International Journal of Latest Trends in Engineering and Technology
Vol.(9)Issue(1), pp.225-229
DOI: http://dx.doi.org/10.21172/1.91.34
e-ISSN:2278-621X

HEAT TRANSFER ANALYSIS ON DIMPLED SURFACE BY USING FORCED CONVECTION- AN EXPERIMENTAL APPROACH

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Abstract-Over the past years the focus on using dimples provides enhanced heat transfer has been documented by a number of researchers. Dimples are used on the surface of internal flow passages because they produce substantial heat transfer augmentation. This work is concerned with heat transfer enhancement by experimental investigation by using forced convection method of heat transfer over the dimpled surface. The objective of the experiment is to find out the heat transfer at different discharge of air flow on dimpled surfaces and all the results obtained are compared those with a flat surface. This study evaluated the heat transfer characteristics using two different dimpled shapes on a wall of rectangular channel by experimental: 1) Circular (spherical) dimples, and 2) Conical dimples. The average heat transfer coefficient and heat transfer performance were obtained experimentally. An array of staggered spherical and conical dimples with rounded sharp edges (r=6mm) are located on the bottom side of the rectangular channel (with the height h= 25.4mm) at whose entrance a profile of completely developed turbulent air flow is assigned when the Reynolds number Re is varied over the range of 1,000 to 10,000. The dimple density, dimple arrangement and the mean mass flow velocity V are chosen as characteristic parameters. Channel side walls and bottom wall are thermally insulated. Natural convection effects are neglected. A moderate depth (d=3mm) dimple is considered. With the staggered dimple arrangement, the heat transfer coefficients, Nusselt number and the thermal performance factors were higher for the staggered conical arrangement.
Keywords – Heat Transfer Coefficient, Reynolds Number, Nusselt number.

1. INTRODUCTION

The importance of heat transfer enhancement has gained greater significance in such areas as microelectronic cooling, especially in central processing units, macro and micro heat exchangers, gas turbine internal airfoil cooling, fuel elements of nuclear power plant, and bio medical devices. A tremendous amount of efforts has been devoted to developing new methods to increase heat transfer from finned surfaces to the surroundings owing fluid. Rib turbulators, an array of pin fins have been employed for this purpose.

Heat transfer augmentation using these methods always results in pressure drop penalties that adversely an effect aerodynamics and efficiencies. In the case of cooling of turbine blades, surface protrusions induce excessive pressure loss, which increases the compressor load. The separated flow field over ribs or pin fins can make significant non uniform cooling, which leads to thermal stresses. For the case of concavities or dimples on the surfaces of internal flow passages, it produces substantial heat transfer augmentations. Because of easiness in manufacturing, dimples are also attractive as a heat transfer augmentation device.

2. LITRATURE REVIEW

A variety of experimental, analytical and computational research work has been carried out on enhancement of heat transfer. Specially, the heat transfer enhancement through manipulation of surface geometry have concerned by many researcher and practitioners. Relevant literature pertaining to enhancement of heat transfer by introducing protrusions mounted on the heat transfer surface, reviewed from different points is presented here.

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

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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 and Ligrani [4] 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.[5] 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).

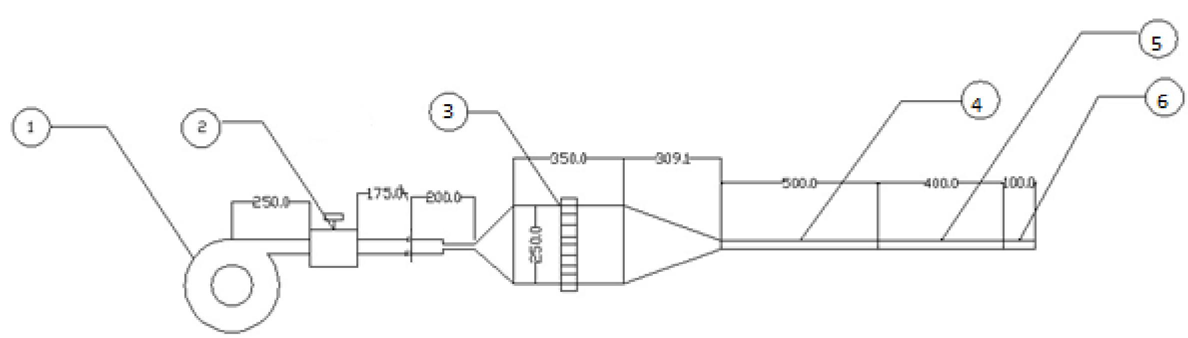
Giram and Patil [6] have studied the heat transfer characteristics and the pressure drop of the forced convection apparatus of six dimpled plates. Six test plates with varying dimple densities were taken and by varying the input voltage Nusselt Number variation was recorded. It was found that Nusselt 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.

