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Experimental Investigation of Forced Convection Heat Transfer Augmentation from a Dimpled Surface

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ABSTRACT

Over the past couple of years the focus on using concavities or 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 project work is concerned with experimental investigation of the forced convection heat transfer over the dimpled surface. The objective of the experiment is to find out the convective heat transfer coefficient and pressure drop across the test plate. The varying parameters were i) Dimple arrangement on the plate i.e. staggered and inline arrangement and ii) Heat input iii)Dimple diameter on the plate. Convective heat transfer coefficients and Nusselt number were measured in a channel. The spherical type dimples were fabricated, and the diameter and the depth of dimple were 10,15 mm and 5,4 mm, respectively. The Reynolds number based on the channel hydraulic diameter was varied from 4000 to 9000.Study shown that thermal performance is increasing with Reynolds number. With the inline and staggered dimple arrangement, the heat transfer coefficients, Nusselt number and the thermal performance factors were higher for the staggered arrangement.

***Keywords*— Convective heat transfer, Dimpled plate, Turbulent flow, Inline arrangement, Staggered arrangement, Forced convection.**

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I. INTRODUCTION

There are various techniques used for the purpose of heat transfer augmentation. Forced convection heat transfer is one of the passive techniques used to improve the heat transfer rate. To increase the heat transfer rate, there is need to increase the surface area. In order to increase the surface area, extended fins was implemented. Recently, dimples are being used in the heat transfer augmentation technique, which results in the increase of heat transfer rate with lesser pressure drop.This is the experimental investigation of forced convection heat transfer augmentation from a dimpled surface. In this investigation, the test plate material is made up of copper. The circular shaped dimples are printed on the test plate arranged in the staggered manner. Heat input given to the test plate is the only varying parameter. To enhance the heat transfer in some applications like turbine blade cooling, tube heat exchangers in chemical and textile industries, car radiators, electronic devices dimpled surface is used.

For the investigation of heat transfers using forced convection, the set-up mainly consist of: Blower, Flow control valve, Orifice meter, Rectangular duct, Test plate with

dimpled surface, Control unit- heater, pressure measurement device, temperature indicator. Blower is used to supply air. Heater is fixed at the bottom of the test plate, is connected to power socket through dimmer stat. Control unit is used to show the temperature and pressure readings.

Many researchers are carried out experimental and numerical investigations over dimpled surface with different materials such as mild steel, aluminium. There need is to study the various parameters such as heat transfer rate, pressure drop across the dimple surface, Reynolds number, Nusselt number, flow rate, friction factor and dimple diameter to print diameter, on which the thermal performance depends.

II. LITERATURE REVIEW

Number of researchers worked in this field and there obtained results and conclusion were discussed briefly below.

Forced convection heat transfer in smooth and roughened ducts has been investigated by several investigators. Also the performance over a dimpled surface for the improvement of heat transfer has been investigated by many researchers.

Kuethe et al. [1], was the first one who suggested the use of dimpled surface for heat transfer enhancement. Dimpled surfaces are expected to promote turbulent mixing in the flow and enhance the heat transfer as they behave vortex generator.

Mahmood et al. [2], investigated the effect of dimples on local heat transfer and flow structure over a dimpled channel. Experimental results obtained on a dimpled test surface placed on one wall of a channel were given for Reynolds number varying from 1250 to 61500. The H/D ratio was kept constant as 0.5.

Ligrani et al. [3], then added protrusions on the wall opposite to the dimpled wall, and found the protrusions produced more vertical and flow mixing resulting higher heat transfer enhancement and higher pressure drop compared to channel with dimpled smooth opposite walls.

Mahmood et al. [4], analysed 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. As the temperature ratio decreases, the local Nusselt number increases.

