

Experimental Investigations On Heat Transfer And Frictional Characteristics Of A Turbulator Roughened Flat Plate

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ABSTRACT

a Experiment has been carried out to study the effect of artificial roughness on the heat transfer coefficient and frictional factor by incorporating inverted U-shaped turbulators in inline and staggered arrangement on the flat plate inclined at an angle 12° to the plate surface. The roughened plate is uniformly heated by heater and three sides of the turbulator are insulated. Experiment is carried out by taking Reynolds no. ranges from 12000 to 30000. Turbulator height 12mm and length 40mm. Pitch is varied from 10mm to 30mm. The angle of attack of flow on turbulators 90° kept constant during the whole experimentation. Carry the experiment for both types of plates having inline and staggered arrangement of turbulator. The heat transfer and friction factor data obtained is compared with smooth flat plate without roughness of same dimensions and in same flow conditions. Found out the optimum arrangement of turbulator geometry amongst all plate geometry. As compared to the smooth duct, the turbulator roughened plate enhances the heat transfer and friction factor by 3 to 4 time than that of smooth flat plate.

Keywords— a Heat transfer enhancement, artificial roughness, turbulator, spoiler.

ARTICLE INFO

Article History

Received : 18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Many engineering techniques have been investigated for increasing convective heat transfer rate. One of the most common methods is to induce roughness on the surface to be implemented by various techniques which would improve turbulence in the flow. By the engineering point of view that there must be minimum drag while making the design for roughness on the surface. Flow characteristics and the flow structure is very important in heat transfer enhancement technique. Because the heat transfer depends on characteristic flow, three dimensionality, share layer reattachment. The use of artificial roughness or turbulence promoters on a surface is an effective technique to enhance the rate of heat transfer to the fluid flowing in a duct.

However, it will result in an increase in frictional losses leading to more power required by fan or blower. So to reduce such losses turbulence must be induced at the closed to the surface. When a flowing fluid comes in contact with a stationary surface, a thin viscous sub layer develops adjacent to the wall in turbulent boundary layers. "The viscous sublayer is nothing but the thin layer in the vicinity of the wall in which the damping effect of the molecular viscosity on the turbulent velocity fluctuations is dominating. This layer is bounded from one side by a fixed surface to be tested and on the other side by continuously changing flow. In a very thin layer in the vicinity of wall, conduction process is maximum. Beyond this, heat transfer process is dominated by convection since heat transfer rate is adversely affected due to lower thermal conductivity of the

air. So to overcome this problem one of the method is used called 'interruption to flow pattern' instead of increase in surface area. This technique is mainly used in internal cooling passages for turbulent gas flows, apart from that it is giving better performance in solar air heaters (Reynolds number, $Re \leq 12,000$). By roughness can be produced by several methods, forming, pasting ribs, dimples, using expanded metal mesh over the surface, orienting ribs transverse, oblique, V-shaped either upstream or downstream to the flow. Ribs of different cross-sections like semicircular, triangular, square, rectangular, trapezoidal, wedge shaped or chamfered ribs and circular wires have been examined so far.

One of the passive technique used is repeated ribs or turbulators have been used for enhancing heat transfer rate. Such a arrangement breaks the laminar flow which induced intermixing of the laminar flow pattern which fluctuates the flow pattern followed by improving heat transfer from surface to surrounding by fluid. The parameters govern the flow characteristics in a rib-roughened channel consist of the channel aspect ratio, angle of attack, pitch-to-height ratio, rib aspect ratio and the pattern of rib arrangement on walls such as in-line, staggered, crisscross. Other passive technique used is surface roughening which contains dimpled surfacing. This method is easy and fabrication of this type of surfaces is cheap.

