Experimental Investigation of Heat Transfer Enhancement through Elliptical Dimples

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Abstract-*An experimental investigation was conducted to determine whether dimples on micro heat exchangers, Turbine blades, Electrical Cooling Systems can increase heat transfer for turbulent airflows. This was accomplished by performing experimental studies using setup which consist of divergent test channel in which Elliptical dimples are milled with aspect ratio (AR=1.5) arrangement (Inline, offline and Smooth) on one side of aluminum plates (Test Section) with a and relative depth c/D1=0.33. The main physical mechanisms causing the enhancement of heat transfer is the generation and amplification of sufficiently strong longitudinal vortices which are interacting with the thermal boundary layer. Since vortex induced heat transfer enhancement depends strongly on shape and position of Dimples, the subject of ongoing research is to find design strategies for device shape and placement optimization.*

For those configurations the heat transfer coefficient (h), Nusselt number (Nu), pressure drop (Dp), thermal performance (Tp) and Nusselt number ratio (Nu/Nu_s) were determined experimentally. For Elliptical dimples, heat transfer enhancements (relative to a flat plate) were observed for Reynolds number range from 5000 to 10000 (Reynolds number based on channel height). Also the results are validated analytically for Nusselt number and heat transfer coefficient for plain horizontal plate. Specifically, this investigation was conducted to determine whether the use of dimples can enhance heat transfer characteristics for Compact Heat Exchangers and turbine blades applications. With different types of arrangements such as flat, inline and offset and with variable parameter such as velocity, v = 5 m/sec, v = 10m/sec, v = 15m/sec for this conditions cases are studied and it has been found that the offset arrangement in divergent plate gives optimum solution as compared to other arrangements.

Keywords*-*Dimpled surface, Divergent channel flow, heat transfer enhancement, forced convection

I. INTRODUCTION

The various techniques are used to enhance the rate of heat transfer over surface of plate. It may be passive or active technique. The significant pressure drag produced by the rib or pin fin protrusion into the flow. Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib tabulators, protrusions, pin fins, and dimples. These heat transfer enhancement techniques have practical application for internal cooling of turbine aerofoils, combustion chamber liners and electronics cooling devices, biomedical devices and heat exchangers. The heat transfer can be increased by the following different Augmentation Techniques. They are broadly classified into three different categories:

1) Passive Techniques
2) Active Techniques
3) Compound Techniques.

1.1 Passive techniques:

These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power. 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. They promote higher heat transfer coefficients by disturbing or altering the existing flow behaviour except for extended surfaces. Heat transfer augmentation by these techniques can be achieved by using;

(i) Treated Surfaces: Such surfaces have a fine scale alteration to their finish or coating which may be continuous or discontinuous. They are primarily used for Boiling and condensing duties.
(ii) 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.

(iii) 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.
(iv) Displaced enhancement devices: These are the inserts that are used primarily in confined forced convection, and they improve energy transport indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct with bulk fluid from the core flow.
(v) 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.
(vi) 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.
(vii) Surface tension devices: These consist of wicking or grooved surfaces, which direct and improve the flow of liquid to boiling surfaces and from condensing surfaces.
(viii) Additives for liquids: These include the addition of solid particles, soluble trace additives and gas bubbles in single phase flows and trace additives which usually depress the surface tension of the liquid for boiling systems.
(ix) Additives for gases: These include liquid droplets or solid particles, which are introduced in single- phase gas flows either as dilute phase (gas-solid suspensions) or as dense phase (fluidized beds).

1.2 Active techniques

These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases.

In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer. Augmentation of heat transfer by this method can be achieved by:

(i) 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.
(ii) Surface vibration: They have been applied in single phase flows to obtain higher heat transfer coefficients.
(iii) Fluid vibration: These are primarily used in single phase flows and are considered to be perhaps the most practical type of vibration enhancement technique.
(iv) Electrostatic fields: It can be in the form of electric or magnetic fields or a combination of the two from dc or ac sources, which can be applied in heat exchange systems involving dielectric fluids. Depending on the application, it can also produce greater bulk mixing and induce forced convection or electromagnetic pumping to enhance heat transfer
(v) Injection: Such a technique is used in single phase flow and pertains to the method of injecting the same or a different fluid into the main bulk fluid either through a porous heat transfer interface or upstream of the heat transfer section.
(vi) Suction: It involves either vapour removal through a porous heated surface in nucleate or film boiling, or fluid withdrawal through a porous heated surface in single-phase flow.
(vii) Jet impingement: It involves the direction of heating or cooling fluid perpendicularly or obliquely to the heat transfer surface.

1.3 Compound techniques

A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger.

When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement. This technique involves complex design and hence has limited applications.

1.5 Use of Dimples for Heat Transfer Enhancement

There are number of heat transfer enhancement techniques are in use now; however, due to design and technological restrictions, designers of blade internal cooling systems still apply very few of these techniques. Among the basic technologies are: impingement cooling, pin fins, plain and broken ribs and their combinations. Excessive pressure loss can result in a more complex air supply system and higher cost or its application may be totally precluded. The surface heat transfer enhancement using dimples recently attracted interest due to its relatively low pressure-loss characteristics. Surface dimples are expected to promote turbulent mixing in the flow and enhance the heat transfer, as they behave as a vortex generator.

1.5.1 Flow mechanism over dimpled surface:

Dimples may be formed in an infinite variation of geometries which results in various heat transfer and friction characteristics. Heat Transfer enhancement using dimples based on the principle of scrubbing action of cooling fluid taking place inside the dimple and phenomenon of intensifying the delay of flow separation over the surface.

1.5.2 Mechanism of flow separation:

At sufficiently high velocities, the fluid stream detaches itself from the surface of the body, this is called as flow separation. the location of the separation point depends on several factors such as Reynolds number, the surface roughness & the level of fluctuations in the free stream. It is usually difficult to predict exactly where separation will occur. Dimpled surfaces are commonly known for their drag reduction characteristics in external flows over bodies. Heat Transfer enhancement using dimples based on the principle of scrubbing action of cooling fluid taking place inside the dimple and phenomenon of intensifying the delay of flow separation over the surface [6]. Dimple-surface can lower the coefficient of discharge (CD) to a fraction of its original value. This is because dimples cause a change in the critical Reynolds numbers (the Reynolds number at which a transition from laminar to turbulent flow begins in the boundary layer).

