EXPERIMENTAL INVESTIGATIONS OF HEAT TRANSFER ENHANCEMENT FROM DIMPLED SURFACE IN A CHANNEL.

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An experimental investigation has been carried out to study heat transfer and friction coefficient by dimpled surface. The aspect ratio of rectangular channel is kept 4:1 and Reynolds number based on hydraulic diameter is varied from 10000 to 40000. The ratios of dimple depth to dimple print diameter is varied from 0.02 to 0.04 to provide information on the influences of dimple depth. The ratio of channel height to print diameter is 0.5. The heat transfer and friction factor data obtained is compared with the data obtained from smooth plate under similar geometric and flow conditions. It is observed that at all Reynolds number as depth increases from 0.2 to 0.3, the normalised Nusselt number and thermal performance increases and then after when depth increase from 0.3 to 0.4 normalised Nusselt number and thermal performance decreases.These are because of increase in strength and intensity of vortices and associated secondary flows ejected from the dimples.

Keywords: *Dimpled surface, channel flow, Vortex generator, heat transfer enhancement, forced convection*

1 INTRODUCTION

Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib turbulators, protrusions, pin fins, and dimples. These heat transfer enhancement techniques have practical application for internal cooling of turbine airfoils, combustion chamber liners and electronics cooling devices, biomedical devices and heat exchangers. Recently, dimples have drawn more attention because of the significant enhancement in heat transfer with a lower penalty in the increased pressure drop. Using dimpled surfaces in these situations requires knowledge of the effects of different dimple geometry characteristics; however, at present, the archival literature is deficient in providing such data for many important geometric parameters.

A number of heat transfer studies from Russia utilize dimples. These studies employ flows over flat walls with regular arrays of spherical pits [2], flows in annular passages with a staggered array of concave dimples on the interior cylindrical surface [3], flows in single hemispherical cavities [4, 5], flows in diffuser and convergent channels each with a single hemispherical cavity [6], and flows in a narrow channel with spherically shaped dimples placed in relative positions on two opposite surfaces [7]. Heat transfer augmentations as high as 150 percent, compared to smooth surfaces are reported sometimes with appreciable pressure losses [3]. Other recent data show that the enhancement of the overall heat transfer rate is about 2.5 times smooth surface values over a range of Reynolds numbers and pressure losses are about half the values produced by conventional rib turbulators. Moon et al. give data that show that improvements in heat transfer intensification and pressure losses remain at approximately constant levels for different Reynolds numbers and channel heights. Mahmood et al. [8] describe the mechanisms responsible for local and spatially averaged heat transfer augmentations on flat channel surfaces with an array of dimples on one wall for one channel height, equal to 50% of the dimple print diameter. Other investigations consider flow and heat transfer in single spherical cavities [9]; effects of dimples and protrusions on opposite channel walls [4, 5]; the effects of dimple depth on vortex structure and surface heat transfer [11]; the effects of deep dimples on local surface Nusselt number distributions [2]; the combined influences of aspect ratio, temperature ratio, Reynolds number and flow structure [3]; and the flow structure due to dimple depressions on a channel surface [10].

The present study is different from existing investigations because the effects of dimple depth δδ/D values ranging from 0.2 to 0.4 are investigated. The experimental results are given for a ratio of channel height to dimple print diameter 0.5 and Reynolds number based on hydraulic diameter 10,000 to 40,000. The data presented in this paper is average Nusselt number and friction factors.

2 EXPERIMENTAL INVESTIGATIONS

2.1 Experimental set up

An experimental set-up has been designed and fabricated to study the effect of dimpled surface on heat transfer and fluid flow characteristics in rectangular duct. A schematic diagram of the experimental set-up is shown in Figure1

The test apparatus is an open air flow loop that consists of a centrifugal blower (1), flow control valve (2), orifice meter (3) (for flow measurement), an entrance section (6), flow straightner, the test section (5), and an exit section (4). The duct is of size (100mm x 25mm) of aspect ratio 4. The test section is fabricated from M.S. sheet. The Test section is of length 100 mm (2.5Dh). The exit and entry lengths are 250 mm (6.25Dh) and 400 mm (10.40Dh) respectively. The exit section of 250mm is used after the test section in order to reduce the end effects. To provide uniform heat flux, cylindrical electric heater of size 10 mm x 12mm is used. The heaters are located below the dimpled surface. The test section is insulated from three sides with bakelite sheet of thickness 14mm. The mass flow rate of air is measured by a orifice meter and the Gate valve provided in the line for the flow control. For temperature measurement calibrated k-type thermocouples are used (24 SWG). 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.