S.S. Kore et al. [5], presented experimental investigation to study heat transfer and friction coefficient by dimple surface. The aspect ratio of rectangular channel is kept 4:1 and Re no. based on hydraulic diameter ranges from 10,000 to 40,000. Author concluded that at all Re no. a depth increases from 0.2 to 0.3, the Nu no. and thermal performance increases and when depth increases from 0.3 to 0.4, the Nu no. and thermal performance decreases. This is because of increase in strength and intensity of vortices.

Patel et al. [6], has reported for heat transfer coefficient and Nusselt number measured in a channel with one side dimpled surface. The spherical type dimples were fabricated and the diameter and depth of the dimple were 6 mm and 3 mm respectively. Channel height is 25.4mm. The Reynolds number based on the channel hydraulic diameter was varied from 5000 to 15000. This study shows that thermal performance is increasing with Reynolds number with in-line and staggered dimple arrangement, the heat transfer coefficients, Nusselt number and the thermal performance factors were higher.

Tay et al. [7], has experimented for the development of flow structures over dimples. In this experimental study dimple with depth to diameter ranging from 5% to 50% shows six different flow stages as Reynolds number varies from 1000 to 28000. This flow development can be achieved either by changing the Reynolds number or dimple depth to diameter ratio. Dimples with low depth to diameter ratios show fewer development than deeper ones. Sharp and round edged dimples both show similar behaviour.

Yu Rao et al. [8], presented that experimental study of effects of dimple depth on the pressure loss and heat transfer characteristics in a pin-fin dimple channel, where dimples are located on the end wall transversely between the pin fins. The dimple channel consists of ten rows of pin-fin dimpled combined structure. The transverse spacing to diameter ratio S/D= 2.5, H/D= 1.0.

The dimples have same diameter but different depths to diameter ratios, i.e. δ/ D= 0.1, 0.2and 0.3. Reynolds number range of 8,200- 50,500.

Giram and Patil et al. [9], analysed that heat transfer characteristics of plate with dimpled surface. It is found that Nusselt number increases as dimple density increases. Also it was found that percentage increase in Nusselt number is greater for staggered dimple arrangement.

III.EXPERIMENTAL RIG

Actual experimentation will be conducted to investigate the effect of dimple on the heated plate in the forced convection heat transfer from heated plate. The heat will be generated within the plate with plate type heating element located below heating plate. An arrangement will be made to measure and vary the heat input with the help of transformer, and additional measuring instruments like voltmeter and ammeter. A provision is made to measure the mass flow rate of flowing air with orifice meter and U-tube manometer. The surface temperatures will be measured with the help of RTD's (PT-100 sensors) mounted at different locations of plates. Ten RTD's are to be fixed on the plate (5 on test sides) in order to measure the base temperature and one RTD is to be used to measure air temperature. Each RTD (PT-100 sensor) is to be fixed to the surface of the test plate at equal space locations along the plate length.

Another inclined tube manometer is used to measure pressure drop across the test section. The apparatus will be allowed to run until the steady state for particular heat input. The recording of temperature will begin after steady state has been reached. Same procedure will be repeated for different heat input and readings will be noted at steady state. Heat input to heater will be varied from 20W to 100 W in the step of 20W.

The procedure is used to take the reading on setup includes

1) Make all electrical connections i.e. main supply to Dimmerstat from Dimmerstat to voltmeter and ammeter after this supply is connected to the plate type heater. For blower and digital temperature indicator also required electrical connections.
2) After connection all start blower adjust mass flow rate as 5×10^{-3} kg/s and adjust the wattage given to heater as 20 W and for same wattage adjust the mass

flow rate as 6×10^{-3}, 7×10^{-3}, and 8×10^{-3} and for constant wattage of 20W and for different mass flow rate note down corresponding temperature readings till steady state is reached with respect to time.