1.5.3 Vortex heat transfer enhancement technique:

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 VHTE, as a) higher heat transfer coefficient b) negligible pressure drop penalty c) potential fro fouling rate reduction d) simplicity in design and fabrication e) compactness and/or lower cost. This method is potentially used in heat transfer enhancement in convective passages for industrial boilers, process heaters and furnaces and heat exchangers variety for other industries like automotive (radiators, oil coolers etc.),heat treating (recuperates etc.), power electronics(convective coolers etc.), aerospace ,military, food processors etc.

II. LITERATURE SURVEY

Chang ShyyWoei, Jan Yih Jena , Chang ShuenFei[1] A detailed heat transfer measurement over a convex-dimpled surface of impinging jet-array with three eccentricities (E/H) between jet Centre and dimple-center is performed. These surface dimples considerably modify heat transfers from smooth-walled scenarios due to different impinging topologies for jet array with modified inter-jet reactions.

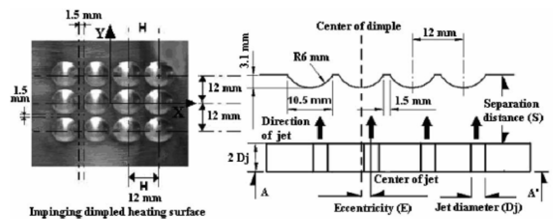

Fig No.1 Test Piece Geometry [1]

Heat transfer variations caused by adjusting jet Reynolds number (Re) and separation distance (S/Dj) over the ranges of $5000 < Re < 15,000$ and $0.5 < S/Dj<11$ with three eccentricities of $E/H = 0$, 1/4 and 1/2 are examined. A selection of experimental data illustrates the isolated and interactive influences of Re, S/Dj and E/H on local and spatially averaged heat transfers. In conformity with the experimentally revealed heat transfer physics, a regression-type analysis is performed to generate a set of heat transfer correlations, which permit the evaluations of spatially averaged Nusselt numbers over central jet region of dimpled impinging surface.

GongnanXie , Bengt Sundén[2]The heat transferred to the turbine blade is substantially increased as the turbine inlet temperature is increased. Improved cooling methods are therefore needed for the turbine blades to ensure a long durability and safe operation. The blade tip region is exposed to very hot gas flow, and suffers high local thermal loads due to the external tip leakage flow. A common way to cool the tip is to design serpentine passages with 180 turn under the blade tip-cap taking advantage of the three-dimensional turning effect and impingement. Increased internal convective cooling is therefore required to increase the blade tipLife time. In this paper, augmented heat transfer of a blade tip with internal hemispherical dimples has been investigated numerically. The computational models consist of two-pass channels with 180 turn and arrays of dimples depressed on the internal tip-cap.

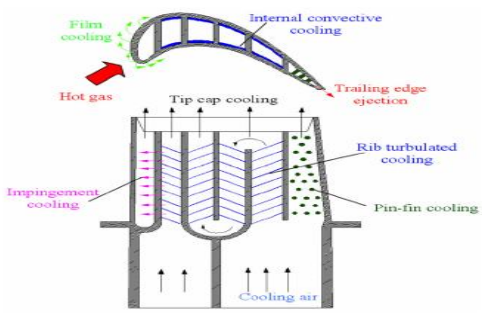

Fig. No. 2 Typical cooling techniques for a blade.[2]

Turbulent convective heat transfer between the fluid and dimples, and heat conduction within dimples and tip are simultaneously computed. The inlet Reynolds number is ranging from 100,000 to 600,000. Details of the 3D fluid flow and heat transfer over the tip-walls are presented. Comparisons of the overall performance of the models are presented.

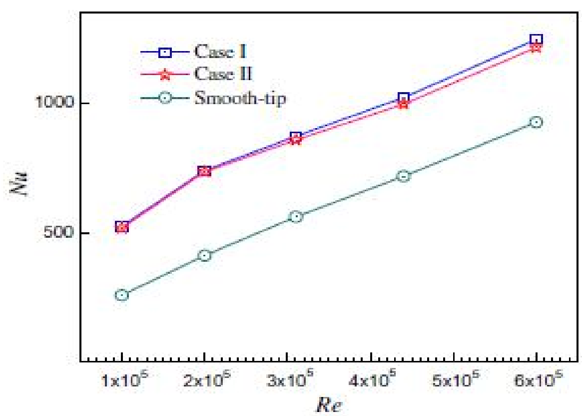

Fig. No.3 Heat Transfer Rate [2]

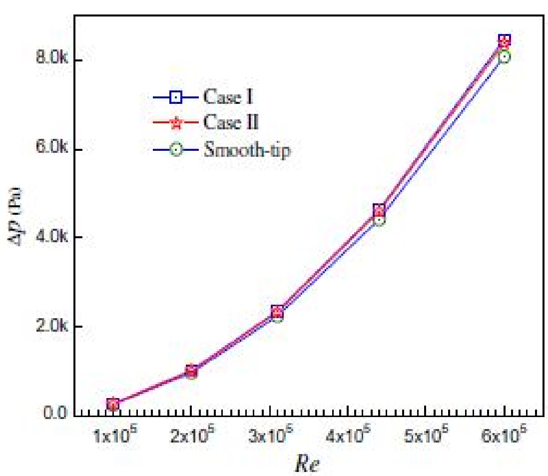

Fig. No.4 Pressure drop [2]

Itis found that due to the combination of turning impingement and dimple-induced advection flow, the heat transfer coefficient of the dimpled tip is up to two times higher than that of a smooth tip with less than 5% pressure drop penalty. It is suggested that the use of dimples is suitable for augmenting blade tip cooling to achieve an optimal balance between thermal and mechanical design requirements.

S.W. Chang, K.F. Chiang, T.C. Chou [3] Measurements of detailed Nusselt number (Nu) distributions and pressure drop coefficients (f) for four hexagonal ducts with smooth and dimpled walls are performed to comparatively examine the thermal performances of three sets of dimpled walls with concave–concave, convex–convex and concave–convex configurations at Reynolds numbers (Re) in the range of 900–30,000.

