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Experimental investigation of Heat Transfer Enhancement from Dimple and PIN-FIN

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ABSTRACT

Heat transfer enhancement is investigated for six sets of dimpled fin channel. The Reynolds number varying from 5000 to 20000 may be used in order to determine the effect of arrangement of dimpled fin and its aspect ratio on heat transfer enhancement. The channel is of rectangular shape with aspect ratio 4:1[1]. The constant heat of 125 watt is supplied at the bottom of test plate. The dimple and pin-fin diameter as 10 mm is kept constant throughout the experimentation. The dimple print diameter to depth ratio is kept 0.3[9]. The pin-fin diameter to height ratio is 1:1. The heat transfer, friction factor and thermal performance data is compared with the smooth flat plate data under the same geometric and flow conditions.
Due to the presence of the dimples in the pin fin arrays, extra strong vortex flows are generated near the pin fin downstream to main flow region which increase the turbulent mixing there and enhance the heat transfer rates. Due to presence of pin fin in the path turbulence is created.

Keywords: **dimple surface and pin fin, heat transfer enhancement, forced convection, channel flow, vortex generator, heat transfer augmentation.**

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I. INTRODUCTION

Energy is the one of the basic needs of human being. As world is growing fast more and more energy is required to satisfy the need. In 80th century more effort is concentrated to provide energy. But due depleting sources of energy more and more focus is to save energy. For electric energy production there are many sources available but more than 60 to 70% of energy is produced from heat energy. So study area is concentrated on heat energy saving technique.

There are so many ways by which heat enhancement is done, including active and passive technique. The active technique requires external source for heat enhancement while passive technique does not require any external source. It works on physical variation in shape or introductions of various materials into it. [13]
The major heat enhancement technique includes dimple and pin fin. The

NOMENCLATURE:-

A	heat transfer surface area, m2
h	heat transfer coefficient, W/m2 K
k	thermal conductivity, W/m K
L	length, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandlt number
Sx	Pitch of dimples along X axis, m
Sy	Pitch of dimples along Y axis, m
Q	heat transfer rate, W
Re	Reynolds number
Ts	Temperature, °C
v	flow velocity, m/s
W	pumping power, W
Δp	pressure drop, N/m2
d	diameter of dimples, m

Huang et al.[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) in each test case. Authors investigated that at high Reynolds numbers, the heat transfer pattern on the protrusion surface is 'pea-shaped' and upon decreasing the Reynolds number, and the pattern becomes circular. The authors studied heat transfer characteristics for four sets of dimpled fin channels with Reynolds use of dimple produces vortices which create more and more turbulence .Due to turbulence proper mixing of fluid is done and due to this fluid carry more heat with it. Use of dimple will not affect drop in pressure very much. On the other hand the pin fin increases surface area causes increase in heat transfer enhancement. Since pin fins are protruding inside the flow produce more pressure drop.
number (Re) ranging from 1500 to 11,000. These dimpled fin channels share the identical rectangular section of a channel aspect ratio (AR) of 6 with three different L/d of 8.9, 6.2 and 3.5. Johann Turnow et al. [4], studied vortex structures and heat transfer enhancement mechanism in a turbulent flow over a staggered dimple array in a narrow channel using Large Eddy Simulation (LES). Laser Doppler Velocimetry (LDV) which measures velocity of flow and pressure measurements for ReD = 10000 and ReD = 20000. Simulations are validated by comparison with experimental data obtained for smooth and dimpled channels., Yu Rao, [8] investigated the 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. N. Sahiti et.al.[12], studied both effective surface enhancement elements and the optimal flow arrangement during the experimental investigation of the pin fin heat exchanger.
The objective of this research work is to study the effect of dimpled surface and dimpled pin fin on heat transfer and fluid flow.