3) Next step is to note down the properties of air at bulk mean temperature properties include density, specific heat, Dynamic viscosity etc.
4) This stage we have to calculate Reynolds number, Prandtl number from available data. And find out the convective heat transfer coefficient and this experimental convective heat transfer coefficient is compared with Theoretical convective heat transfer coefficient.
5) Calculate the drop of pressure across the test plate from readings obtained in inclined manometer

Figure No. 1. Schematic of Experimental setup

Fig. 2 Image of Rectangular plate with dimple diameter 15mm and inline arrangement (config. A)

Fig. 3 Image of Rectangular plate with dimple diameter 15mm and staggered arrangement (Config. B)

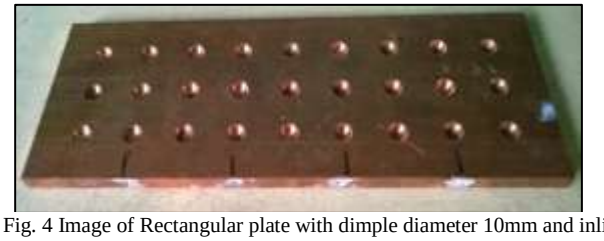

Fig. 4 Image of Rectangular plate with dimple diameter 10mm and inline arrangement (Config. C)

Fig. 5 Image of Rectangular plate with dimple diameter 10mm and Staggered arrangement (Config. D)

IV. DATA REDUCTION

To obtain Experimental convective heat transfer coefficient following method is employed.

$Q= h_{exp}\times A\times \Delta T$

$h=Q/ (A\times \Delta T)$

Where,

Q= Total heat supplied in Watt considering losses of conduction and radiation

A= Surface area of plate in m^2

And $\Delta T = T_w-T_b$

Where T_w= Surface wall temperature

T_b= bulk mean temperature

$$Tb =\frac{Tin+Tout}{2}$$

(T_{in}, T_{out}) are inlet and outlet temperatures and T_w is the tube wall temperature which is the mean value measured by the four surface RTDs.

$$Tw=\frac{T1+T2+T3+T4}{4}$$

To obtain Experimental convective heat transfer coefficient following method is employed.

We know that correlation for Nusselt number for Dimpled surface

$Nu = 0.023\times Re^{0.8}\times Pr^{0.4}$

Reynolds number is calculated as follows

$$Re =\frac{m\times Dch}{Ach\times\mu}$$

Where, m = Mass flow rate of air in kg/s

D_{ch}= Hydraulic meandiameter in m

A_{ch} = Area of rectangular channel in m^2

μ = Dynamic viscosity in Kg/ms

$$Re =\frac{m\times Dch}{Ach\times\mu}$$

Prandtl Number is calculated as,

$$Pr =\frac{\mu\times Cp}{K}$$

Where,
μ = Dynamic viscosity in Kg/ms
Cp = Specific heat of air in J/Kg K
K = Thermal conductivity W/mK

$$Nu=\frac{hth\times Dch}{K}$$

$$h_{th}=\frac{Nu\times K}{Dch}$$

After completing all calculations we compared $\mathbf{h_{th}}$ and $\mathbf{h_{exp}}$. As both values are near to each other hence developed setup is validated and can further proceed for readings on dimpled plates.

V. RESULTS AND DISSCUSSION

A number of experimental runs are carried out on the setup with plain copper plate and by changing the wattage input and varying mass flow rate of the air by the mentioned testing methodology the obtained results are presented in the following section.