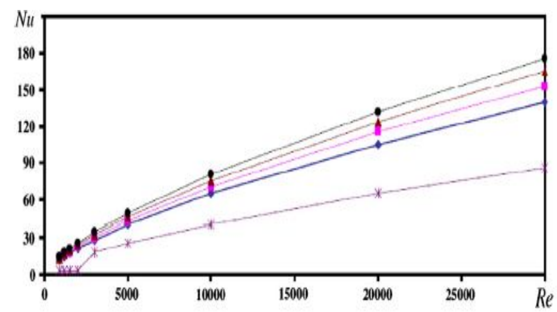

Fig.5 Heat Transfer Rate [3]

A set of selected experimental data illustrates the influences of dimple configuration and Re on the detailed Nu distributions, the area-averaged over developed flow region (Nu) and the pressure drop coefficients. Relative enhancements of Nu and f from the smooth-walled references (Nu1 and f1) along with the thermal performance factor (g) defined as (Nu/Nu1)/(f/f1)1/3 are examined. Nu and f correlations are individually obtained for each tested hexagonal duct using Re as the controlling parameter.

Yu Rao , Yamin Xu , Chaoyi Wan [4]An experimental and numerical study was conducted to investigate the flow friction and heat transfer performance in rectangular channels with staggered arrays of pin fin-dimple hybrid structures and pin fins in the Reynolds number range of 8200–54,000.

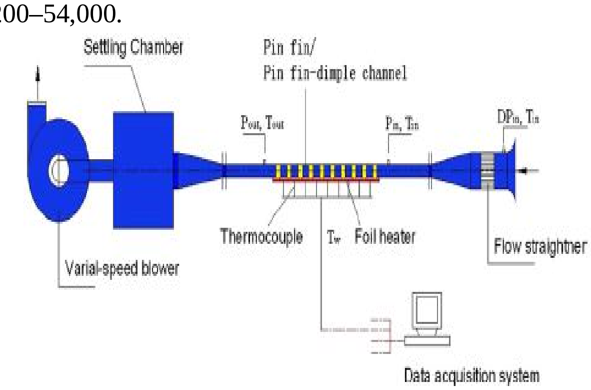

Fig. No.6Schematic diagram of the experimental system [4]

The study aims at improving the cooling design for the gas turbine components. The friction factor, Nusselt number and the overall thermal performance parameters of the pin fin-dimple and the pin fin channels have been obtained and compared with the experimental data of a smooth rectangular channel and previously published data of the pin fin channel.

The comparisons showed that, compared with the pin fin channel, the pin fin-dimple channel has further improved convective heat transfer performance by about 8.0% and whereas lowered flow friction by more than 18.0%.

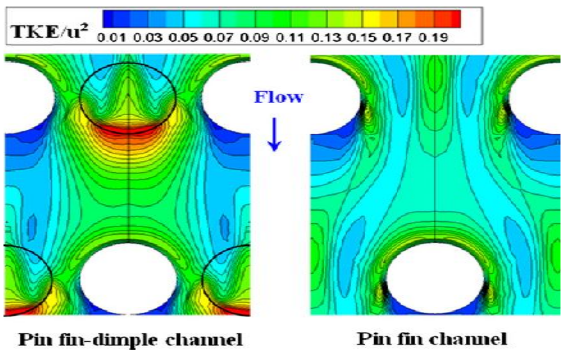

Fig No 7Comparison of the turbulent kinetic energy distribution in a plane with a distance of 0.5 mm away from the end wall between pin fin rows of 8 and 9 in the pin fin and pin fin-dimple channels at Re = 15,300.[4]

In addition, fully three-dimensional numerical computations have been done to investigate the physical details about the flow and heat transfer in the pin fin and pin fin-dimple channels. The computations showed that the dimples increase the near-wall turbulent mixing level by producing strong vortex flows, and therefore enhance the convective heat transfer in the channel. On the other hand, the dimples enlarge the minimum cross section area transversely between the pin fins, and therefore the pressure loss in the flow can be reduced in the pin fin-dimple channels.

Sang Dong Hwang , Hyun Goo Kwon , HyungHee Cho[5] In this study, heat transfer and thermal performance of a periodically dimple-protrusion patterned surface have been investigated to enhance energy-efficiency in compact heat exchangers. The local heat transfer coefficients on the dimple/protrusion walls are derived using a transient TLC (Thermo chromic Liquid Crystal) technique. The periodically patterned surface is applied to the bottom wall only or both the bottom and top walls in the test duct. The ratio of dimple (or protrusion) depth to duct height is 0.25and the ratio of duct height to dimple (or protrusion) print diameter is 1.15. The Reynolds number is tested in low range values from 1000 to 10000.

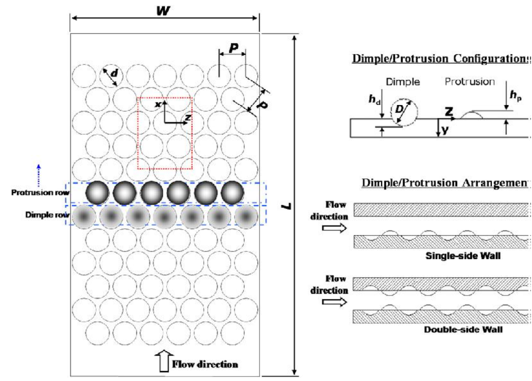

Fig No: 8Test section of dimple and protrusion [5]

On the single-side patterned walls, various secondary flows generated from the dimple/protrusion coexist. The vortices induced from the upstream affect strongly on the downstream pattern. For the double-side patterned wall case, vortex interaction affected by the opposite wall enhances highly the heat transfer. The heat transfer augmentation is higher in the lower Reynolds number due to the effective vortex interactions. Therefore, the performance factor considering both heat transfer enhancement and pressure loss increases with decreasing the Reynolds number.

R.P. Saini , JitendraVerma[6] The heat transfer coefficient between the absorber plate and air can be considerably increased by using artificial roughness on the underside of the absorber plate of a solar air heater duct. Under the present work, an experimental study has been carried out to investigate the effect of roughness and operating parameters on heat transfer and friction factor in a roughened duct provided with dimple-shape roughness geometry.