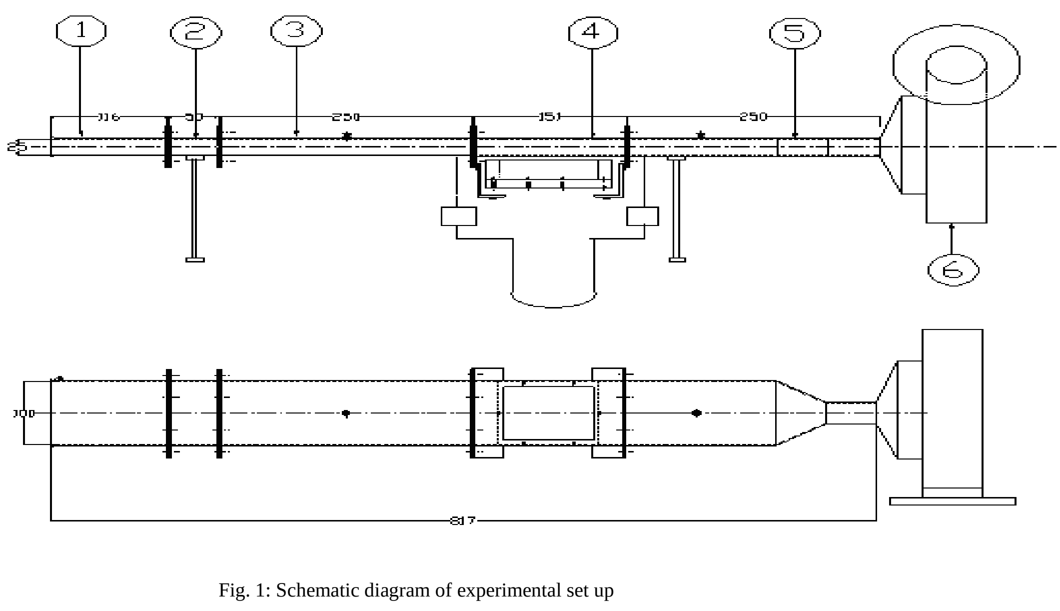

Fig. 1: Schematic diagram of experimental set up

(1) Entrance Section, 115 mm.

(2) Flow Straightener, 50 mm

(3) Settling Chamber, 250mm

(4) Test Section, 150mm

(5) Mixing Chamber, 250mm

(6) Centrifugal Blower, (1 Hp, 2990 rpm)

2.2 Experimental procedure

The test section is assembled in test bracket and checked for air leakage. The blower was switched on to let a predetermined rate of airflow through the duct. A constant heat flux is applied to the dimpled surface. The net heat flux and the average test surface to bulk mean air temperature difference was determined over a test section. Four values of flow rates were used for each set at same or fixed uniform heat flux. At each value of flow rate and the corresponding heat flux, system was allowed to attain a steady state before the temperature data were recorded. The pressure drops were measured when steady state is reached.

During experimentation the following parameters were measured:

i) Pressure difference across the orifice meter.

ii) Temperature of the heated surface and temperatures of air at inlet and outlet of the test section and

iii) Pressure drop across the test section.

2.2 Data reduction

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

Table 1 Performance parameters

Range of flow and dimple parameters	
Reynolds number based on hydraulic diameter $=Re_{Dh}$	10000 to 40000
Dimple print diameter=D	50mm
Dimple depth to print diameter ratio= δδ/D	0.2, 0.3, 0.4
Channel height to dimple print diameter ratio H/D	0.5
Channel aspect ratio	4

