Experimental Analysis of Heat Transfer Enhancement over Dimpled Surface on One Side of Plate

Avinash A. Ranaware
Mechanical Engineering Department
ME Student, G. S. Moze COE, Balewadi
Pune, India
aviranaware83@yahoo.in

Iftikarahmed H. Patel
Mechanical Engineering Department
Asst. Prof., SVPM's COE, Malegaon (Bk), Baramati
Pune, India
patelih@gmail.com

Abstract— Many researchers perform enhancement over various dimpled surface. Dimples are used on the surface of internal flow passages because they produce substantial heat transfer augmentation. This work is concerned with experimental investigation of the forced convection heat transfer over the dimpled surface. The experimental analysis objective is to find out the heat transfer and air flow distribution on dimpled surfaces and all the results obtained are compared with those from a flat surface. The varying parameters were i) Inline dimples on the plate i.e. spherical and conical ii) Heat input and iii)Dimple density on the plate.

Heat transfer coefficients and Nusselt number were measured in a channel with one side dimpled surface. The spherical and conical type dimples were fabricated, and the diameter and the depth of dimple were 6 mm and 3 mm, respectively. Two dimple arrangement of inline spherical and conical were tested. The Reynolds number based on the channel hydraulic diameter was varied from 2000 to 8500.Study shown that increase with Reynolds number, thermal performance also increases. With the inline dimple arrangement for spherical and conical arrangements, the heat transfer coefficients, Nusselt number and the thermal performance factors were higher for the conical inline arrangement and also pressure drop penalties smaller than with other types of heat augmentation.

Keywords-*Heat trnsfer; Nusselt number;Reynolds Number;dimple.*

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I. INTRODUCTION

Use of dimples on the surface can significantly intensify the heat transfer enhancement. Some typical examples for use of dimples for heat transfer enhancement are turbine blade cooling, tube heat exchangers in chemical and textile industries, car radiators and on no. of electronic devices like CPU, LED/LCD Monitors etc. Further concept for reduction in thermal resistance and consequent enhancement in heat transfer is to increase the depth of the dimples. Introducing the dimples on the surface not only increase the surface area available for heat transfer but also reduces the hydrodynamic resistance for the fluid flow over the surface, resulting in less pressure drop. The vortices formed inside the dimples results in thinning and to disturb the thermal boundary layer formed over the surface during coolant flow and serve ultimately to bring about enhancement of heat transfer between the fluid and its neighboring surface at the price of less increase in pressure.

II. LITRATURE REVIEW

Moon et. al. [1] studied the channel height effect on heat transfer over the dimpled surfaces. Heat transfer coefficient and friction factors were computationally investigated in rectangular channels, which had dimples on one wall. The heat transfer coefficients were calculated for relative channel heights (H/D ratio of 0.37, 0.74, 1.11 and 1.49) in a Reynolds number range from 12,000 to 60,000.The heat transfer enhancement was reported mostly outside of the dimples. The heat transfer enhancement was lowest on the upstream dimpled wall and highest in the vicinity of the downstream rim (edge) of the dimple. The heat transfer coefficient distribution exhibited a similar pattern throughout the study H/D range (0.37<H/D<1.49).

Patel and Borse [2] have experimentally investigated the forced convection heat transfer over the dimpled surface. The objective of the experiment is to find out the heat transfer and air flow distribution on dimpled surfaces and all the results obtained are compared with those from a flat surface. The results showed that use of dimples on the surface results in heat transfer augmentation in forced convection heat transfer with lesser pressure drop penalty and the value of maximum Nusselt number obtained for staggered arrangement of dimples is greater than that for inline arrangement, keeping all other parameters constant.

Kuethe[3] was the first one to suggest using surface dimples for heat transfer enhancement. Surface dimples are expected to promote turbulent mixing in the flow and enhance the heat transfer, as they behave as a vortex generator.

Mahmood et.al. [4] investigated the effect of dimples on local heat transfer and flow structure over a dimpled channel. Experimental results obtained on and above a dimpled test surface placed on one wall of a channel were given for Reynolds number varying from 1250 to 61500. These include flow visualizations, stream wise velocity and local Nusselt numbers. The H/D ratio was kept constant as 0.5. They reported that the flow visualizations show vortical fluid and vortex pairs shed from the dimples, including a large up wash region and packets of fluid emanating from the central regions of each dimple, as well as vortex pairs and vortical fluid that form near dimple diagonals. These vortex structures augment local Nusselt numbers near the downstream rims of

each dimple, both slightly within each depression, and especially on the flat surface just downstream of each dimple. The results also showed that as the ratio of inlet to wall temperature decreases, the coolest part of the test surface which corresponds to the highest value of baseline Nusselt number ratio (Nu/Nu_o) intensifies and extends farther away from the downstream rims of the dimples.