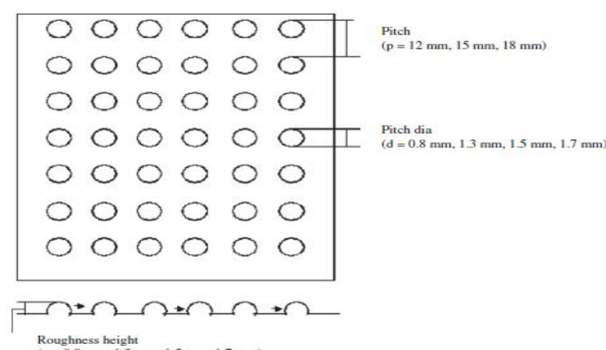

Fig No.9 Schematic diagram of dimple-shape geometry[6].

The investigation has covered the range of Reynolds number (Re) from 2000 to12000, relative roughness height (e/D) from 0.018 to 0.037 and relative pitch (p/e) from 8 to 12. Based on the experimental data, values of Nusselt number (Nu) and friction factor (fr) have been determined for different values of roughness and operating parameters. In order to determine the enhancement in heat transfer and increment in friction factor values of Nusselt number and friction factor have been compared with those of smooth duct under similar flow conditions. Correlations for Nusselt number and friction factor have been developed for solar air heater duct provided such artificial roughness geometry.

Sang Dong Hwang, Hyun Goo Kwon, HyungHeeCho[7]This study 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. The test duct was 15 mm in height and 105 mm wide. The print diameter of the dimple/protrusion was 12.99 mm and the height of the dimple/protrusion was 3.75 mm.

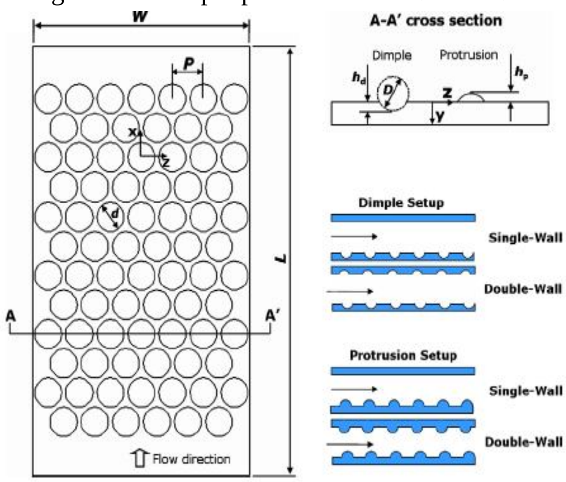

Fig no.10 Schematic diagram of Test Section Geometry[7].

Local heat transfer coefficients on the dimple/protrusion wall were measured using a transient TLC technique. Friction factors and performance levels are presented with the test cases. The Reynolds number, based on the duct hydraulic diameter, was varied from 1000 to 10,000.From the results, thermal characteristics and performance levels were different in each test case. For the dimple wall case, on the single and double-walls, thermal characteristics had similar patterns. However, flow mixing was higher for the double-wall than the single-wall, which resulted in enhanced heat transfer. As the Reynolds number decreased, the relatively low heat transfer region induced inside the dimple became wider and the local minimum of the heat transfer coefficient within the dimple moved downstream. For the protrusion wall case with the double-wall, the heat transfer coefficient increased greatly due to flow acceleration and stronger mixing flow. However, the heat transfer pattern was similar in both the single and double-wall cases. At high Reynolds numbers, the heat transfer pattern on the protrusion surface was 'pea-shaped' and upon decreasing the Reynolds number, the pattern became circular. Heat transfer enhancement was very high at low Reynolds numbers at both the dimple and protrusion walls. At ReDh = 1000, the enhancement levels were 14 and 7 for the double protrusion wall and the double dimple wall, respectively. However, at a high Reynolds number of 10,000, the enhancement level observed was from 2 to 3. For such a high heat transfer increment at the low Reynolds number, the performance factor is very high in this flow range. At a Reynolds number of 1000, the performance factors were 6.5 and 6 for the double protrusion wall and the double dimple wall, respectively.

III. PROBLEM STATEMENT

From literature survey we can say that very less work is done on elliptical dimple surface so it is needed to investigate detail geometrical parameters and thermal performance of Elliptical dimples.

IV. OBJECTIVE OF TOPIC

1. Investigation of Heat transfer enhancement in Elliptical Dimples
2. Pressure drop estimation in Elliptical Dimples.
3. Thermal Performance enhancement analysis in Elliptical Dimples.
4. Will propose best geometry which gives maximum Thermal Performance.

V. EXPERIMENTAL SETUP

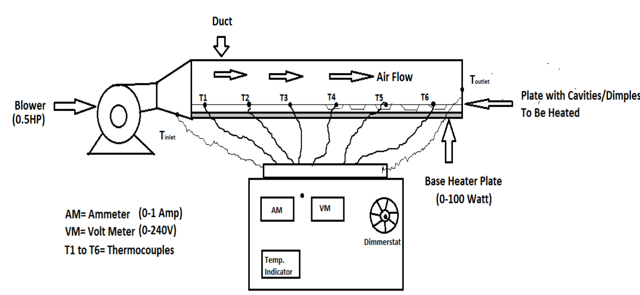

Fig No.11. Experimental setup

An experimental set-up has been designed and fabricated to study the effect of elliptical dimpled surface on heat transfer and fluid flow characteristics in rectangular duct. A schematic diagram of the experimental set-up is shown in Figure11.

1. Tests were carried out in a rectangular divergent Acrylic duct of size (100mmx25mm) of aspect ratio 1.5 and the Acrylic duct is connected to the blower of (0.5hp capacity)by means of a cylindrical pipe section via aluminium tape section.
2. The duct is open from the bottom of the duct to insert the

plates and heater assembly.

