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Heat Transfer Enhancement by Using Dimpled Surface

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***Abstract*— The importance of heat transfer enhancement has gained greater significance in such areas as microelectronic cooling, especially in central processing units, macro and micro scale heat exchangers, gas turbine internal airfoil cooling, fuel elements of nuclear power plants, and bio medical devices. A tremendous amount of effort has been devoted to developing new methods to increase heat transfer from fined surface to the surrounding flowing fluid. The experiment will be carried out for laminar forced convection conditions with air as working fluid.**

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 varying parameters will be-

i) Dimple arrangement on the plate i.e. staggered and inline arrangement.

ii) Heat input.

I. Introduction

Enhancing the efficiency of heat transfer is useful in variety of practical applications such as macro and micro scale heat exchangers, gas turbine internal aerofoil cooling, fuel elements of nuclear power plants, powerful semiconductor devices, electronic cooling, combustion chambers liners, biomedical devices etc. Compact heat exchangers and gas turbine aerofoil cooling are two applications which have been the subject of study for a number of researchers over the recent years. Compact heat exchangers are used extensively in trucking industry as radiators to reduce the excess thermal energy. Improved efficiency of compact heat exchanger scan permit smaller radiators leading to smaller frontal area & thus can lead to substantial fuel saving. In a compact heat exchanger there are three important aspects of heat transfer. The first aspect to consider is the convection of heat from the fluid to the tube wall of heat exchanger. Then the heat is conducted through walls of tube. Finally, the heat is removed from tube surface by convection to the air flowing through it. Air-side resistance to heat transfer in heat exchangers comprises between 70-80% of the total resistance & hence any improvement in the efficiency of a compact heat exchangers is focused on augmenting the air side convective heat transfer.

II. HEAT TRANSFER AUGMENTATIO TECHNIQUES

Heat transfer augmentation is of special interest in channel flow where the rate of heat transfer between fluid & channel wall deteriorates as boundary layers grows on channel walls & flow tends to become fully developed augmentation techniques can be classified as active & passive methods.

A. *Active Techniques:*

These techniques do not require any direct input of external Power, rather they Use it from the system itself which ultimately leads to an increase in fluid pressure drop. They generally use surface or geometrical modifications to the Flow channel by incorporating inserts or additional devices.

- Dimpled Surfaces are used for heat transfer augmentation. Instead of protruding the area in the flow stream, concavities or impression are imprinted inside the surface.
- Extended surfaces: They provide effective heat transfer enlargement. The newer developments have led to modified finned surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.
- Coiled tubes: These lead to relatively more compact heat exchangers. It produces secondary flows and vortices which promote higher heat transfer coefficients in single phase flows as well as in most regions of boiling.
- Swirl flow devices: They produce and superimpose swirl flow or secondary recirculation on the axial flow in a channel. These include helical strip or cored screw type tube inserts, twisted tapes. They can be used for single phase and two-phase flows.
- Rough surfaces: These are the surface modifications that promote turbulence in the flow field in the wall region, primarily in single phase

flows, without increase in heat transfer surface area.

B. *Passive Techniques:*

In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer.

- Jet impingement: It involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface.
- Surface vibration: They have been applied in single phase flows to obtain higher heat transfer coefficient.
- Mechanical Aids: Such instruments stir the fluid by mechanical means or by rotating the surface. These include rotating tube heat exchangers and scrapped surface heat and mass exchangers.
- Fluid vibration: These are primarily used in single phase flows and are considered to be perhaps the most practical type of vibration enhancement technique.
- Suction: It involves either vapor removal through a porous heated surface in nucleate or film boiling, or fluid withdrawal through a porous heated surface in single-phase flow.

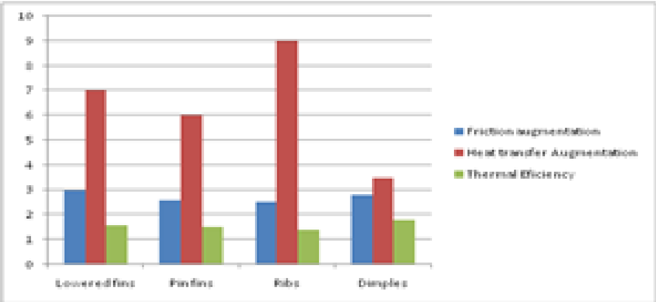

Fig. 1: Thermal efficiency of various heat techniques

III. LITERATURE REVIEW

Kuethe [1] 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.