Mahmood and Ligrani [5] analyzed experimentally the influence of dimple aspect ratio, temperature ratio, Reynolds number and flow structures in a dimpled channel at Reynolds number varying from 600 to 11,000 and H/D ratio varying as 0.20, 0.25, 0.5 and 1.00. The results showed that the vortex pairs which were periodically shed from the dimples become stronger as channel height decreases with respect to the imprint diameter.

Oliveira et.al.[6] studied the Nusselt number behavior on deep dimpled surface . Experimental results were presented for a dimpled test surface placed on one wall of a channel. Reynolds number was varied from 12,000 to 70,000 whereas δ/D ratio was kept as 1.0. These results were compared to measurements from other investigations with different δ/D ratios to provide information on the influences of dimple depth. These results include local Nusselt numbers and globally averaged Nusselt numbers. Results showed that at all Reynolds number considered, local Nusselt number augmentations increases as the δ/D ratio increases from 0.2 to 0.3(and all other experimental and geometric parameters were held constant).

Giram and Patil [7] have studied the heat transfer characteristics and the pressure drop of the forced convection apparatus of six dimpled plates. Six test plates with varying dimple densities were taken and by varying the input voltage Nusselt Number variation was recorded. It was found that Nusselt Number increases as the dimple density increases and also the heat transfer rate from the test surface increases with increase in mass flow rate of flowing fluid and heat input.

Burgess and Ligrani [8] showed the experimental results for the dimple depth to dimple print diameter (δ/D) ratios varying as 0.1, 0.2, and 0.3 to provide information on the influences of dimple depth. They reported that at all Reynolds numbers considered, Nusselt number augmentations increase as dimple depth increases.

Beves et.al.[8] studied the flow structure within a two-dimensional spherical cavity on a flat surface, numerically and experimentally. They observed that the recirculation zone formed inside the cavity slightly reciprocate around itself.

III. Experimental Procedure & Calculations

A. *Experimental Set up*

In this work Strip plate heater was fabricated to provide heat input to the test surface. The capacity of the heater was to vary heat input from 40 W to 140 W. The provision was made to fix the heater at the base of the each test plate. Pressure drop across the test section was measured using micro manometer. In the air flow bench the pipe was used to connect the blower outlet to heat exchange module to carry the forced air from blower to the heat exchange module. Next to the blower outlet, flow regulating valve was connected to the pipe to regulate the air flow. Air flows parallel to the dimpled test surface. Velocity of air flow over plate measure by using turbine meter at exit of experiment set up. The strip plate heater fixed at the bottom of the test plate, was connected

Figure 1. Experimental setup.

to power socket through dimmer stat. Dimmer stat readings were varied to give the required heat input to the test plate. Calibrated Iron-Constantan thermocouple wires were used to measure the temperatures. Provisions were made to fix the thermocouple junction on the test surface. Temperatures of air at inlet and outlet of the heat exchange module are also measured Digital temperature indicators were used to show the temperature readings (in ^oC) recorded by thermocouple wires. Only top dimpled surface of the test plate was exposed to the air stream from which the convective heat transfer to the air stream takes place. The remaining four non-dimpled sides of the test plates were also insulated.

B. *Test Plates*

Figure 2 shows the image and schematic of dimpled plate with spherical inline plate of dimples on the top surface (204 dimples). Figure 3 shows the schematic of conical inline dimpled plate with 192 number of dimples on its top surface.

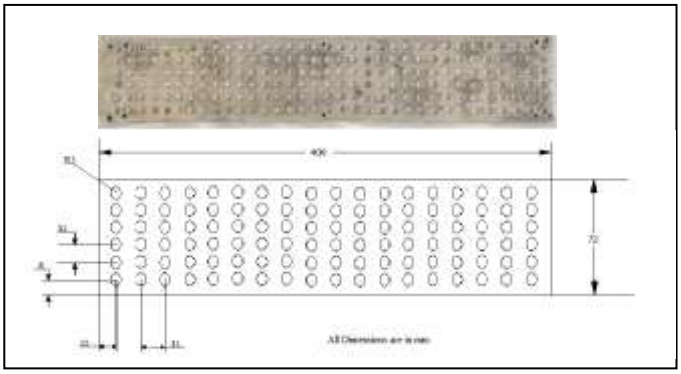

Figure 2. Image and Schematic of test plate with 204 inline spherical dimples.