3. The test plate (500mmx25mm) is directly kept on the Nichrome plate heater.
4. Insulation is provided beneath the plate heater by applying the thermal paint and placing asbestos and plywood sheet.
5. The plate heater of size (500mmx25mm) and the insulating material are clamped together to form a single assembly.
6. This assembly is inserted in the duct from bottom open part is open.
7. A flow control valve is provided on the blower inlet so has to control the discharge of the blower. Flow rate was measured using a digital anemometer.
8. Connected to a wattmeter which is further connected to the dimmerstat. The wattage of the plate heater is the heater is varied with the help of the dimmerstat.
9. The blower, wattmeter, digital temperature indicator and dimmer stat are connected individually to the main supply of 230 volt.
10. K–type thermocouples (24SWG) are used to measure the temperatures inside the duct. Eight thermocouples are used to measure the surface temperature of the test plate at different locations. One thermocouple is used to measure the inlet temperature of air inside the duct and another thermocouple is used to measure outlet temperature. an universal data logger is used to displayed the measured temperature by the thermocouples.
11. Differential pressure sensor is connected by means of probes to the duct. One probe is connected just before the test section and one just after the test section.

Range of flow and dimple parameters:-

Reynolds number based on hydraulic diameter Re= 5000 to 15000
Dimple print diameter D= 15mm
Dimple depth to print diameter ratio=δ/D= 0.33
Channel height to dimple print diameter ratio H/D 0.5

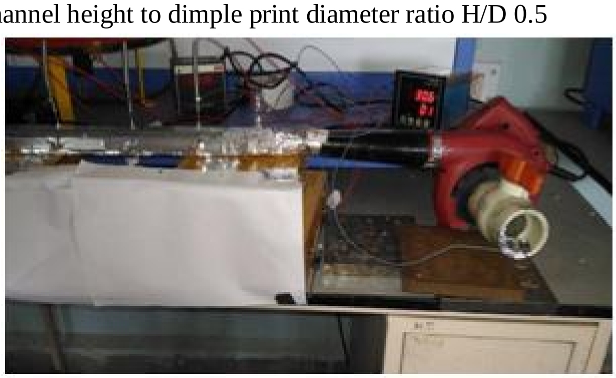

Fig No.12 Actual Experimental Setup

Experimental Procedure

- The experimental set up was assembled and all the electrical connections were made as shown in Fig No. 12..
- After checking all electrical connections power supply was switched on.
- The controller on the dimmerstat is operated to increase the voltage supplied to the plate heater from zero to a certain value so that the power input to the plate heater is set at 3 different stages 15 W, 25W, 35W.
- The wattmeter displays the power input to the heater.
- The temperature of the plate is continuously monitored until the plate reaches steady state
- With the help of flow control valve, the discharge of air from the blower is controlled for required speed over dimple plate.
- Temperatures of different thermocouples were continuously recorded at a regular interval of 5 min till the steady state is reached.
- After the steady state is reached, temperatures of different thermocouples were recorded from the temperature indicator display and power rating from wattmeter was recorded.
- Flow control valve was then adjusted to take observation at next higher speed.
- Again wait till the next steady state is attained.
- This procedure was first carried for flat pate (i.e. for plate without dimples) and then for plates with different dimpled configurations.

Test Plate

It is the main component of the experiment. The heat transfer rate is to be measured from the plate. The plate is made of aluminum because of its good thermal conductivity and lesser cost.

Fig No.13 Divergent Smooth plate without dimple geometry

Fig No.14 Divergent plate with Inline arranged Elliptical dimple geometry

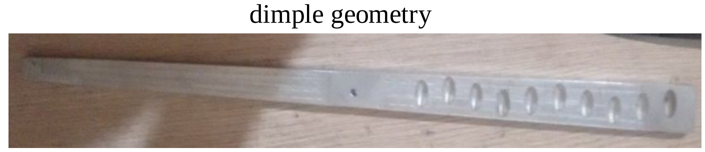

Fig No.15 Divergent plate with Offset arranged Elliptical dimple geometry

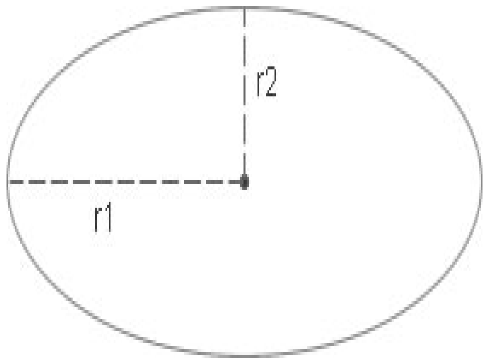

Fig No. 16 Elliptical geometry

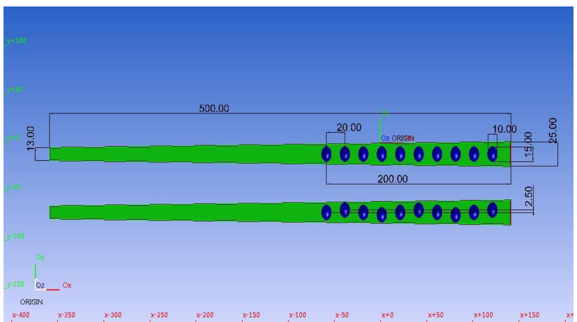

Fig No.17 Elliptical Dimple Geometry

Data Reduction:

Heat absorbed by air

$Q = \dot{m} C_p \Delta T$

$\dot{m} = \rho A V$

ρ = Density of air

Q = Heat absorbed by air

A = Area of cross section of duct

$\dot{m}$ = Mass flow rate of air v = Velocity of air

Cp = Sp. heat of air at constant press.

$\Delta T = T_{out} - T_{in}$

Heat Convected

Heat absorbed by air = Heat Convected

$$Q = h\, A_p\, \Delta T$$

Heat transfer coefficient is = $\mathbf{h} = \frac{Q}{A_p \times \Delta T}$

h =

Ap = Surface area of plate

ΔT= mean temp. of plate – mean temp. of air

Nusselt number calculations:

$\mathbf{Nu} = \frac{hD}{k}$ Where, $\mathbf{D} = \frac{2WH}{W+H}$

D = Hydraulic diameter of duct, H= Height of duct, k = Conductivity of air, W= width of duct

Turbulent flow Dittus-Boelter Correlation through closed conduits for Nusselt number

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

Where Re= Reynolds Number and Pr= Prandlt Number

VII. RESULTS AND FINDINGS

Figure 18, 19 and 20 shows that variation of Heat transfer coefficient with Reynolds Number on Smooth plate, inline and offline Elliptical dimples with different power input. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum heat transfer coefficient at 35 W power input.