Moon et.al. [2] studied the channel height effect on heat transfer over the dimpled surfaces. Heat transfer coefficient and friction factors were experimentally investigated in rectangular channels which had dimples on one wall. The heat transfer coefficients were measured for relative channel height (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 studied H/D range ($0.37<H/D<1.49$). The heat transfer coefficient on a dimpled wall was approximately constant at a value of 2.1 that of a smooth channel over the entire H/D range in the thermally developed region. The heat transfer enhancement ratio reported was independent with Reynolds number. The friction factor in the aerodynamically fully developed region were consistently measured around 0.0412(only 1.6to 2.0 times that of a smooth channel) and found relatively independent on the Reynolds number and channel height. Neither the heat transfer coefficient distribution nor the friction factor exhibited a detectable effect of the channel height within the studied relative height range (i.e.$0.37<H/D<1.49$).

Syred et.al. [3] studied the effect of surface curvature on heat transfer and hydrodynamics within a single hemispherical dimple. Heat transfer behavior in a “curved” dimple is identical to that in a flat dimple. Turbulent fluctuations in a concavely curved dimple exceed those occurring in a convex dimple. The surface curvature considerably influenced the dimple heat transfer rate. It enhances heat transfer rate in a “concave” and reduces in the “convex” one; however, the more remarkable effect occurred in a concavely curved dimple. The correction factors describing the effect of curvature on average heat transfer in a “curved” dimple was obtained as a result of experimental study.

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 Reynolds number varying from 1250 to 61500. These includes flow visualizations show vertical flow 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 vertical fluid that from near the downstream of each dimple. He results were also showed that as the ratio of inlet to wall temperature decreases, the coolest part of the test surface which corresponds the highest value of baseline Nusselt number ratio (Nu/Nuo) intensifies and extends farther away from downstream rims of the dimples. In each case of Reynolds number, low Nusselt number corresponds with upstream portions of the dimple cavities, and the highest ratios were from locations near the rims or near the downstream edges of the dimples. For Reynolds number from 10,200 to 13,800 ratios of dimple surface friction factor to smooth surface friction factors reported were 1.5 to 1.55. Such values along with heat transfer augmentations at the same Reynolds number provide further evidence of the feasibility of dimpled passages for internal turbine airfoil cooling.

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 600to 11,000 and H/D ratio varying as 0.20, 0.25, 0.50and1.00 .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 behaviour on deep dimpled surface. Experimental results were presented for a dimpled test surface placed on one wall of a channel. Reynolds number varied from 12,000to70,000 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 numbers include local Nusselt numbers and globally averaged Nusselt numbers. Result showed that at all Reynolds considered, local Nusselt number augmentation increases as the /d ratio increases from 0.2 to 0.3. Here two local maxima are evident in local Nusselt number ratios when δ/D=0.3, compared to a single maximum value when δ/D=0.2.

Beves et. al. [7] 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. The interaction of the fluid ejecting from the rear of the cavity and the boundary layer on the surface caused an oscillation within the boundary layer even at low Reynolds numbers. They also reported the absence of vortex shedding, possibly due to a two dimensional nature of a experiment which can be observed using three- dimensional cavity.

Yaroslav Chudnovsky[8] investigated vortex heat transfer enhancement of heat transfer by a system of 3D surface cavities (dimples) having specific geometry, dimensions and mutual orientation Each dimple acts as a “vortex generator” which provides an intensive and stable heat and mass transfer between the dimpled surface and gaseous heating media. The obtained method includes mixing of the fluid by creating artificial surface such as dimples or other techniques.

IV. PRINCIPLE OF VORTEX HEAT TRANSFER ENHANCEMENT

The vortex formed inside the dimple causes the scrubbing action of flowing fluid inside the dimple. Vortex heat transfer enhancement is the enhancement of heat transfer by three dimensional surface cavities (dimples) having specific geometry, dimensions and mutual orientation. Each dimple acts as a vortex generator which provides an intensive and stable heat and mass transfer between the dimpled surface and gaseous heating/cooling media. Taking advantages of Vortex heat transfer enhancement as:

- Higher heat transfer coefficient
- Negligible pressure drop penalty
- Potential for fouling reduction
- Simplicity in design and fabrication
- Compactness and/or lower cost