Ya-Ling He et.al.[7] studied and concluded that ellipsoidal dimple tube has better performance on heat transfer relatively low penalty of pressure drop than spherical dimpled tube .The performance study shows that ellipsoidal dimple tube is suitable for Re ranging between 1500 and 60000. For heat exchanger application, ellipsoidal dimpled tubes can be used to improve the overall performance efficiency and reduce the size of heat transfer system.

3. EXPERIMENT AND RESULT

3.1 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 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 ^{o}C) 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. Figure 1 shows experimental setup for analysis.



(All dimensions are in mm)

Figure 1. Schematic of Experimental set-up Arrangement

Where, 1- Blower, 2- Flow control valve, 3- Flow straightner with honeycomb structure,
4- Entrance section, 5- Test section, 6- Exit section, 7- Anemometer

3.2 Test Plates

Fig.2 shows the image and schematic of dimpled plate with spherical staggered 187 dimples on the top surface of plate and conical staggered 192 dimpled on its top surface.

(a)

(b)

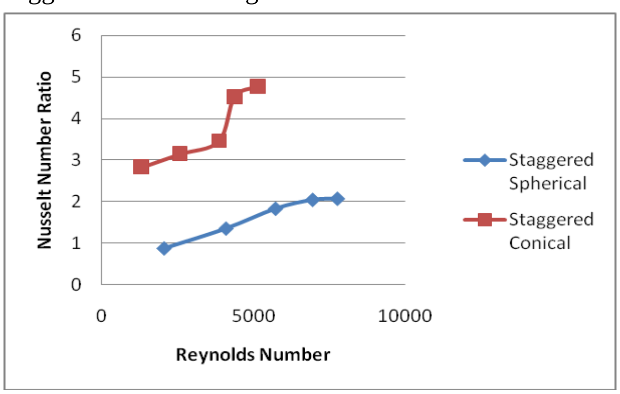
Figure 2. (a) Staggered spherical plate with 187 dimples and (b) Staggered conical plate with 192 dimples

4. RESULT AND DISCUSSION:

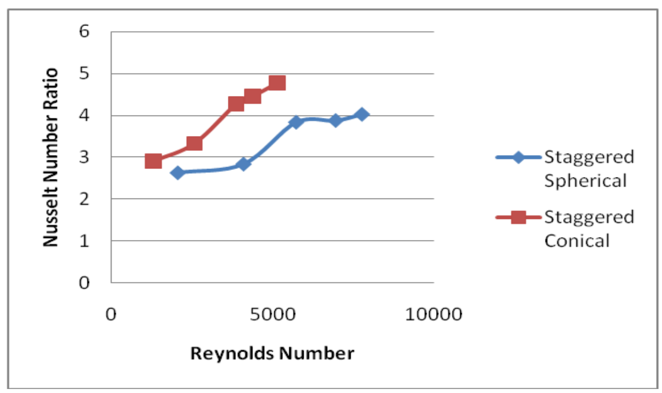
The data obtained after experimentation is used to plot different thermal characteristics. The details are discussed below.

4.1 Nusselt Number Ratio:

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 staggered arrangement (192 dimples) is higher compared to the dimpled plate with staggered spherical arrangement (187 dimples) as shown in Fugure 3. The Nu/Nu_o ratio is higher due strong vortices formation in staggered conical arrangement. Because of this turbulence is more in case of staggered conical dimples.