A Smooth plate

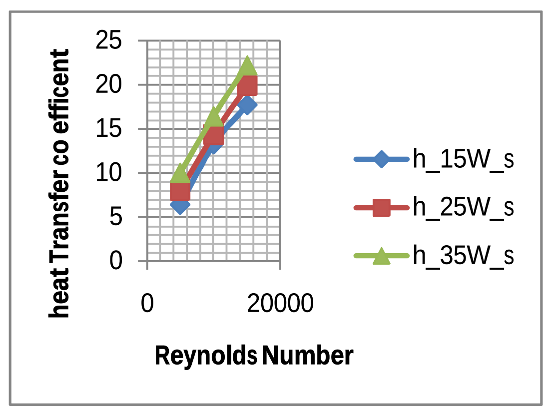

Fig No 18 Variation of Heat transfer coefficient with Reynolds Number on Smooth plate

Inline Elliptical Dimple

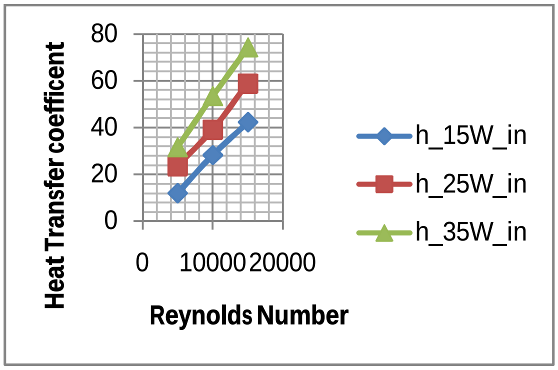

Fig No. 19 Variation of Heat transfer coefficient with Reynolds Number on Inline Elliptical Dimples

Offline Elliptical Dimple

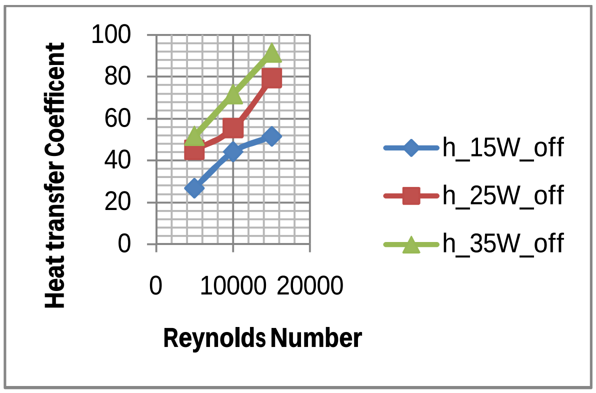

Fig No. 20 Variation of Heat transfer coefficient with Reynolds Number on offline Elliptical Dimples

Figure 21 shows that combined variation of Heat transfer coefficient with Reynolds Number on Smooth plate, inline and offline Elliptical Dimples with different power input 15, 25, 35. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum heat transfer coefficient at 35 W power input with offline Elliptical dimples.

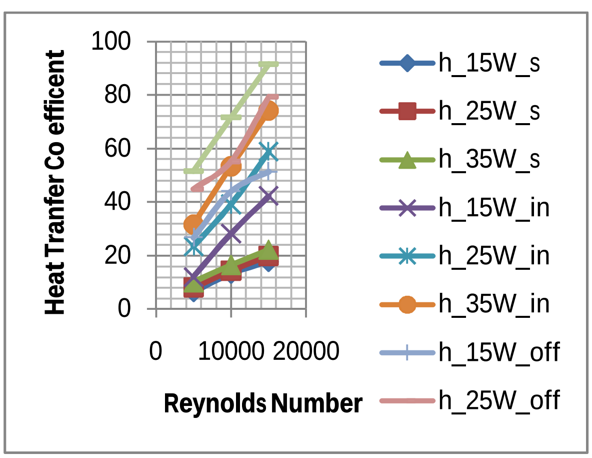

Fig No. 21Combined Variation of Heat transfer coefficient with Reynolds Number at 15, 25 and 35 Watt on Smooth, inline and offline Elliptical dimples.

Figure 22, 23 and 24 shows that variation of Nusselt number with Reynolds Number on Smooth plate, inline and offline Elliptical dimples with different power input. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum Nusselt number at 35 W power input.

A Smooth plate

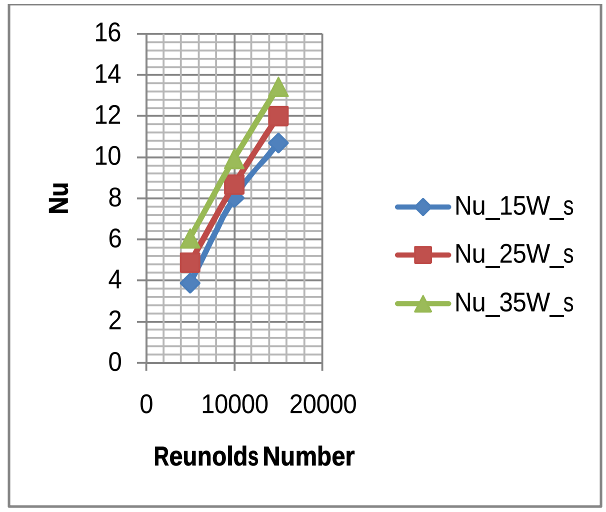

Fig No. 22 Variation of Nusselt number with Reynolds Number on Smooth plate

Inline Elliptical Dimple

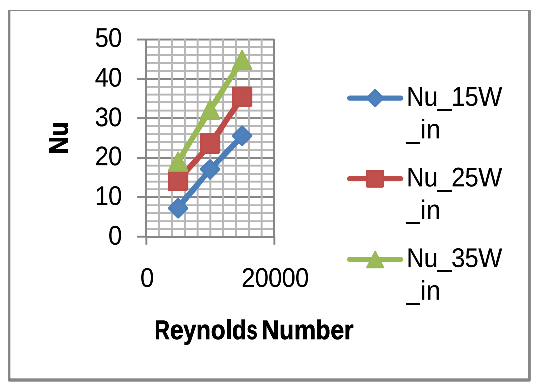

Fig No. 23 Variation of Nusselt number with Reynolds Number on Inline Elliptical Dimples