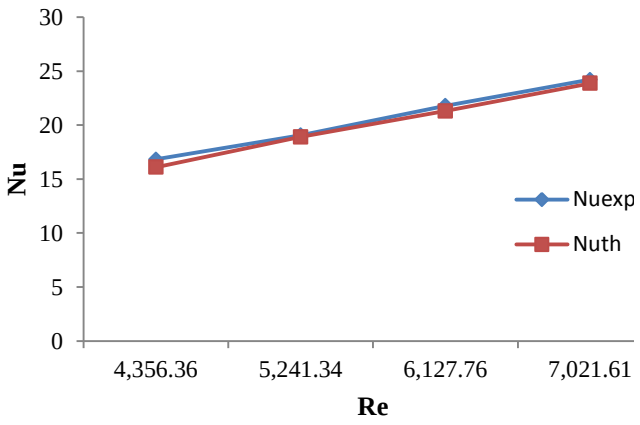

Fig No.6:- Graph of Nu V/S Re for plain Plate no. 01 @ P= 20 W

From above graphs of Nu V/S Re for plain copper plate we clearly identify that there is slightly variation in the values of theoretical Nusselt number and experimental Nusselt number for Reynolds number. Range is of Reynolds number various from 4000 – 7500 and for these values curves of Nusselt number theoretical and experimental variation is there but from this we can conclude that setup is validated and further readings on same setup for Dimpled plates taken and convective heat transfer coefficient is calculated and compared with the plain plate.

As Reynolds number increases the Nusselt number also increases as shown in above.