II. EXPERIMENTAL SET UP

An experimental set-up for the study of dimpled pin on heat transfer enhancement is as shown in Fig. 1. The set-up consists of rectangular duct with aspect ratio of 4:1. The test apparatus is an open air flow loop that consists of a centrifugal blower, flow control valve, orifice meter (for flow measurement), an entrance section, flow straightner, the test section, and an exit section.

The duct is of size (100 mm x 25 mm).The air is drawn into the test channel by the 2 Hp constant speed blower; the blower is of suction type to avoid pulsating flow. The air mass flow rate is measured by the orifice flow meter. The first section is entrance section from where the air enters in the channel, which leads to a rectangular cross section, 25 mm by 100 mm test channel. The inlet section has a length of 645 mm, which is made of 5 mm-acrylic sheet. The entrance section wall is provided with one thermocouple attachment to measure inlet air temperature. The acrylic will provide a good optical access to the flow and heat transfer in the channel. Immediately upstream and downstream the test plate, there are pressure taps installed in the bottom wall of the test channel for attachment of micro-manometer to measure the pressure drop across the pin fin channel. The aluminium test plate having dimension of 100mm×1000mm× 5mm is attached to test section from below with the help of wooden clamps. Careful fabrication can ensure that the plate is flush with the top air pass of the test channel. The test plate with pin fin/pin fin-dimple array mounted on the surface is as shown in Fig. 2. The aluminium fins are attached to plate with the help of aluminium glue. To supply constant heat flux a plate type heater is provided on the bottom of test plate. The plate heater was connected with an AC power supply with a precise and controllable power output, and the total heating power can be accurately measured. To avoid heat loss due to radiation and convection from the heat supplied by the heater, 5 mm thick asbestos sheet is attached from bottom side. To reduce the heat loss from the test plate to the environment, the test section is wrapped with a layer of silica wool insulation. The pressure drop across the test section is measured by a micro- manometer, with double reservoir (range = 0.002–5 mbar) filled with benzyl alcohol and water.

After test section there is mixing chamber. The mixing chamber (250 mm length) is used for proper mixing of fluid after test section. The mixing chamber is provided with three thermocouple holes to measure average outlet temperature. the mixed-mean temperatures of the air leaving the test section are measured respectively by using three calibrated Type-K thermocouples spread across the cross section with an immersion depth of about half of the channel height. The three thermocouples are inserted at different levels. One placed at centerline and other two placed above and bellow the centerline in order to let the outflow become well mixed. Averaged values of these three thermocouples are obtained as the outlet mean temperature.

Figure 2 shows the geometrical configurations of the pin fin and the pin fin-dimple plates used in the experiment. There is a 45 row inline array of pin fin/pin fin-dimples in the streamwise direction, and there are five pin fins/dimples each row in the spanwise direction. The pin fin geometrical configurations of the pin fin channel are the same with those of the pin fin-dimple plate. The pin fin diameter D =10 mm, the spanwise spacing is S = 10, the streamwise spacing is varied from X = 18, 20, 22 and the pin fin height-to-diameter ratio H/D = 1.0. The dimples are arranged in-between the pin fins. The dimple print diameter is also 10 mm, and the dimple depth is 3 mm (0.3 D), and the dimple diameter is 10 mm.

The dimple plates are manufactured on the aluminum plates, by CNC machining, for that form tool of required curvature is made. To measure the surface temperature of the test plate eighteen calibrated thermocouples are used which are routed in the plate by drilled passage. This gives accurate measurement of wall temperature at number of junctions.