Offline Elliptical Dimple

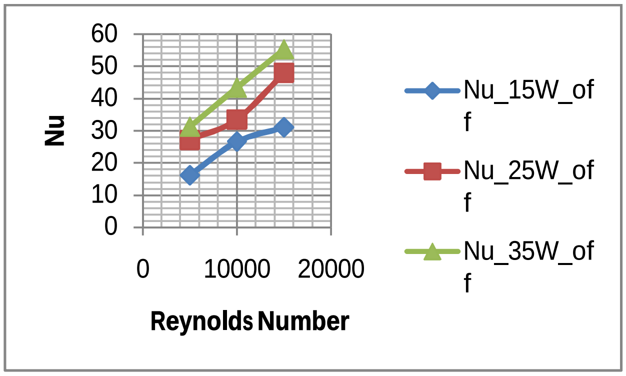

Fig No.24 Variation of Nusselt number with Reynolds Number on offline Elliptical Dimples.

Figure 25 shows that combined variation of Nusselt number with Reynolds Number on Smooth plate, inline and offline Elliptical and dimples with different power input 15, 25, 35. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum Nusselt number at 35 W power input with offline Elliptical dimples.

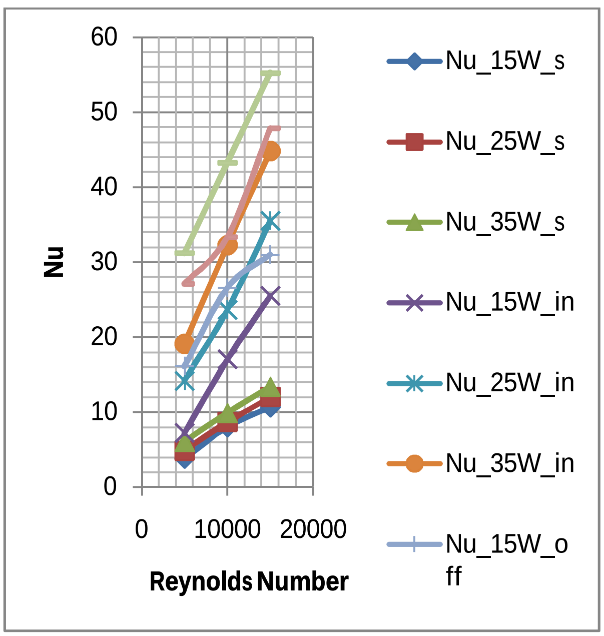

Fig No.25 Variation of Nusselt number with Reynolds Number at 15,25 and 35 Watt on Smooth, inline and offline Elliptical dimples

Figure 26 and 27 shows that variation of Heat transfer Enhancement with Reynolds Number on inline and offline Elliptical dimples with different power input. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum heat transfer coefficient at 35 W power input.

Inline Elliptical Dimple

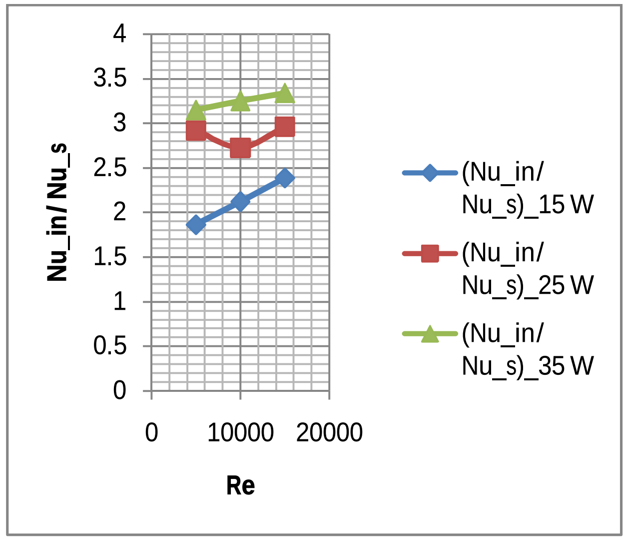

Fig No.26 Variation of Heat transfer Enhancement with Reynolds Number on Inline Elliptical Dimple

Offline Elliptical Dimple

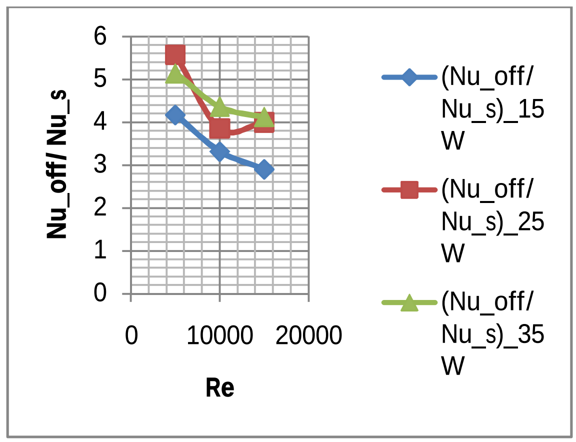

Fig No 27 Variation of Heat transfer Enhancement with Reynolds Number on offline Elliptical Dimples.

Figure 28 shows that combined variation of Heat transfer Enhancement with Reynolds Number on Smooth plate, inline and offline Elliptical dimples with different power input 15, 25, 35. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum Heat transfer Enhancement at 35 W power input with offline Elliptical dimples.

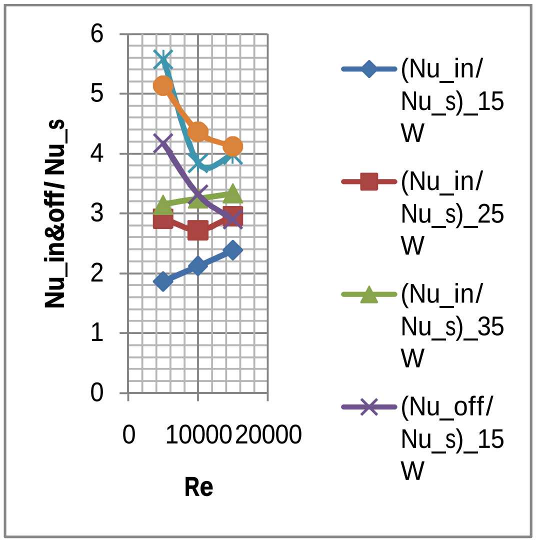

Fig No. 28 Variation of Heat Transfer Enhancement with Reynolds Number at 15, 25 and 35 Watt on inline and offline with respect to smooth Elliptical dimples.

