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Experimental Investigation of Heat Transfer Enhancement over the Dimple Surface under Forced Convection

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***Abstract*— An experimental study has been conducted to study heat transfer over the dimpled plates under forced convection. The Reynolds number is varied in the range of 10000 to 30000 based on hydraulic diameter. The dimpled depth is varied from 0.2 to 0.4 keeping 0.1 frequency and keeping the dimple density constant in both the arrangements. The dimple print diameter is kept as 10mm where dimple depths are 2mm,3mm,4mm.A constant heat of 0.5 amperes was given. It is noticed that, heat transfer is augmented for a depth of 0..3 in both the arrangements.0.4 depth plate shown very least augmentation. The maximum thermal performance is for staggered arrangement and for 0.3 depth. The maximum thermal performance for inline arrangement is lowest value of thermal performance of staggered plate.**

Index Terms*— *Dimple plate, Heat transfer augmentation, forced convection.

I. 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 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. Rib turbulators, an array of pin fins, and dimples have been employed for this purpose. An investigation was conducted to determine whether dimples on a heat sink fin can increase heat transfer for laminar airflows. This was accomplished by performing an experimental and numerical investigation using two different types of dimples [1] 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 passage because they produce substantial heat transfer augmentation.[5] Heat transfer enhancement over surface results from the depression forming recesses rather than projections. Generically, such features are known as dimples, and may be formed in an infinite variation of geometries which results in various heats transfer and friction characteristics [1].

U S Gawai, Mathew V K, Murtuza S D[9] studied experimental set up for enhancement of the forced convection heat transfer over the dimpled surface and flow structure analysis within a dimple. Pankaj N. Shrirao,..[2]studied on the mean Nusselt number, friction factor and thermal enhancement factor characteristics in a circular tube with different types of internal threads of 120 mm pitch under uniform wall heat flux boundary conditions. In the experiments, measured data are taken at Reynolds number in range of 7,000 to 14,000 with air as the test fluid. The experiments were conducted on circular tube with three different types of internal threads.S Suresh..[3] have made an experimental investigation on the convective heat transfer and friction factor characteristics in the plain and helically dimpled tube under turbulent flow with constant heat flux using CuO/water nanofluid as working fluid. The effects of the dimples and nanofluid on the Nusselt number and the friction factor are determined in a circular tube with a fully developed turbulent flow for the Reynolds number in the range between 2500 and 6000. The height of the dimple/protrusion was 0.6mm. Friction and compound heat transfer behaviors in a dimpled tube fitted with a twisted tape swirl generator are investigated experimentally using air as working fluid. The effects of the pitch and twist ratio on the average heat transfer coefficient and the pressure loss are determined in a circular tube with the fully developed flow for the Reynolds number in the range of 12,000 to 44,000.[6].

Sandeep kore[4]have made An experimental investigation has been carried out to study heat transfer and friction coefficient by dimpled Surface in a channel with standard asoect ratio of 4:1.Sang Dong Hwang[7] investigated heat transfer characteristics on various dimple/protrusion patterned walls along with a straight and rectangular test channel. The dimple/protrusion arrays were positioned on one side of the wall (single) or on two sides of the wall (double) in each test case.

II. EXPERIMENTAL INVESTIGATION

A. Experimental setup:

In this work, a rectangular box is fabricated in which a strip plate heater is attached to plates and it is insulated with wooden glass wool.The heat input is 0.5 amperes constant.Blower is connected to inlet side to carry the forced air parell to surface of dimpled plates and a water manometer is connected across a venturimeter to indicate the pressure differential in terms of water column difference.

Next to the venturimeter a control valve is connected to control the flow of air.The power socket through dimmerstat is connected to heater.Caliberated thermocouple wires are attached at each dimples to note down the temperatures and thermocouple wires are also connected to inlet and outlet sides of the heat exchange module.Digital temperature indicator is used to get temperature readings.

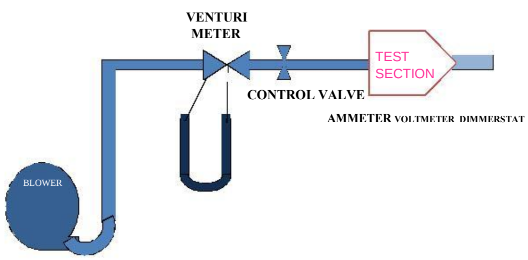

Fig 1. Block diagram of experimental setup

Fig 2. Photographic view of Test plates

III DATA REDUCTION

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

TABLE I: PERFORMANCE PARAMETERS

Parameter	Values
Dimple depth to print diameter	0.2,0.3,0.4
Reynolds Number	10000 to 30000
Heat Input	0.5 amp

The discharge of air is calculated from the following relation:

$$Q = Cd * a1 * a2 \frac{\sqrt{2gH}}{\sqrt{(a1^2 - a2^2)}} \quad m^3/s \qquad (1)$$

Where,

Cd=Coefficient of discharge for venturimeter ,
a_1 and a_2= Cross sectional areas of inlet and throat of venturimeter, H=Manometric difference

The useful heat gain of the air is calculated as:

$$q = ma * Cpa * (Tao - Tai) \text{ watts} \qquad (2)$$

Where,

m_a=mass flow rate of air, C_p=specific heat of air,
Ta_o and Ta_i= Fluid temperature at outlet and inlet of the duct