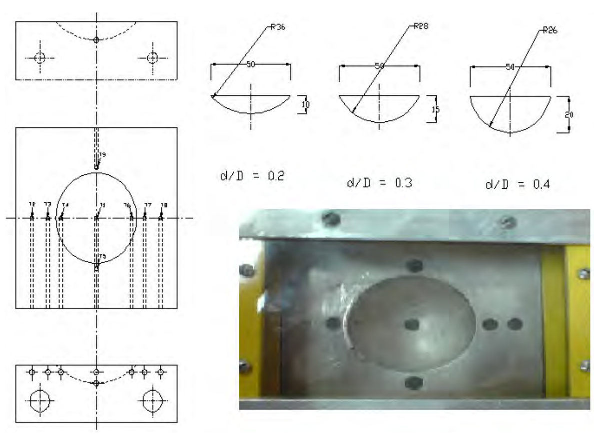

Fig. 2: Dimple Geometry

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

$$\dot{m} = C_D\ a_o x\ \frac{\rho_{air}\ X\ \sqrt{2 g H_a}}{\sqrt{1-\beta^4}} \qquad (1)$$

Where,

C_D = Coefficient of discharge for orifice
a_o = Cross sectional area of orifice
β = d/D, dimeter of pipe/ diameter of orifice
g = Acceleration due to gravity
Ha = Height of air column

The useful heat gain of the air is calculated as:

$$Q = \dot{m} C_p\ (T_{FO} - T_{FI}) \qquad (2)$$

Where,

T_{FO} = Fluid temperature at the exit of the duct (°C)

T_{FI} = Fluid temperature at the inlet of the duct (°C)
$\dot{m}$ = Mass flow rate of air
Cp = Specific heat of air
Q = Convective heat transfer to air

The heat transfer coefficient for the test section is:

$$h = \frac{Q}{A_s}\ X\ (T_{pm} - T_{Fm}) \qquad (3)$$

where,

T_{pm} is the average temperature of the test surface
T_{Fm} is the average temperature of air in the duct = $T_{FO} + T_{FI} / 2$
As is projected surface area of test surface
h is convective heat transfer coefficient

The Nusselt number as,

$$Nu = \frac{h\ x\ Dh}{K_{air}} \qquad (4)$$

where,

Nu is the average Nusselt number of the test surface
Dh is the hydralic diameter of the rectangular duct
Kair is the thermal conductivity of air

The friction factor was determined from measured vales of pressure drop across the test section using :

$$f = \Delta P\ D_h / 2\ \rho_{air}\ L\ V^2_{air} \qquad (5)$$

where,

f is friction factor of the test surface
ΔP is pressure drop across the test surface.
ρ_{air} is the density of air
L is the length of the test surface
V_{air} is the velocity of the air through rectangular duct

The thermo-physical properties of air used in the calculation of heat transfer and friction parameter were taken from available standard data tables which corresponding to mean bulk air temperature. The effect of humidity has been neglected since the relative humidity values during experimentation were found to be low and variation was small, ranging between 20% and 32%.

3 Results and discussion

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

3.1 Baseline Nusselt number

Baseline Nusselt numbers are in a smooth rectangular test section with smooth walls on all surfaces and no dimples. Baseline Nusselt numbers Nuo 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

$$Nu_o = 0.023\, Re_{Dh}^{0.8}\, P_r^{0.4} \qquad (6)$$

Where,

- Baseline Nusselt number,

Re_{Dh} - Reynolds number based on hydraulic diameter,

Pr - Prandt number,

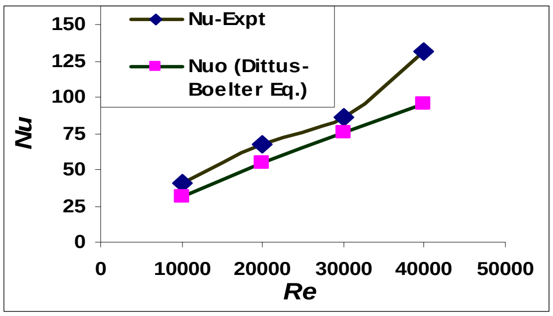

Fig.3. Comparison of experimental Nusselt number with Ditus-Boelter equation

The average absolute percentage deviation of the present experimental Nusselt number data is ±18.27 %

3.2 Effect of Reynolds Number

Nusselt number ratios are measured with dimple on one channel surface and heating on one channel surface, for different dimple depths, varying from δ/D = 0.2 to 0.4.The Baseline Nusselt numbers, used for normalization of the values presented, are obtained using the same thermal boundary conditions and heating arrangement as when dimples are used on the measurement surfaces. In addition, heat transfer coefficients and heat flux values (used to determine Nusselt numbers) are based on flat projected areas in both cases. It is observed that Nusselt number increases with Reynolds number for dimpled surface as well as for smooth channel, but rate of increase is more for the dimpled surface as compared to smooth surface. The maximum value of Nusselt number observed is 225 for a dimple depth of0.03 at Reynolds number 40,000. It is also observed that as Reynolds number increases above 30,000 the increase in Nusselt number is less.