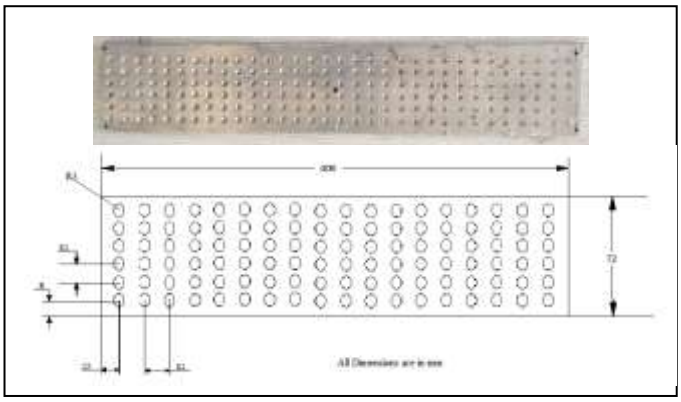

Figure 3. Image and Schematic of test plate with 192 conical inline and dimples.

IV. FORMULAE USED FOR CALCULATION

The mass flow rate of air is determined from flow velocity measured by turbine meter, using a following relation:

$$Q = A \times V$$

Where, Q= Discharge through c/s of test section,
A= C/S area at test section
V= Velocity of Flow

The useful heat gain of the air is calculated as

$$q = m_a \times C_{pa}(T_{ao} - T_{ai})$$

Where,
T_{ao} = Fluid temperature at the exit of the duct (°C),
m_a = Mass flow rate of air
T_{ai} = Fluid temperature at the inlet of the duct (°C),
C_p = Specific heat of air
q = Convective heat transfer to air

The heat transfer coefficient is found out by using Newton's law of cooling, which states that the heat flux from the surface to fluid is proportional to the temperature difference between surface and fluid.

$$h = q/(A \times (T_p - T_{bulk\ mean})$$

Where,
A = Heat transfer area available (m^2) ,
h = Coefficient of Heat Transfer (W/m^2K)
T_p = Average surface temperature,
$T_{bulk\ mean}$ = Mean bulk temperature $(T_{ai} + T_{ao})/2$

Nusselt number based on test plate length is defined as

$$N_u = (h \times L)/K$$

Where, L = Characteristic length,
h= Hydraulic diameter = (4 × Area)/ Perimeter
K = thermal conductivity of air

The Friction Factor (f) is evaluated by:

$$f = 2 \times \Delta P/ [(L/D_h) \rho V^2]$$

Where ΔP is a pressure drop across the test section and V is mean air velocity of the channel. All of thermo physical properties of the air are determined at the mean bulk air temperature.

The thermal enhancement factor, η defined as the ratio of heat transfer coefficient of an augmented surface, h to that of a smooth surface, h_o at the same pumping power.

$$\eta = (h/h_o)_{pp} = (N_u/N_{uo}) (f/f_o)^{(-1/3)}$$

V. RESULT AND DISSCUSSION

The data obtained after experimentation is used to plot different thermal characteristics at different values of dimmer state readings in Volt (V). The details are discussed below.