Figure 29 shows the combined comparison between experimental and theoretical variation of Heat transfer coefficient with Reynolds Number and Nusselt number on Smooth plate, Reynolds number varies from 5000, 10000, 15000. A graph shows maximum heat transfer and Nusselt number is observed in theoretical calculations than experimental calculations due to various heat loses.

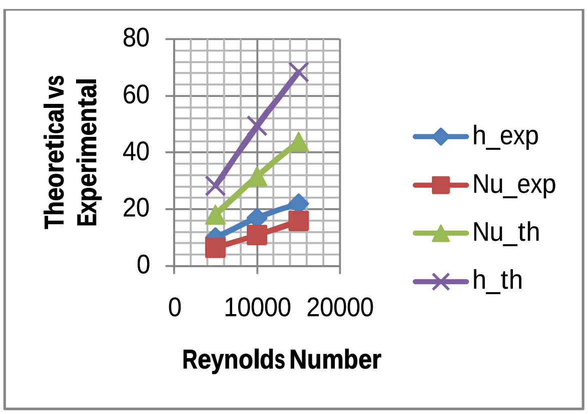

Fig No. 29 Combined theoretical and experimental Variation of Heat Transfer coefficient and Nusselt number with Reynolds Number for smooth plate.

Figure 30 Shows variation of pressure drop on Smooth plate, inline and offline Elliptical dimples. This graph shows there is maximum pressure drop is in offline Elliptical dimples.

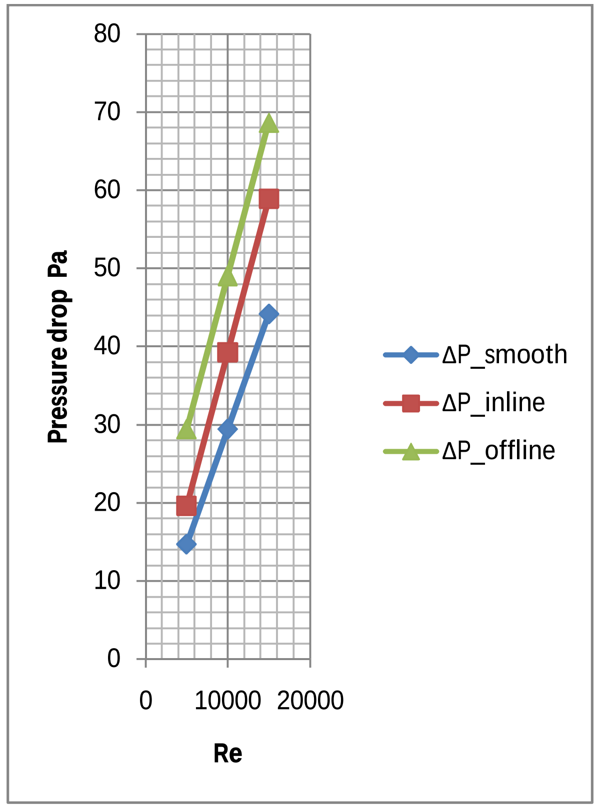

Fig No. 30 Variation of Pressure drop with Reynolds Number on Smooth, inline and offline Elliptical dimples.

Figure 31 shows that variation of Performance Enhancement factor (PEF) with Reynolds Number on inline and offline Elliptical dimples when compared with smooth plate. Reynolds number varies from 5000, 10000, 15000. Graphs shows maximum Performance Enhancement factor for offline Elliptical dimples as compared with inline Elliptical dimples.

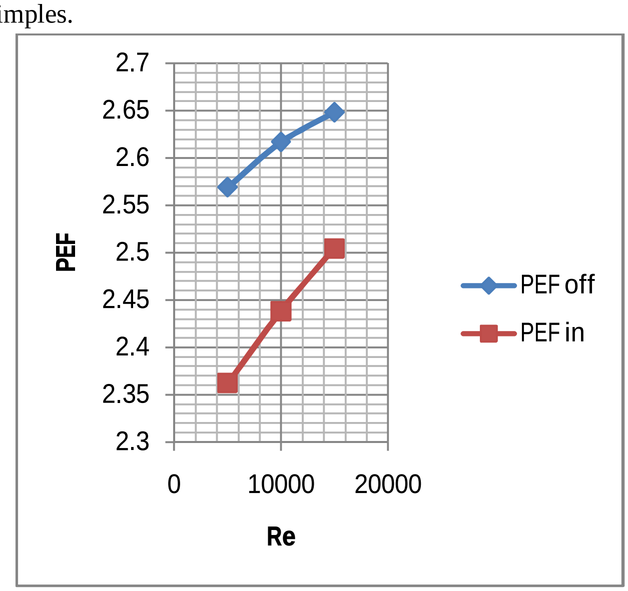

Fig No. 31 Performance Enhancement Factor of inline and offline Elliptical dimples

VIII. CONCLUSION

Experimental investigation of smooth, inline and offline type of Elliptical dimples on a flat plate is carried out and heat transfer characteristics were studied for each case with different variable parameters with respect to input power and velocity.

- The important findings of the experimental investigations are as follows-
- With different types of arrangements such as flat, inline and offline and with variable parameter such as velocity, v = 5 m/sec, v = 10m/sec, v = 15m/sec for this conditions cases are studied and it has been found that the offline arrangement in divergent plate gives optimum solution as compared to other arrangements.
- It is observed that Theoretical calculated heat transfer coefficient and Nusselt number is greater than experimental investigated versus Reynolds number.
- It is observed from the experimental results that the Performance Enhancement factor for offline Elliptical dimples is optimum solution than inline Elliptical dimples.

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