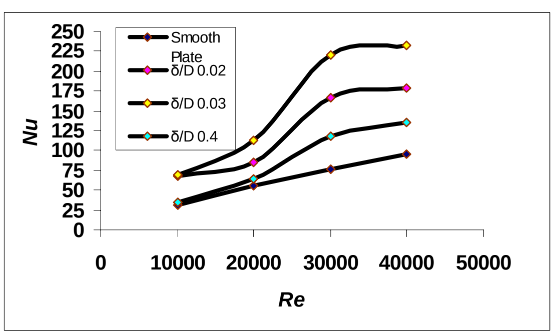

Fig.4. Variation of Nusselt number with different Dimple Depths.

3.3 Effects of Dimple Depth

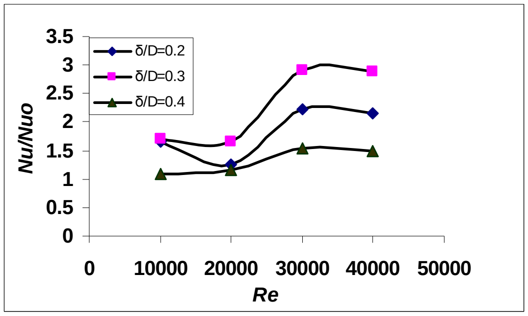

Fig. 5. Normalized Nusselt number ratio vs Reynolds number for different dimple Depths.

Figure 6 presents Nusselt number ratios for different dimple depths. In all dimple depths Nu/Nuo ratio decreases in first 25% of dimple due to formation of recirculation zone and shears layer dettachment and then increases in continuous over last 75 % of dimple. The Nu/Nuo ratios are higher at just down stream or at the trailing edge of the dimple. It is also observed that Nu/Nuo ratio is higher for dimple depth 0.3 and lowest for dimple depth of 0.4. These differences for different δ/D are due to shedding of vortex pairs with different strengths and to different shear layer formation, development, and reattachment. The Nu/Nuo ratio is higher for δ/D=0.3 because of formation of stronger vortices, with more pronounced shear layer reattachment. It is also because of the ejection and local jetting of flow from the dimple of δ/D=0.3 is stronger, which results in vortex pairs with stronger and more pronounced secondary flows. The Nu/Nuo ratio is lower for δ/D=0.4 because of larger regions of stronger re-circulating flow. These are believed to trap fluid which then acts as a partially insulating pocket to decrease local Nusselt numbers compared to values measured at dimple depth of δ/D=0.3.

3.4 Friction Factor

Figure 6 shows friction factor ratio of the present study for δ/D =0.2 to0.04. The value of friction factor ranges from 1 to 2.5 at Reynolds number 10000 and 1 to 2.2 at Reynolds number 40000. It is observed that, as Reynolds number increases friction factor decreases for all depths. Friction factor is higher for a dimple depth of δ/D =0.4, because of strong recirculation of the flow and lowest for δ/D =0.2. As depth increases vortices becomes stronger and stronger increase the friction factor.

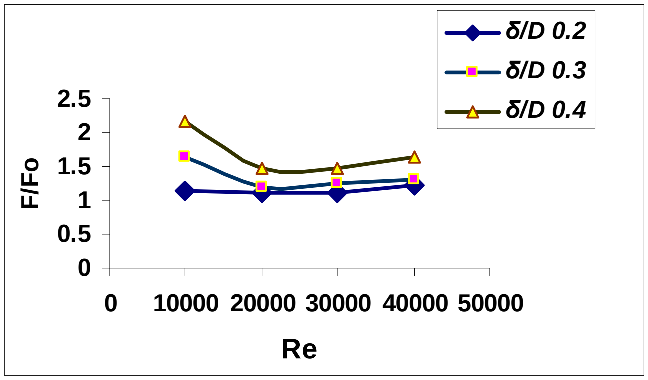

Fig.6. Friction factor vs Reynolds number for different dimple depths.