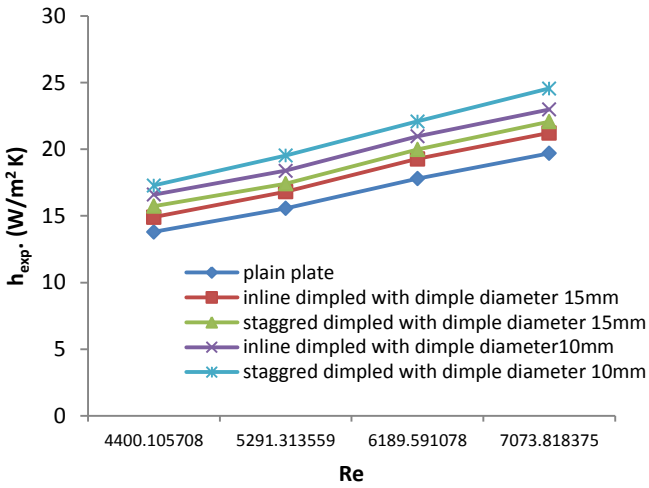

Fig No. 7:- Graph of h_{exp} V/S Re for all Plates @ P= 20 W

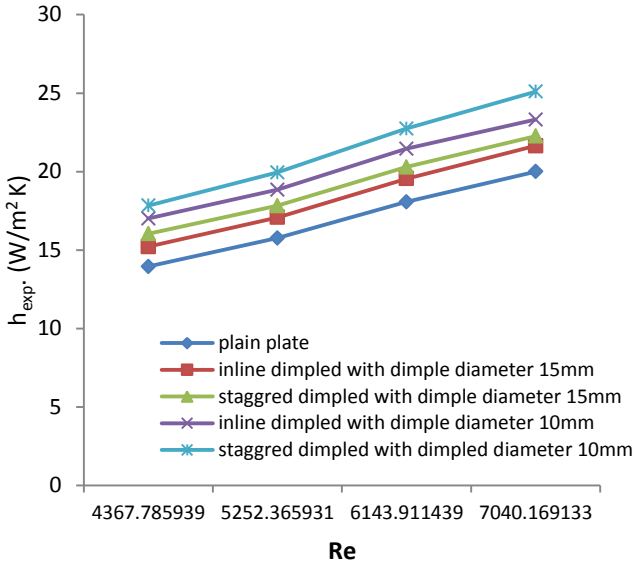

Fig No. 8:- Graph of h_{exp} V/S Re for all Plates @ P= 40 W

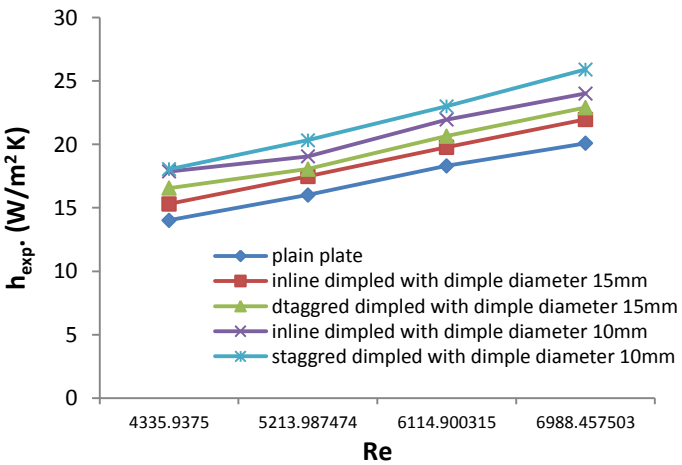

Fig No. 9:- Graph of h_{exp} V/S Re for all Plates @ P= 60 W

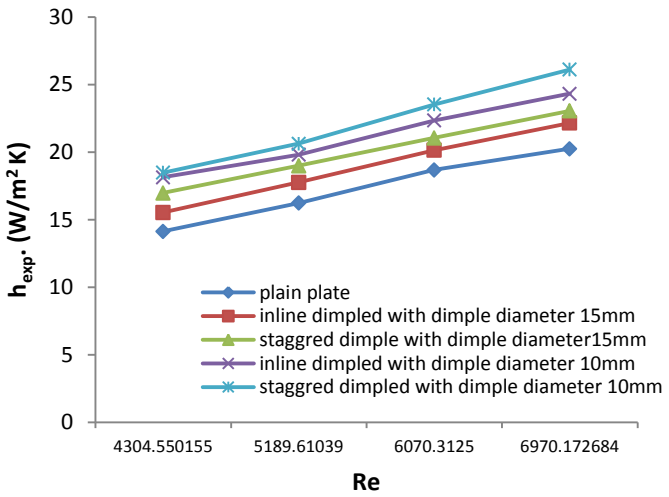

Fig No. 10:- Graph of h_{exp} V/S Re for all Plates @ P= 80 W

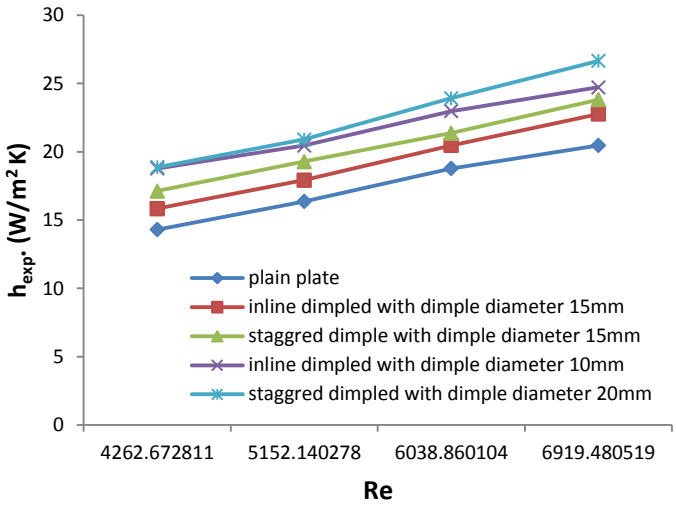

Fig No. 11:- Graph of h_{exp} V/S Re for all Plates @ P= 100 W

From all above graphs of $\mathbf{h_{exp}}$ V/S Re for all plates we clearly identify that there is more enhancement in the convective heat transfer coefficient for plate having staggered dimple with small dimple diameter 10mm and depth of dimple 4mm. and less enhancement with dimpled having inline arrangement with 15mm dimple diameter and depth of dimple 5mm. As Reynolds number increases convective heat transfer coefficient increases is clearly shown by the graph for all readings

VI. RESULTS AND DISSCUSSION

The experimental investigation of forced convective heat transfer from dimpled surface is calculated for different dimple diameter configurations with either inline or staggered arrangement of dimples and for single plain plate with varying heat supplied and different mass flow rates of air after completing experiment for all readings of plain plates and dimpled plates of different dimple diameter configurations theoretical and experimental convective heat transfer coefficient is calculated the results obtained were found to agree with literature data and the following conclusions were made.