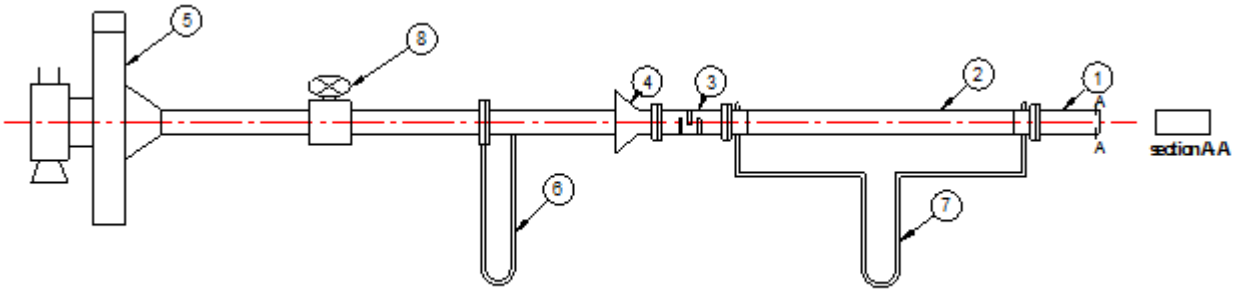

Fig. 1: Schematic diagram of experimental set up (HT lab-SAE)

(1) Entrance Section, 115 mm.

(2) Mixing Chamber, 250 mm

(3) Flow Straightner, 50 mm

(4) Micro-manometer

(5) Test Section, 1000 mm

(6) Centrifugal Blower, (2 Hp, 2990 rpm)

(7)U tube manometer

(8) Control valve

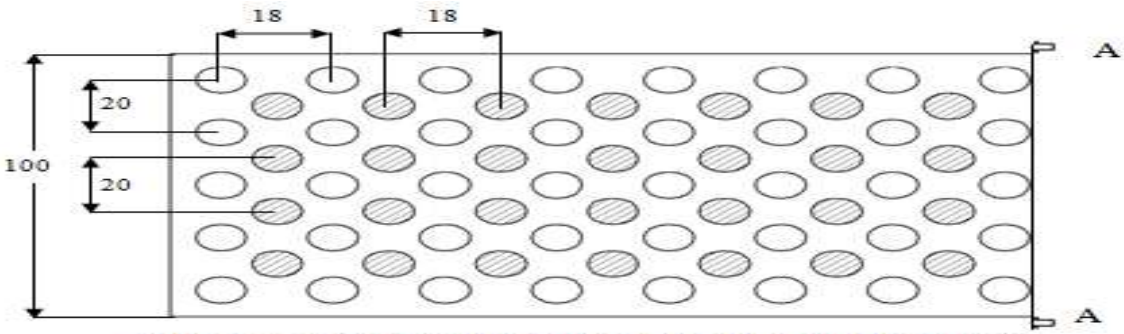

Fig. 2:-Array of pin-fin and dimple arrangement on plate.
(All dimensions are in mm)(Above figure shows eight arrays of dimples)

III.RESULTS AND DISCUSSION

In this research work, six plates with different dimple pin fin combination are used. Out of six plates three plates contain dimples and other three contain dimples with pin-fins.

The bulk mean temperature is calculated from following formula.

Tbm = (Tai + Tae) / 2

The mass flow rate of air is determined from the pressure drop across the orifice meter, using a following relation:

$(ma)^{\cdot}=\pi/4\times 〖d_0〗^2\times\rho_a\times C_d\times \sqrt{2gHa}/\sqrt{(1-\beta^4)}$

Where,

ρa =Density of air at mean temperature in kg/m3.
Cd = coefficient of discharge.
Ha = height of air column.
β = d/D = 0.7

For calculating Cd following formula is used.

$$C_d = f(\beta) + 91.71\beta^{2.5}Re^{-0.75} + \frac{0.09\beta^4}{1-\beta^4}F1 - 0.0337\beta^3 F2$$

Where,

F(β)=0.5959 + 0.0312×β2.1 -0.184 ×β8

The correlation factor F1 and F2 vary with tap position.

Corner taps F1=0 ,F2=0

For D=1/2D taps F1=0.43, F2=0.47

Flange Taps F2=1/D(in) , F1 = 1/D

D > 2.3 in, F1=0.433 2.0⩽ D ⩽2.3

Using the data obtained from experiments, the heat transfer, friction factor and the thermal performance characteristics of duct are analyzed in the following subsections.