3.5 Effect of Dimple Depth and Reynolds number on thermal performance

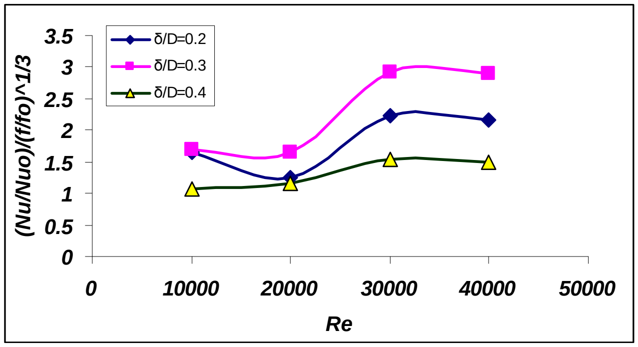

Fig. 7. Thermal performance vs Reynolds number for different depths

Averaged Nusselt numbers are determined by averaging local data over one complete period of dimple geometry. The baseline Nusselt numbers Nuo (used for normalization) are obtained at the same Reynolds numbers as the Nusselt numbers measured on the dimpled surfaces. For all Re_{Dh} examined, the data in Figure the data shows that averaged Nusselt number ratios increases as depth increases from0.2 to0.3 and then decreases for depth 0.4.Figures show averaged dimpled-channel thermal performance parameters,(Nu/Nuo / f / fo $^{1/3}$) as dependent on Reynolds number for δ/D=0.2,0.3,0.4 and H/D=0.5. The form of the (Nu/Nuo / f / fo$^{1/3}$) performance parameter provides a heat transfer augmentation quantity (Nu/Nuo) and a friction factor augmentation quantity (f / fo$^{1/3}$), where each is given for the same ratio of mass flux in an internal passage with augmentation devices to mass flux in an internal passage with smooth surfaces. The fig shows, Nu/Nuo / f / fo$^{1/3}$ performance parameter increases as depth increases from 0.2 to 0.3 and decreases when depth increases from 0.3 to 0.4.

4 Conclusions

An experimental study of the flow of air in a rectangular channel with dimpled surface, subjected to uniform heat flux boundary condition has been performed.

The effect of Reynolds number and dimple depth on the heat transfer coefficient and friction factor has been studied. Experimental results measured on dimpled test surfaces placed on one of the walls of channels with an aspect ratio of 8 are given for Reynolds numbers from 10,000 to 40,000. The ratio of channel height to dimple print diameter is 0.5, and the ratios of dimple depth to dimple print diameter δ /D are 0.2, 0.3, and 0.4 to provide information on the influences of dimple depth. The data for all three δδ /D values are all obtained in channels with the same dimple print diameter, same channel aspect ratio, same H/D ratio, and same type of thermal boundary condition. Results have been compared with those of a smooth channel under similar flow conditions to determine enhancement in heat transfer coefficient and friction factor.

Following conclusions have been drawn:

1) Heat transfer improvement with dimples seems to have a maximum value of approximately 2.88 and overall maximum thermal performance of about 2.63 for a depth of 0.3.
2) Heat transfer coefficients are (relatively) low on the leading edge of the dimple and high on the trailing edge and flat area immediately downstream of the dimple.
3) At all Reynolds numbers considered, average Nusselt number augmentations increase as the ratio of dimple depth to dimple print diameter δ /D increases from 0.2 to 0.3 and then decreases when depth increases from 0.3 to 0.4 (all other experimental and geometric parameters are held constant). This is because as depth increases from 0.2 to 0.3 the deeper dimples produce (i) increases in the strengths and intensity of vortices and associated secondary flows ejected from the dimples, as well as (ii) increases in the magnitudes of three-dimensional turbulence production and turbulence transport.
4) Decreasing Nusselt number ratios with increasing dimple depth is because of larger regions of

stronger re-circulating flow. These are believed to trap fluid which then acts as a partially insulating pocket to decrease Nusselt numbers.

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