1. As the heat supplied to heater increases convective heat transfer rate increases
2. As the mass flow rate increased it increases the Nusselt number
3. When compared with plain plate maximum convective heat transfer coefficient is for dimpled plate with staggered arrangement having diameter of dimple 10 mm and depth of dimple 4mm.
4. When compared with plain plate minimum convective heat transfer coefficient is for dimpled plate for inline arrangement having dimple diameter 15mm and dimple depth 5mm.
5. From obtained result for configuration “A” it is seen that 8% increase in convective heat transfer coefficient compared with plain plate.
6. From obtained result for configuration “B” it is seen that 12% increase in convective heat transfer coefficient compared with plain plate.
7. From obtained result for configuration “C” it is seen that 17% increase in convective heat transfer coefficient compared with plain plate.
8. From obtained result for configuration “D” it is seen that 25% increase in convective heat transfer coefficient compared with plain plate.
9. As dimple depth increases up to certain limit of depth convective heat transfer rate increases after that it decreases.

REFRENCES

[1]. Kuethe A.M. 1971, “Boundary Layer Control of Flow Seperation and Heat Exchange,” US Patent No. 3,578,264.

[2]. Mahmood G.I., Hill M.L., Nelson D.L., Ligrani, P.M., Moon H.K., and Glezer B. “Local Heat Transfer and Flow Structure On and Above a Dimpled Surface in a Channel”, ASME Journal of Turbo machinery, 123(1), January (2001), 115-123.

[3].P. M. Ligrani, G. I. Mahmood , “Flow structure and local Nusselt number no variation in channel with dimple and protrusion on opposite walls”, International Journal of Heat and Mass Transfer, vol.44, (2001),4413-4425.

[4]. Mahmood, G. I., Ligrani, P. M., “Heat Transfer in a dimpled channel: combined influences of aspect ratio, temperature ratio, Reynolds number and flow structure”, International Journal of Heat and Mass Transfer 45, 2011-2020, 2002.

[5]. Sandeep S. Kore, S. V. Joshi, N.K. Sane, “Experimental investigations of heat transfer enhancement from dimpled surface in a channel”, International Journal of Engineering Science and Technology, Vol.03,(2011), 6227-6234.

[6]. Iftikarahamad H. Patel, “Experimental investigation of heat transfer enhancement over the dimpled surface”,

International Journal of Engineering Science and Technology, vol.4, (2012), 3666-3673.

[7].C. M. Tay, Y. T. Chew, B. C. Khoo, J. B. Zhao, (2013) Elsevier "Development of Flow Structure Over Dimples".

[8]. Yu Rao, Chaoyi Wan, YaminXu, "An experimental study of pressure loss and heat transfer in the pin fin-dimple channels with various dimple depths", International Journal of Heat and Mass Transfer, (2012).

[9].Dhananjay R. Giram, A. M. Patil, "Experimental and theoretical analysis of heat transfer augmentation from dimple surface", International Journal of Engineering Research and Applications, vol.3, Issue 5, Sep- Oct (2013), 19-23.

[10] R.K.Ali 2009 Elsevier Inc "Augmentation of heat transfer from heat source placed downstream a guide"

[11] Arthur E. Bergles "high-flux processes through enhanced heat transfer"

[12] YaroslavChudnovsky 2004 AI ChESpring Technical Meeting "vortex heat transfer enhancement for chemical industry fired heaters

[13] Qiuwang Wang , Qiuyang Chen, Ling Wang , Min Zeng, Yanping Huang, Zejun Xiao ,2006 Elsevier " Experimental study of heat transfer enhancement in narrow rectangular channel with longitudinal vortex generators"

[14] Srinath V. Ekkad, David Kontrovitz 2002 Elsevier Science "jet impingement heat transfer on dimpled target surfaces"

[15] A.Slanciauskas oct1999 Elsevier "Two friendly rules for the turbulent heat transfer enhancement"