3.2 Baseline Nusselt number

Baseline Nusselt numbers go with a smooth rectangular test section with smooth walls on all surfaces and no dimples. Baseline Nusselt numbers Nu_o are used to normalize values of measured Nusselt numbers on dimpled Surface. The baseline Nusselt numbers obtained from experiment are compared with Ditus-Boelter correlation which is given by

$$Nu_0 = 0.023 \times Re^{0.8} \times Pr^{0.4}$$

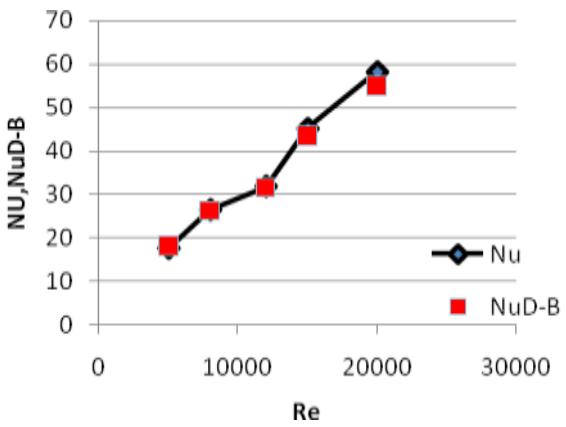

Fig. 3 Comparison of experimental Nusselt number with Ditus-Boelter equation

It is observed that for smooth plate as well as dimple plate, the Nusselt number increases with Reynolds number, however increases at faster rate in case of dimpled plate.

3.3 Effect of Reynolds Number on Nusselt number ratio.

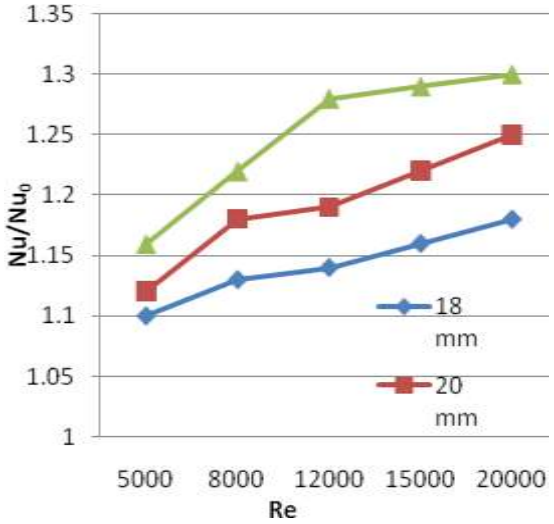

Fig. 4 comparison of Nusselt number ratio with Reynolds number for three dimple pitches.

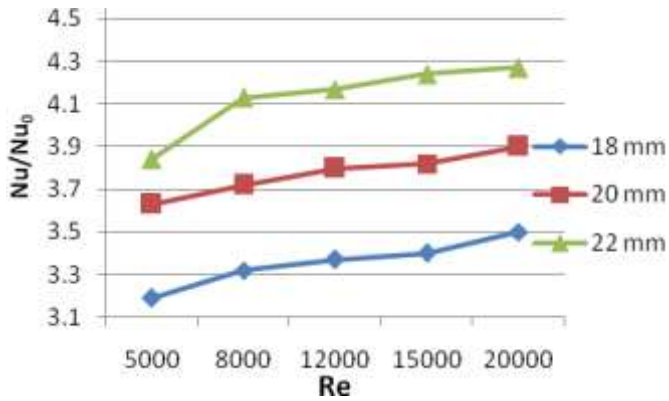

Fig. 5 Comparison of Nusselt number ratio with Reynolds number for three plates 18 mm, 20 mm and 22 mm with both dimple and pin fin.

3.4 Friction Factor

The friction factor indicates the degree to which the resistance is offered to the flow[14].For plate with dimple on it resistance for flow is less. But for the pin-fin plate it is more because pin-fins are extrusion and they directly reduce the flow area.