A. *Friction factor Ratio*

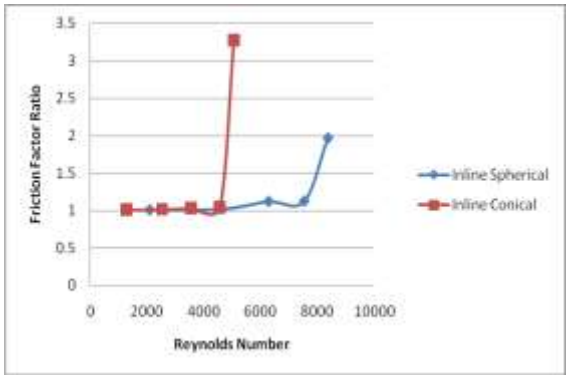

Figure 4. Graph of variation of Friction factor ratio with Re at 40 V

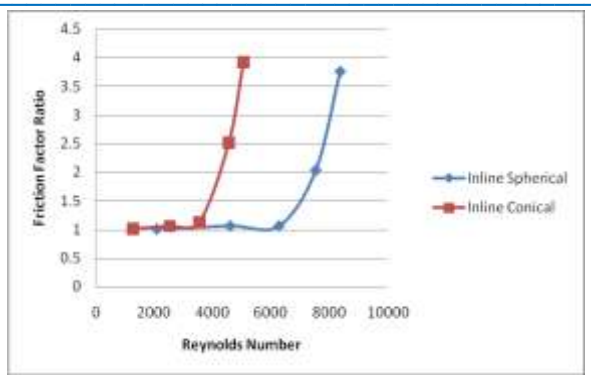

Figure 5. Graph of variation of Friction factor ratio with Re at 80 V

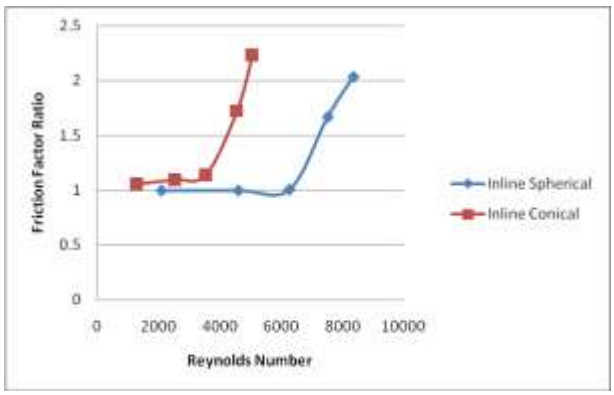

Figure 6. Graph of variation of Friction factor ratio with Re at 120 V

Figure 4 to figure 6 shows the variation of friction factor ratio with Reynolds number for plate with one side dimples. It clearly shows that friction factor ratio for the dimpled surface is far better than for the flat plate. Also it shows that range of friction factor for inline conical dimple is higher than for the inline spherical plate. Because of strong vortex formation in the downstream region friction factor is increased.

B. *Nusselt Number Raio*

Nusselt number ratios are computed for different dimple density on plate, it is found that range of Nusselt number ratio for dimpled plate with conical inline arrangement is higher compared to the dimpled plate with inline spherical arrangement as shown in figure 7, 8, 9. The Nu/Nu_o ratio is higher due to strong vortices formation in conical dimples. Because of this turbulence is more in case of conical dimples.