Heat transfer coefficient is calculated by using

$$h = \frac{q}{A * \Delta T} \quad w/m^2K \qquad (3)$$

Nusselt number is calculated by $N_u = \frac{(h*L)}{k}$

IV. RESULTS AND DISCUSSIONS

A. Effect of Reynolds number

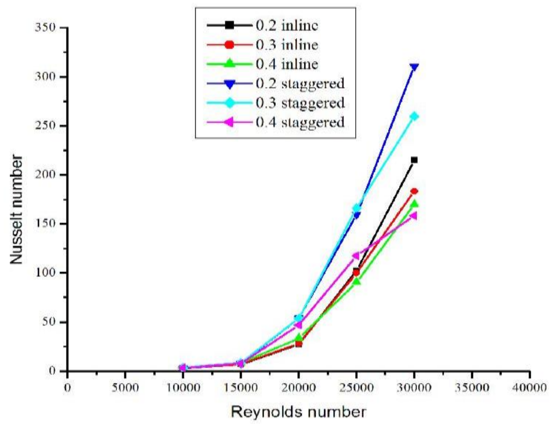

Fig 3. Effect of Reynolds number

From the above graph, it is observed that nusselt number increases gradually from Reynolds number 10000 to 20000 as no any layers are formed over the surfaces and also there is no sufficient air available for formation of vortex pair over the dimples, but as Reynolds number increases from 20000 to 25000 the rise in nusselt number is very rapid for staggered plate as more number of secondary flow layer are formed which forms the vortices at the dimple causing more heat transfer except for the 0.4 depth plate because of formation of insulating packets inside the geometry of dimple. The air as poor conductor of heat prevents the rate of heat transfer. Inline plates are having lesser nusselt number than staggered as there are effective formations of vortex pair over the plates and no secondary flow development. The highest nusselt number is observed for 0.2 staggered plate as the vortices are not diminished at larger nusselt number but 0.3 staggered plate looses the secondary flow layers at higher Reynolds number and least is for 0.4 staggered plate as the air entrapped in the dimples.

B. Effect of Dimple depth

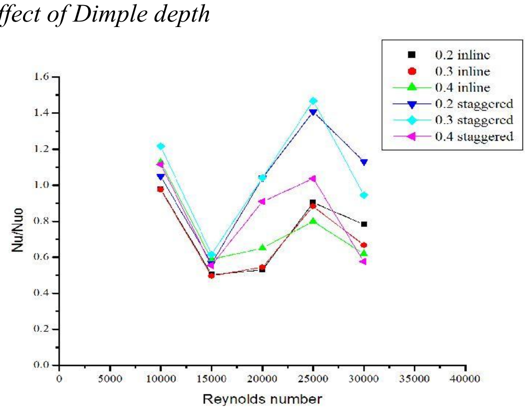

Fig 4. Graph of Effect of Dimple depth

From the above graph, it is observed that for all dimple depths the Nu/Nu_o ratio decreases for Reynolds number up to 15000 because of formation of recirculation zone and shear layer detachment and then increases up to 25000 Reynolds number as more number of vortices are formed and due to which the rate of convective heat transfer is more but from Reynolds number 25000 to 30000 the ratio decreases due to diminishing of vortices and effect of secondary flows and shear layers are detached from the surface of the test plates. The ratio is highest for 0.3 staggered plate compared to 0.2 staggered plate and same case for inline plates too because of formation of stronger vortices with more pronounced shear layer reattachment. It’s also due to ejection and spraying of air (local jetting) from the 0.3 staggered plate. In the comparison of both the arrangements the staggered plate is having higher nusselt number ratio than the highest ratio of inline plate arrangement. The behavior of both depths is almost same at Reynolds number 15000 to 25000 but the ratio of staggered plates are more. The Nu/Nu_o ratio is lower for 0.4 depth because of larger regions of stronger re-circulating flow.These are believed to trap fluid which then acts as a partially insulting pocket to decrease local nusselt numbers.

\C. Effect of dimple depth and Reynolds number on Thermal performance:

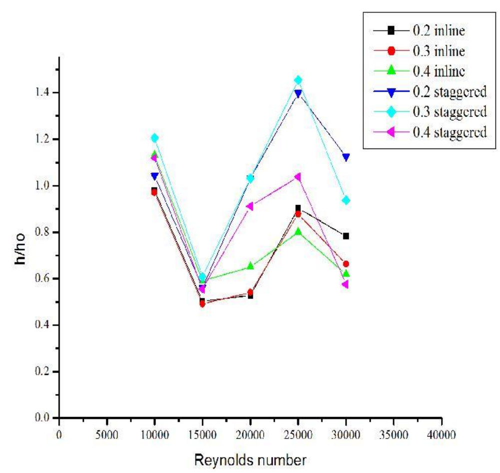

Fig 5. Effect of Thermal performance

From the above graph, it is seen that thermal performance for staggered arrangement is more than inline plate because of more turbulence and strong vortex pairs in staggered plates. The efficiency decreases above Reynolds number of 25000 as all the vortex pairs and shear layers detached. The higher h/h_o ratio of inline plate is lower value of staggered plate.0.3 staggered arrangement and least is for 0.4 depth because of entrapping of fluid inside the dimple behaving as the partial insulating pocket as air is poor conductor of heat and only reaches lower ratio than 0.2 plate because strength of re-circulating zones are more at 0.3 plate than 0.2 plate. In inline case also the behavior is same.

V. Conclusion

From this experimental study, keeping the dimple density same on inline and staggered arrangement, the following conclusion were made.

a .Heat transfer rate increases for staggered arrangement than the inline arrangement.

b. Heat transfer rate is more for 0.3 depth than 0.2 depth and it is least for 0.4 depth of staggered arrangement.

c. The dimples plates are much useful for Reynolds number in the range of 15000 to 25000.

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