Dh1+Dh2}{2}$ ………..... (4)

c) Mass Flow Rate:

The mass flow rate of air is determined from velocity and cross sectional area, using a following relation.

$$M = V\ x\ Ac\ x\ \rho \dots \dots \dots \dots . \dots\ (5)$$

Where,

M is flow rate,

V is velocity,

Ac is cross-sectional area.

P is density of air

d) Heat Gain (Q)

The useful heat gain of the air is calculated as

$$Q = M\ Cp\ (To - Ti)\ W \dots \dots ...(6)$$

Where,

Ti= Fluid temperature at the inlet of the duct (°C)

To = Fluid temperature at the exit of the duct (°C)

M= Mass flow rate of air

Cp = Specific heat of air

Q = Convective heat transfer to air

e). Heat Transfer Coefficient (h)

The heat transfer coefficient for the test section is calculated as

$$h = \frac{Q}{As}\ (Ts - Tf)\ \frac{W}{sqm}K \dots\ (7)$$

Where,

Ts is the average temperature of the test surface

Tf is the average temperature of air in the duct

Tf= To + Ti / 2

As is projected surface area of test surface

h is convective heat transfer coefficient

f) Nusselt Number (Nu):

The Nusselt number is calculated as

$$Nu = \frac{(h\ x\ Dhavg)}{k} \dots\dots\dots\dots (8)$$

Where,

Nu is the average Nusselt number of the test surface

Dhavg is the average hydraulic diameter of the trapezoidal duct

k is the thermal conductivity of air

III. RESULTS AND DISCUSSION

The thermo-physical properties of air used in the calculation of heat transfer and friction parameter were taken from available standard data tables which corresponding to mean bulk air temperature. The effect of humidity has been neglected since the relative humidity values during experimentation were found to be low.

Using the data obtained from experiments, the heat transfer and the thermal performance characteristics are discussed in the following subsections.

a) Effect of Reynolds Number:

Nusselt number proportions are measured with dimple on one channel surface and warming on one channel surface, for various velocitys, shifting from 1 to 4 m/s. The Benchmark Nusselt numbers, utilized for standardization of the qualities introduced, are acquired utilizing a similar warm limit conditions and warming game plan as when dimples are utilized on the estimation surfaces. Moreover, heat transfer coefficients and warmth flux values (used to decide Nusselt numbers) depend on level anticipated ranges in both cases. It is watched that Nusselt number increments with Reynolds number for dimpled surface and in addition for smooth channel, yet rate of increment is more for the dimpled surface when contrasted with smooth surface. The most extreme estimation of Nusselt number watched for velocity 4 m/s at Reynolds number 3500.