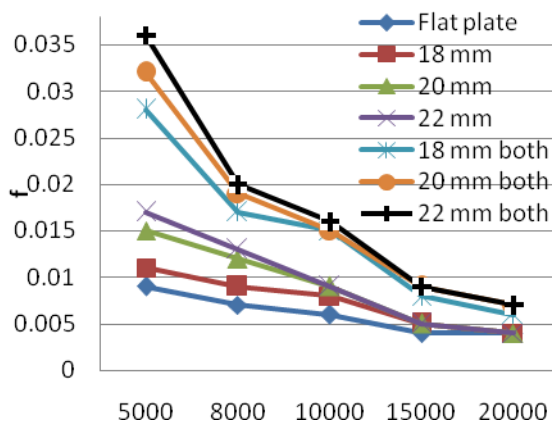

Fig. 6 Variation of friction factor with Reynolds number for all plates

3.5 Effect of Dimple Depth and Reynolds number on thermal performance

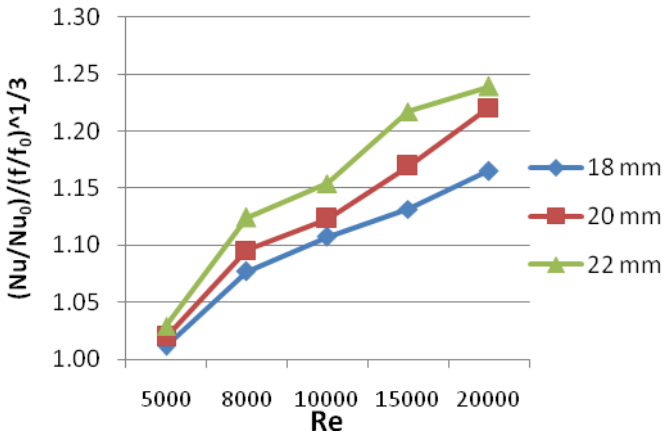

Fig. 7 Thermal performance Vs Reynolds number for dimple plates

Heat performance increases with increase in pitch for both dimple and pin fin because as pitch increases the pin-fin and dimples are well placed on plate surface and due to this heat transfer increases.

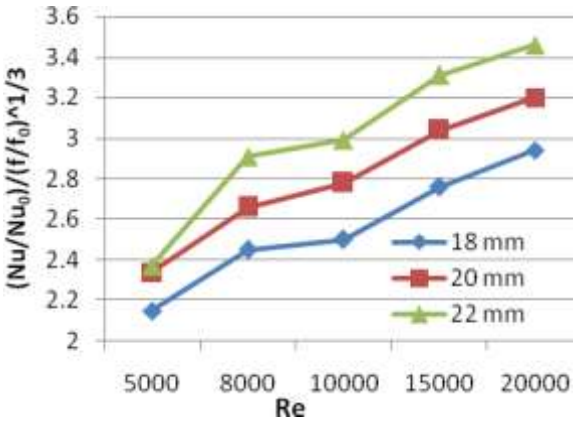

Fig. 8 Variation of heat performance with Reynolds number for dimpled fin plates

VI. SUMMARY AND DISCUSSION

For the experiments performed to study average heat transfer enhancement on different geometrical rectangular fins, the conclusions are as follows:

i. The heat transfer for protruded fin is more than flat plate. As flow pass over the larger area of fin surface and thus extract more heat from the fin.

ii. The maximum heat transfer takes place from the protrusion fin due to the turbulence created by the geometrical blockages.

iii. The heat transfer depends on the pitch, and density of the protrusion. It also depends on the flow pattern and protrusion position that is whether protrusion is inline or staggered.

iv. Heat transfer coefficients are relatively low on the leading edge of the dimple and are high on the trailing edge immediately downstream of the dimple.

v. It is observed that Nusselt number increases with Reynolds number for dimpled surface as well as for smooth channel, but rate of increase in Nusselt number is more for the dimpled surface as compared to smooth surface.

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