(a)



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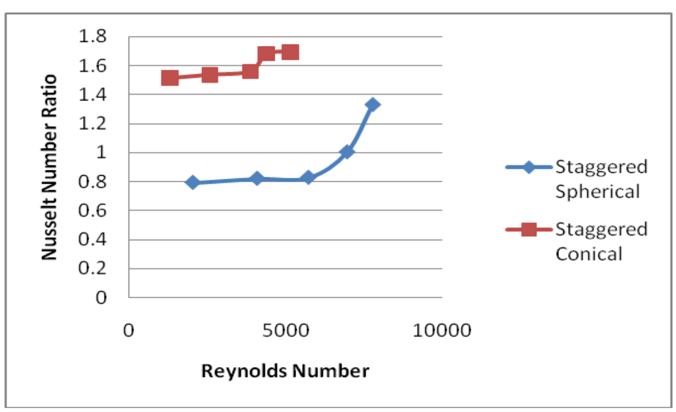
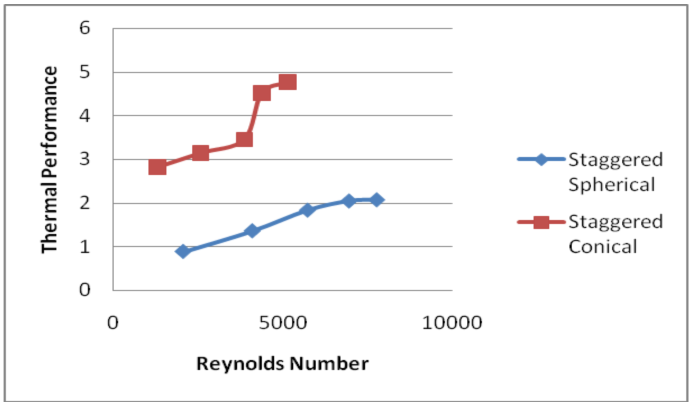


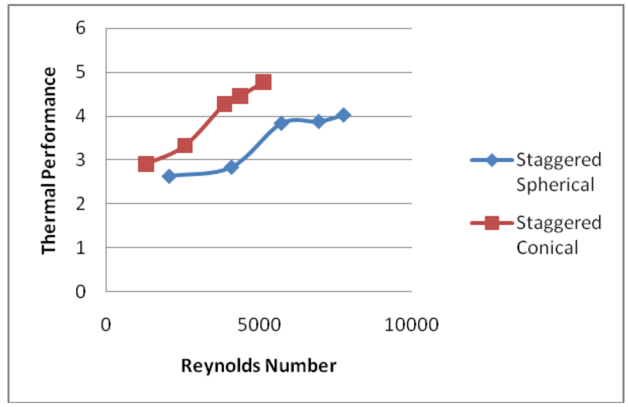
(c)

Figure 3.Graph of variation of Nusselt number ratio with Reynolds number at input voltage (a) 40 V, (b) 80 V, (c) 140 V

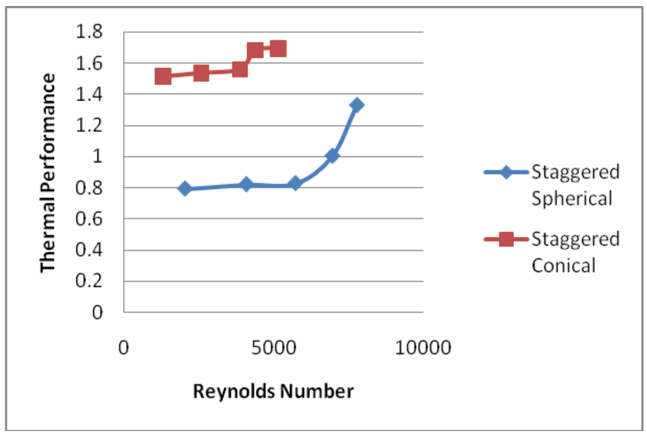
4.2 Thermal Performance:



(a)



(b)

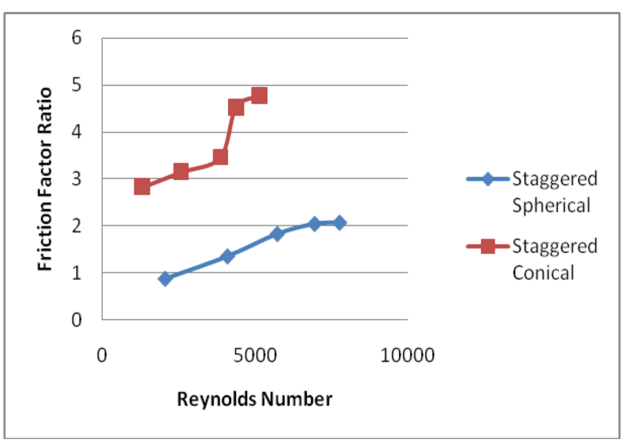


(c)