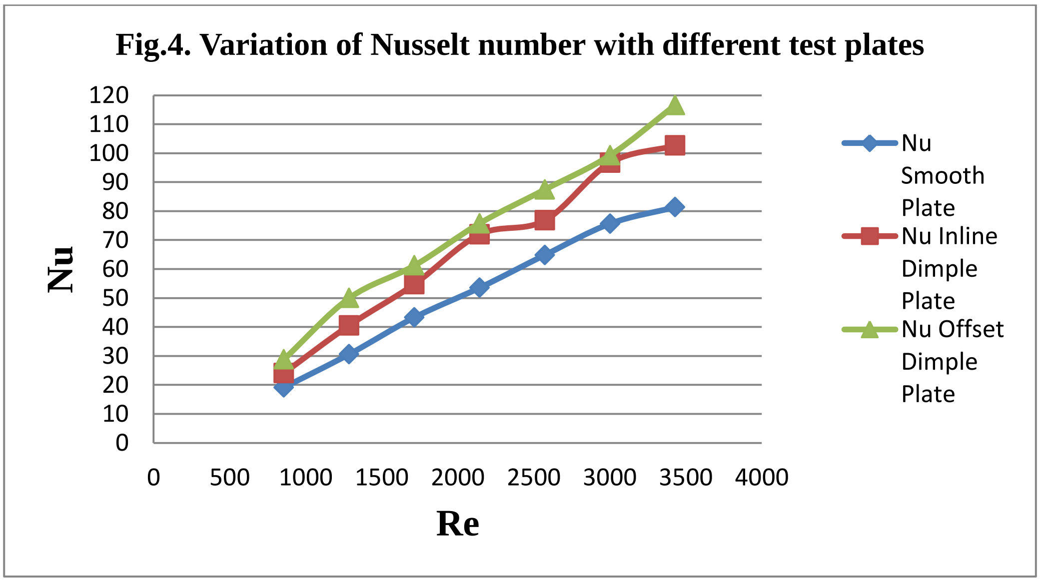

Fig.4

b) Effect of Velocity on Thermal Performance

The heat transfer coefficients are resolved for velocity run from 1 m/s to 4 m/s for all the three test plates considered. It has been watched that the warmth exchange coefficient and Nusselt number are expanded as velocity increments. It is additionally watched that the warmth exchange coefficient and Nusselt number are expanded for inline dimple plate and balance dimple plate contrasted and smooth plate.

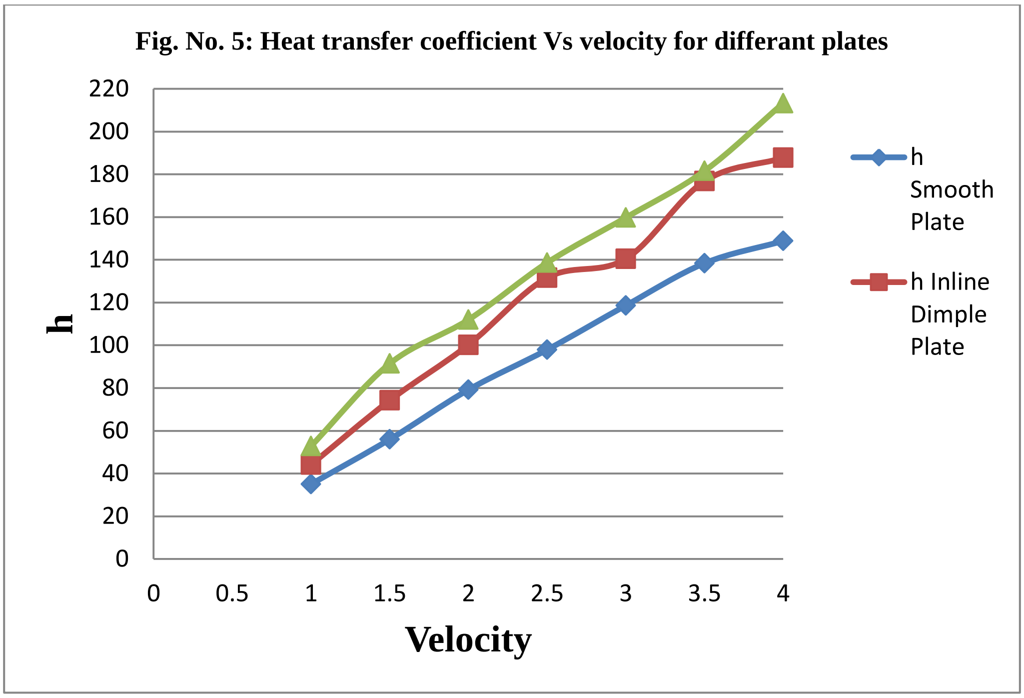

Fig.5

Numerical Setup:

In order to obtain more physical understandings about the flow and heat transfer in the dimple plates, additional fully three-dimensional and steady-state conjugate numerical computations were done. For the computation, the model includes the heat transfer in the solid wall and in the fluid, and the boundary conditions were made identical to the experiments. The channels are heated uniformly from the bottom of the test plate, whereas the top wall of the channel is insulated. The channel and dimple surfaces are treated as no-slip boundaries. A schematic of the for the numerical computation for the smooth plate , inline dimple plate and offset dimple plate are shown in Fig. 6.