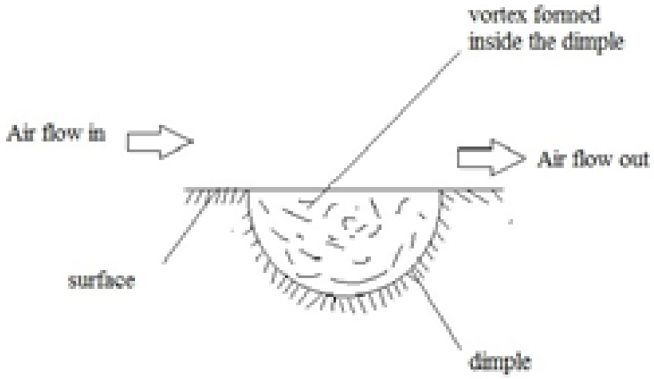

Fig.2: VHTE Mechanism

V. EXPERIMENTAL SETUP

Experimental setup consist of blower, flow control valve, orifice meter, U-tube manometer, test section, test plates (Aluminum). It also includes arrangement of thermocouples (K type), heater plate and temperature indicator. The detail view of Experimental setup is shown in fig below:

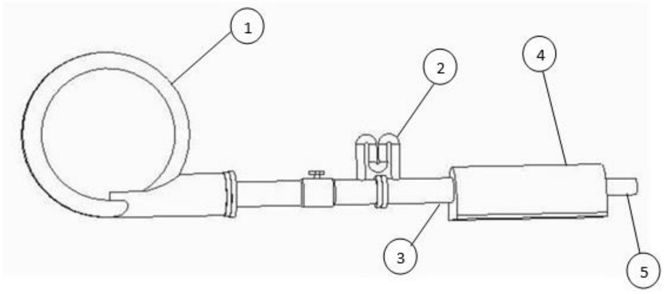

Fig.3: Schematic of Experimental set-up Arrangement

1-Blower; 2-Orifice meter; 3-Entrance section; 4-Test section; 5-Exit section

VI. TEST PLATES

Plain test plate is of aluminum sheet of thickness 6mm & having dimensions as 400*72 mm.

Fig. 4: Original test plate without dimples

The dimples produced on the test plate of 6 mm diameter & 3mm depth. For rectangular inline dimple arrangement, total 34 rows are employed in the stream wise direction& 6 rows are in span wise direction. Test section coordinate system employed for the study. Note that Y coordinate is normal to the test surface & x coordinate system is along flow direction.

The spacing in between center of two dimples in X&Y direction is of 11 mm as shown in fig.

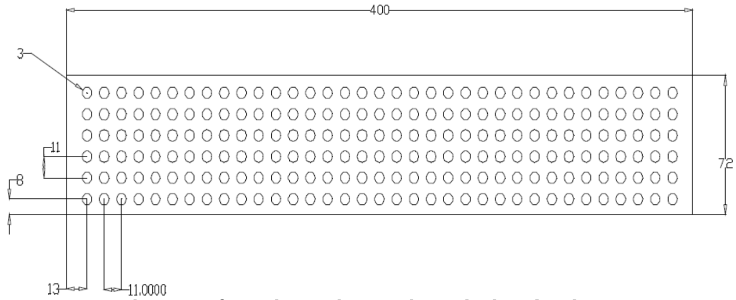

Fig. 5: Schematic of test plate with 204 spherical inline dimples

Fig.6: original dimpled plate with simple arrangement

For staggered arrangement also there are 34 rows in X-direction & 6 rows in span wise direction as shown in fig. The tool used for dimple making on a test plate is a punch tool is made up of carbide material & size of tool is 6mm in diameter.

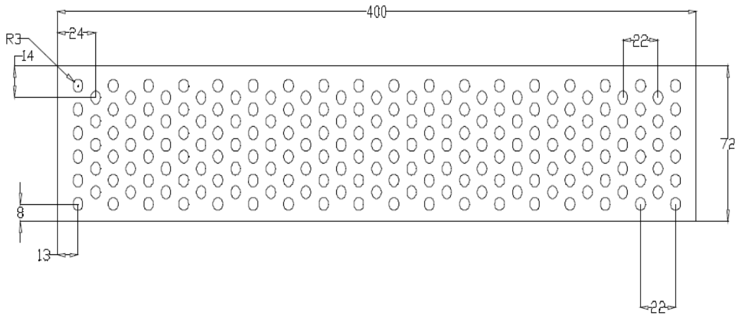

Fig.7: Schematic of test plate with 187 spherical staggered dimples

Fig.8: original dimpled plate with staggered arrangement

VII. ASSEMBLY PROCEDURE

- Attach the blower to the test section by pipe having orifice arrangement.
- Connect the U-tube manometer across the orifice meter to measure manometric height differences.
- At the bottom of test section layer of glass wool is kept to avoid heat losses. Above gloss wool, heater & test plate which are bolted together are placed.
- To measure temperature at different location on test surface one end of thermocouple is screwed to test surface & other is to temperature indicator display.
- Thermocouples are also attached at the inlet & outlet for measuring inlet air temperature & outlet air temperature of test section.
- Bottom section of test section is made up of wood and can be easily screwed and unscrewed for replacement of test plates to be tested.