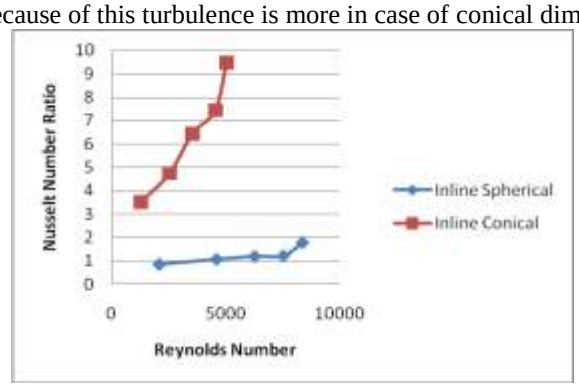

Figure 7. Graph of variation of Nusselt Number ratio with Re at 40 V

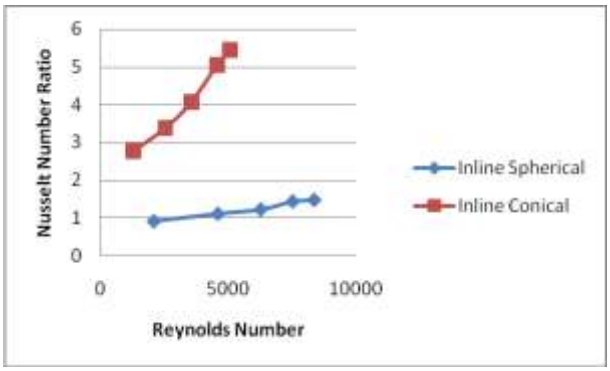

Figure 8. Graph of variation of Nusselt Number ratio with Re at 80 V

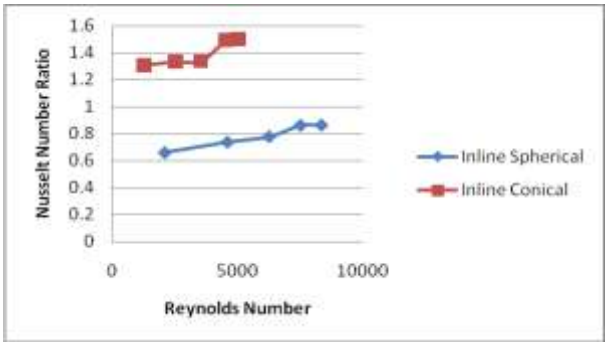

Figure 9. Graph of variation of Nusselt Number ratio with Re at 120 V

C. *Thermal Performance*

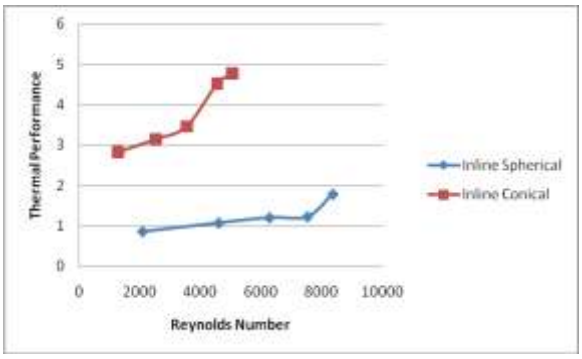

Figure 10. Graph of variation of Thermal Performance with Re at 40 V

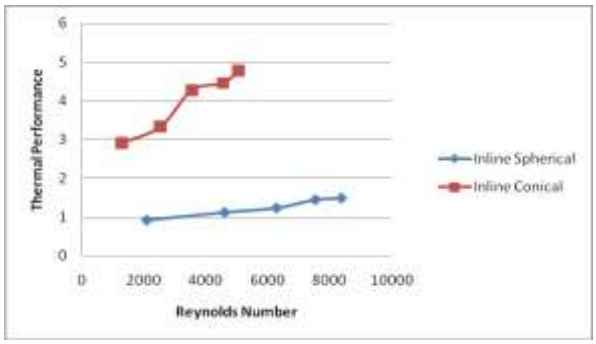

Figure 11. Graph of variation of Thermal Performance with Re at 80 V

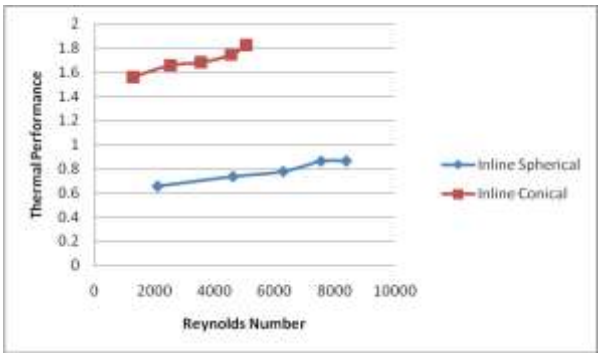

Figure 12. Graph of variation of Thermal Performance with Re at 120 V

Thermal performance for the plate with spherical dimples and conical dimples is drawn in Figure 10, 11 and 12 shows that thermal performance is increasing with increase in Reynolds number. But the thermal performance plate with inline spherical dimples is poor as compared to the plate with inline conical dimples. This is due to more turbulence and strong vortex formation in conical dimple plate. Also it reflects that thermal performance of conical dimple plate is more than spherical dimple plate which means applying inline conical dimples on plate is beneficial to increase heat transfer enhancement.

VI. CONCLUSION

In the present work aluminum plates of dimensions 400x72x6 mm3 were used as test surfaces. Variation of Nusselt numbers with Reynolds numbers is investigated, with various parameters combinations. Effect of dimple density and dimple arrangement on heat transfer in terms of Nusselt number enhancement is also reported. The study concludes to more heat transfer enhancement on dimpled surfaces with lesser pressure drop penalty. The present work was towards experimental determination of effect of dimples on heat transfer over a flat surface under forced convection condition.
The main conclusions are summarized as:

1. With increase in Reynolds number of flowing fluid and heat input, heat transfer rate from the test surface increases
2. The use of dimples on the surface results in heat transfer augmentation in forced convection heat transfer with lesser pressure drop penalty.
3. The value of maximum Nusselt number obtained for staggered arrangement of dimples is greater than that for inline arrangement, keeping all other parameters constant. It shows that for heat transfer enhancement staggered arrangement is more effective than the inline arrangement.
4. At all Reynolds number considered Nusselt number augmentation increases as the dimple density of test plates increases (all other experimental and geometric parameters are kept constant).This is because the more number of dimples produce:(i) increase in the strength and intensity of vortices and associated secondary flows ejected from the dimples (ii) increases in the magnitudes of three-dimensional turbulence production and turbulence transport. More number of dimples beyond a particular value is believed to trap fluid which then acts as a partially insulating pocket to decrease the rate of Nusselt number enhancement with increase in further dimple density. It also results in decrease in rate of Nusselt number enhancement after a certain value of dimple density of plate.

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