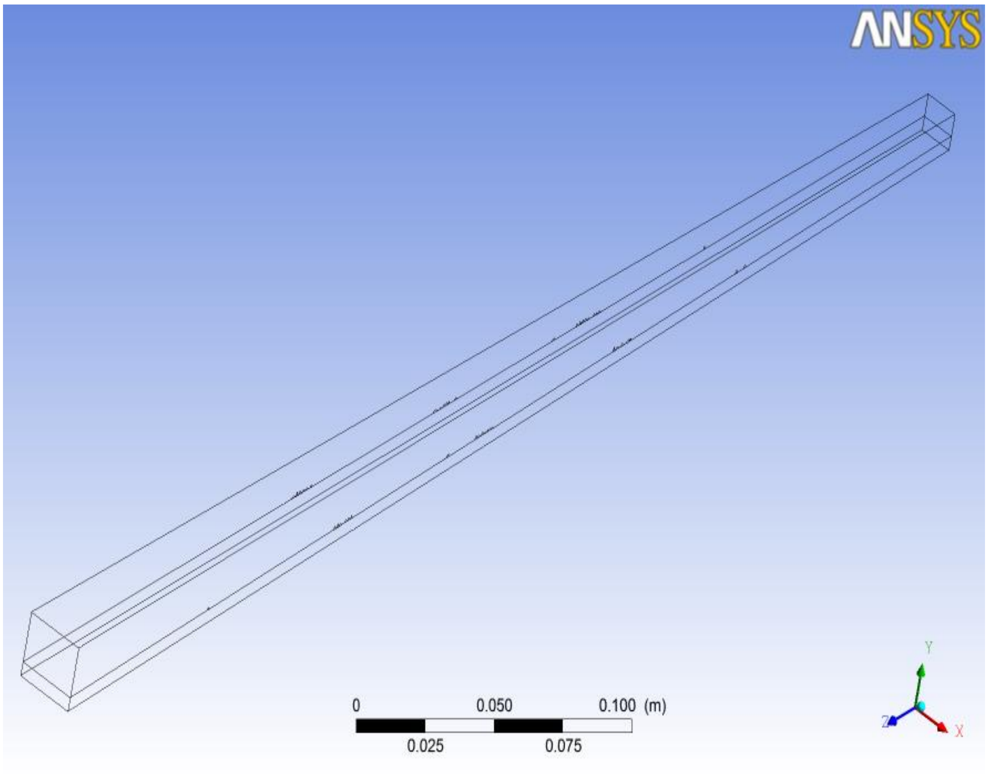

Fig. 6 (a) Smooth Plate

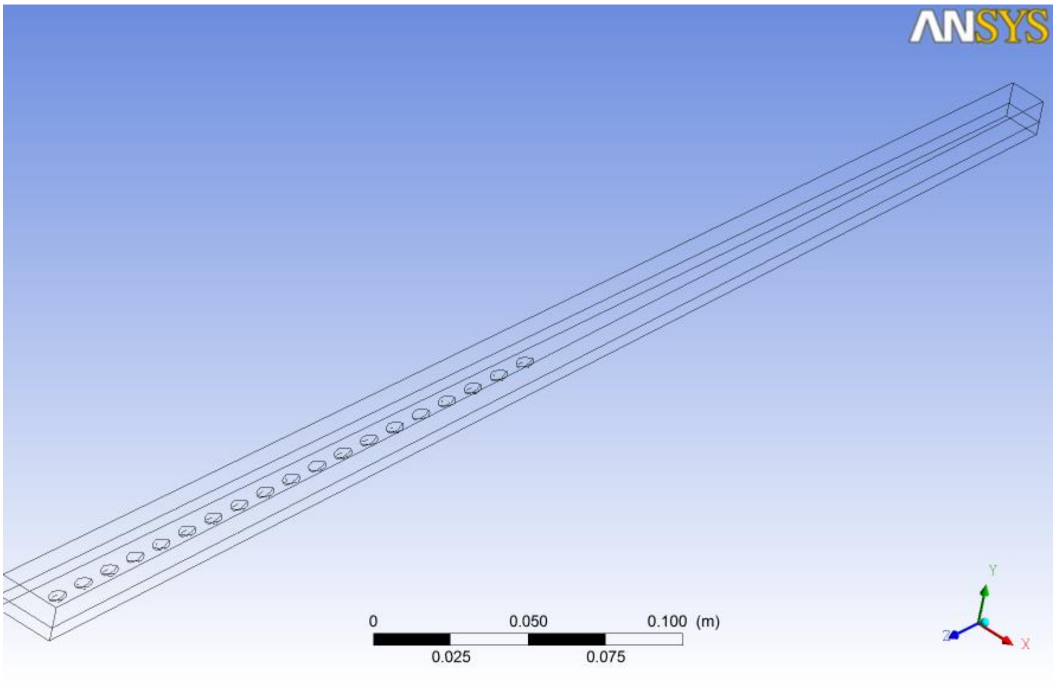

Fig. 6 (b) Inline Dimple Plate

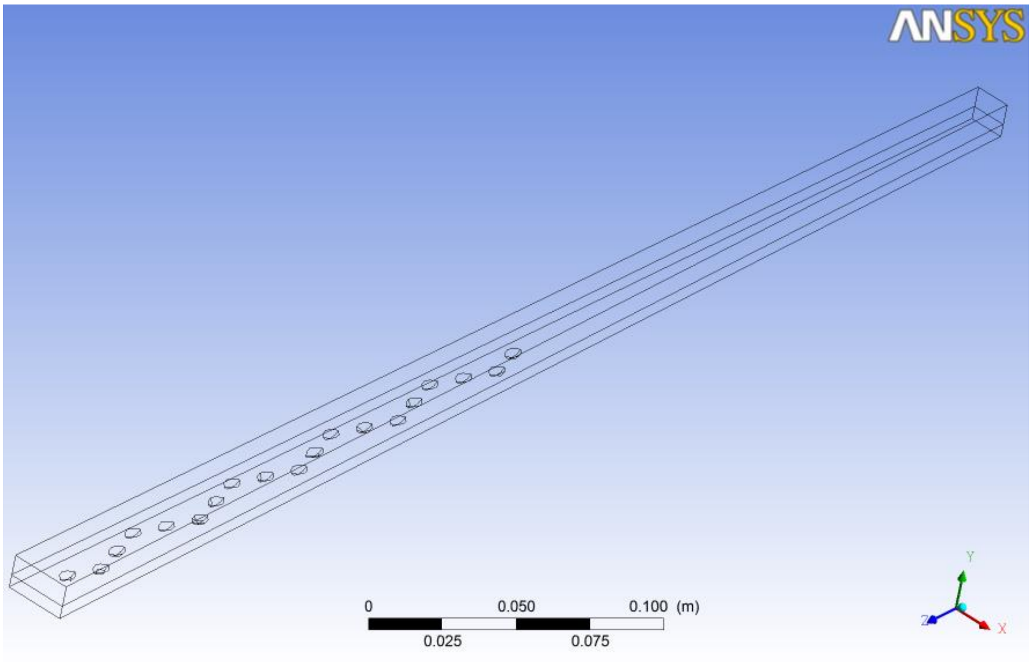

Fig. 6 (c) Offset Dimple Plate

The computations were performed using the commercial solver FLUENT V6 with the realizable k -ε turbulence model. The realizable k -ε turbulence model is believed to have improved predictive capability for complex turbulent flows with flow swirling and separation compared to the standard k -ε model, and was also used to simulate the flow and heat transfer in the channel with dimples. For all computation cases, an unstructured hybrid mesh was generated using the commercial software ANSYS ICEM.