VIII. TESTING PROCEDURE

- Switch on electric supply to heater and allow rise in temperature of plate up to 50-60°C.
- Now start blower.
- Measure height level difference in U-tube manometer.
- Wait for steady state conditions of voltage and current supplied to heater and measure all the temperature at different locations of test surface (T2, T3, T4, T5) and also measure inlet air temperature ($Tai=T_1$) and outlet air temperature ($Tao=T_5$) at the test section by rotating knob on temperature indicator display.
- Stop heater supply. Remove bottom plate of test section & let the test plate be cool.
- After 15 to 20 minutes remove the test plate from test section, put fresh glass wool and repeat the procedure for another plate at same voltage and current supply to heater.

IX. CALCULATIONS AND FORMULAE'S TO BE USED

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

Ts = average surface temperature

$$Ts = (T2 + T3 + T4 + T5)/4$$

Ta = mean bulk temperature

$$Ta = (Tao + Tai)/2$$

H = Manometer head in terms of air column

$$H = X * [(s_w/s_a) - 1]$$

X= Manometer head in terms of water column

s_w/s_a = Specific gravity ratio

$$Q = Cd * \{(a_1 * a_2 * \sqrt{(2gh)}) / (a_1^2 - a_2^2)\}$$

Q- Discharge of fluid flowing through pipe (m³/sec)

Cd - coefficient of discharge for orifice meter

a_1 & a_2 - cross sectional area of pipe & of orifice plate respectively at junction (m²)

ρ_{air} = density of air (kg/ m³)

q = heat transfer rate = heat carried by air through test section (Watts)

$$q = ma * Cpa * (Tao - Tai)$$

Cpa = Specific Heat of air

ma = mass flow rate of air(Kg/sec) = Q * ρ_{air}

$$q = h * A * (Ts - Ta)$$

h = heat transfer coefficient (W/m²K)

A = heat transfer area (m²)

Reynolds number is calculated as

$$Re = \rho * v * d / \mu$$

Nusselt number based on test plate length is

$$Nu = h / K$$

X. CONCLUSION & SCOPE FOR THE FUTURE WORK

The heat transfer rate (q), Nusselt No (Nu) convective heat transfer coefficient (h) and Reynolds No (Re) for the cases of Plain Plate, Plate with Inline Dimples and Staggered Dimples shall be compared.

It is expected that heat transfer rate of dimpled plates shall be more as compared to a plain flat plate. This investigation will present experimental and numerical results for the heat transfer characteristics of a flat plate for various airflow conditions using two different types of dimples. In this work, experiments will be carried out for limited and low range of Reynolds number. This could be extended for the higher range of Reynolds number and using larger dimensions of the test surface. Different shapes like rectangular or triangular shapes of dimples can be used instead of circular dimples on the test surface. Performance of combination of above mentioned shaped dimples can be experimented and compared. Test plate material can be changed such as copper (which is very good conductor of heat) and performance is compared with different material combinations.

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REFERENCES

[1] Kuethe A. M., 1971, ''Boundary Layer Control of Flow Separation and Heat Exchange,'' US Patent No. 3,578,264.
[2] Moon H.K., T. O'Connell, Glezer B., "Channel Height Effect on Heat Transfer and Friction in a Dimpled Passage", ASME J. Gas Turbine and Power, 122, April 2000, pp.307-313.
[3] Syred N., Khalatov A., Kozlov A., Shchukin A., Agachev R., 2001, "Effect of Surface Curvature on Heat Transfer and Hydrodynamics within a Single Hemispherical Dimple" ASME, Journal of Turbomachinery, October 2001, Vol. 123, pp. 609-613.
[4] 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 Turbomachinery, 123(1), January 2001, pp.115-123.
[5] 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 heat Mass Transfer 45, 2011-2020, 2002.
[6] Oliveira M. M., Burgess N. K., Ligrani P. M., "Nusselt Number Behavior on Deep Dimpled Surfaces Within a Channel", ASME Journal of Heat Transfer, 125, August 2003, pp.11-18.
[7] Beves C.C., Barber T.J., Leonardi E., **"**An Investigation of Flow over a Two-Dimensional Circular Cavity", 15th Australasian Fluid Mechanics Conference, The University of Sydney, Sydney, Australia 13-17 December 2004.
[8] Yaroslav Chudnovsky 2004 AI ChESpring Technical Meeting "vortex heat transfer enhancement for chemical industry fired heaters