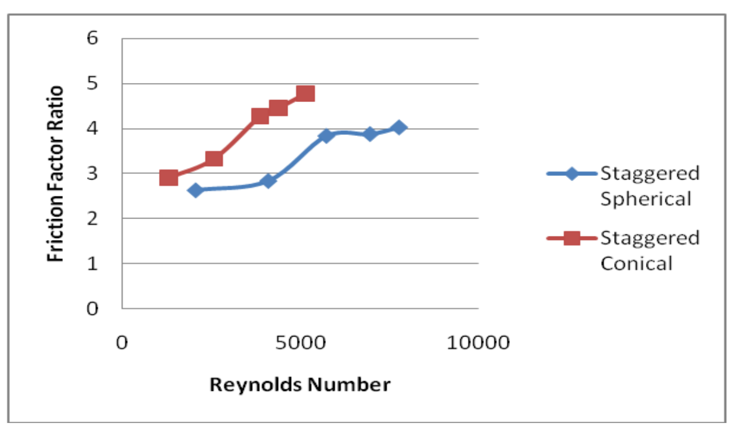
Figure 4.Graph of variation of Thermal Performance with Reynolds number at input voltage (a) 40 V, (b) 80 V, (c) 140 V

Thermal performance for the plate with spherical dimples and conical dimples of staggered arrangement is drawn in figure 4 for 40 V, 80 V and 140 V respectively, shows that thermal performance is increasing with increase in Reynolds number. But the thermal performance plate with staggered spherical (187) dimples is poor as compared to the plate with staggered conical (192) dimples. This is due more turbulence and strong vortex formation in staggered conical dimple plate. Also it reflects that thermal performance of staggered conical dimples is more than other which means applying dimple with staggered arrangement in beneficial to increase heat transfer enhancement.

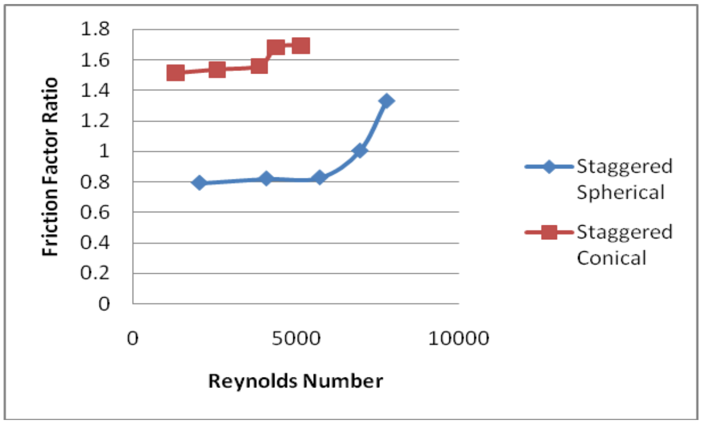
4.3 Friction Factor Ratio:



(a)



(b)



(c)

Figure 4.Graph of variation of Friction Factor Ratio with Reynolds number at input voltage (a) 40 V, (b) 80 V, (c) 140 V

Figure 4 (a), (b) and (c) shows the variation of friction factor with Reynolds number for plate with one side staggered dimples. It clearly shows that friction factor for the dimpled surface is far better than for the flat plate. Also it shows that range of friction factor for staggered conical arrangement is higher than for the staggered spherical arrangement. Because of strong vortex formation in the downstream region friction factor is increased.

5.CONCLUSION

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. Heat transfer rate from the test surface increases, with increase in Reynolds number of flowing fluid and heat input.
2. The value of maximum Nusselt number obtained for staggered conical arrangement of dimples is greater than that for staggered spherical arrangement, keeping discharge and voltage input constant. It shows that for heat transfer enhancement of conical staggered arrangement is more effective than the staggered spherical arrangement.
3. At all Reynolds number considered Nusselt number augmentation increases as the dimple density of test plates increases. This is because the more number of dimples produce increase in the strength and intensity of vortices and associated secondary flows ejected from the dimples.

Based on the plates we used, it can be concluded that staggered arrangement of conical dimples gives us maximum heat transfer rate compared to the staggered arrangement of spherical dimples.

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