The analysis done for seven internment values of velocity in the range 1 m/s to 4 m/s. The properties of air taken at 320K are ρ = 1.103 Kg/m^3, C_p = 1008 J/KgK, μ = 0.00001935 Kg/ms and k = 0.02753 W/mK. The boundary conditions are inlet temperature T_i =310K the average surface temperature are taken as obtained from experimental data.

For all the computations, ideal gas air was used as the working fluid with a linear temperature-dependent thermal conductivity and viscosity being specified. Throughout the study the fluid is considered to be incompressible and hence the flow field and the energy equation were uncoupled. To reduce the numerical errors, a second order volume discretization scheme was used. The minimum convergence criterion for the continuity equation, velocity and turbulence quantities is 10^{-4} and 10^{-7} for the energy equation. Careful grid independence check has been done for the computations by considering several grid systems with nodes.

IV. CONCLUSIONS

In the paper, a comparative experimental and numerical study was conducted on the heat transfer of smooth plate, inline dimple plate and offset dimple plate. The average Nusselt number and the thermal performance parameters of the smooth plate, inline dimple plate and offset dimple plate have been obtained and compared with the experimental data. The comparisons shows that, there is average 30 % and 45 % increase in heat transfer coefficient for inline dimple plate and offset dimple plate to that of smooth plate. The comparisons also shows that, there is average 25 % and 40% increase in Nusselt number for inline dimple plate and offset dimple plate to that of smooth plate.

In addition, fully three-dimensional and steady-state numerical computations were done to investigate the physical details about the heat transfer in smooth plate, inline dimple plate and offset dimple plate. The numerical computations provided the same trends as experimentally observed with average 15 % variation. The variation is due to environmental errors.

REFERENCES

[1] Sane N.K, 1980 “Some Investigations on intended Tube Heat exchangers” proceedings fifth National Heat and Mass Transfer Conference, Hyderabad(India)

[2] Afanasyev, V. N., Chudnovsky, Y. P., Leontiev, A. I., and Roganov, P. S., 1993, ‘‘Turbulent Flow Friction and Heat Transfer Characteristics for Spherical Cavities on a Flat Plate,’’ Exp. Therm. Fluid Sci., 7, pp.

[3] Kesarev, V. S., and Kozlov, A. P., 1994, ‘‘Convective Heat Transfer in Turbulized Flow Past a Hemispherical Cavity,’’ Heat Transfer Research, 25~2!, pp.156–160.

[4] Terekhov, V. I., Kalinina, S. V., and Mshvidobadze, Y. M., 1995, ‘‘Flow Structure and Heat Transfer on a Surface with a Unit Hole Depression,’’ Russ. J. Eng. Thermophys., 5, pp. 11–33.

[5] Schukin, A. V., Koslov, A. P., and Agachev, R. S., 1995, ‘‘Study and Application of Hemispherical Cavities for Surface Heat Transfer Augmentation,’’ ASME Paper No. 95-GT-59.

[6] Chuy.M.K.,yu., 1997 “Concavity Enhanced Heat Transfer in an Internal Cooling Passage”ASME, Paper No.97-GT-437

[7] Moon, 2000 “Channel height effect on heat transfer and friction in dimpled passage”,ASME Journal of Engg. for Gas Turbines and power,vol.122,pp307-313

[8] G.I.Mahmood, P.M.Ligrani 2001 “Local Heat Transfer and Flow Structure Above a Dimpled Surface in a Channel” ASME Journal of turbo machinery, vol.123, and pp115-123.

[9] Gortyshov, Y. F., Popov, I. A., Amirkhanov, R. D., and Gulitsky, K. E., 1998, ‘‘Studies of Hydrodynamics and Heat Exchange in Channels With Various types of Intensifiers,’’ Proceedings of 11th International Heat Transfer Congress, 6, pp. 83–88.

[10] Moffat, R. J., 1988, ‘‘Describing the Uncertainties in Experimental Results,’’ Exp. Therm. Fluid Sci., (1), pp. 3–17.