An experimental investigation has been carried out to study the heat transfer coefficient and friction factor by using artificial roughness in the form of specially prepared inverted U-shaped turbulator on the absorber surface of an air heater duct. Studied by Santosh b. bopche et.al [1] The experiments encompassed the Reynolds number range from 3800 to 18000; ratio of turbulator height to duct hydraulic mean diameter is varied from, $e/D_h = 0.0186$ to 0.03986 ($D_h = 37.63$ mm and $e = 0.7$ to 1.5 mm) and turbulator pitch to height ratio is varied from, $p/e = 6.67$ to 57.14 ($p = 10$ to 40 mm). The angle of attack of flow on turbulator, $\alpha = 90^\circ$ kept constant during the whole experimentation. With varying the number of rib wall P. R. Chandra, C. R. Alexander et.al [2] studied experimental study of surface heat transfer and friction characteristics of a fully developed turbulent air flow in a square channel with transverse ribs on one, two, three, and four walls is reported. Tests were performed for Reynolds numbers ranging from 10,000 to 80,000. The pitch-to-rib height ratio, P/e , was kept at 8 and rib-height-to-channel hydraulic diameter ratio, e/D_h was kept at 0.0625. The channel length-to hydraulic diameter ratio, L/D_h , was 20. The heat transfer coefficient and friction factor results were enhanced with the increase in the number of ribbed walls. Heat transfer in rectangular channels with transverse and V-shaped broken ribs was investigated by Giovanni Tanda. [3]. The ribs, having rectangular or square sections, were deployed transverse to the main direction of flow or V-shaped with an angle of 45 or 60 deg relative to flow direction. Area-averaged data were calculated in order to compare the overall performance of the tested ribbed surfaces and to evaluate the degree of heat transfer enhancement induced by the ribs with respect to the smooth channel. Alok Chaube et.al [9] have studied on the

Analysis of heat transfer augmentation and flow characteristics due to rib roughness over absorber plate of a solar air heater in which the artificial roughness in the form of ribs on a broad, heated wall of a rectangular duct for

turbulent flow (Reynolds number range 3000–20,000, which is relevant in solar air heater) has been carried out. M.M. Sahu, et.al [8] have done the experiment on augmentation of heat transfer coefficient by using 90° broken transverse ribs on absorber plate of solar air heater. The roughened wall has roughness with pitch (P), ranging from 10–30 mm, height of the rib of 1.5 mm and duct aspect ratio of 8. The air flow rate corresponds to Reynolds number between 3000–12,000. The heat transfer results have been compared with those for smooth ducts under similar flow to determine thermal efficiency of solar heater.

This experimental analysis carried out by incorporating the inverted U-shaped turbulator on the flat aluminium plate. The turbulators are arranged in inline and staggered arrangement on the surface of the aluminium plate. The angle of attack is at 90° to the turbulator surface. Turbulators are inclined at 12° to the plate surface. The pitch selected for arrangement are ranges from 10mm, 20mm, 30mm whereas height is constant that is 12mm as shown in figure below. The data is same for both plates having inline and staggered arrangement of turbulator. The experiment is concerned with the change in the Reynolds's number and Nusselt's number with respect to (pitch /height) ratio by enhancing heat transfer rate. And effect of Nusselt's number and friction factor with respect to Reynolds number. Since we know that as number of sharp edges increases the heat transfer rate increases. As turbulator consisting of more number of sharp edges than that of ribs it will shear more the viscous boundary layer. The two horizontal edges of turbulator will induce free separate shear layers which will mixed with each other at downstream of turbulator which will create turbulence due to which leads into heat transfer enhancement. Since by using the ribs it will create eddies which will move upstream and downstream as shown in figure. It will reduce the heat transfer as well as increase friction factor. So use of the U-shaped turbulator is advantageous and it does not only retard formation of eddies but also increases heat transfer.

II. METHODOLOGY

A. Experimental set-up

The experimental setup is designed to study the effect of inverted U-shaped aluminium turbulators (having inline and staggered arrangement) on heat transfer by incorporating test plate inside the rectangular channel. Roughness element is incorporated under side of one broad wall which will acted upon fluid flow. So rectangular channel is having one rough wall and three smooth wall. The schematic diagram of the set-up is as shown in fig.

The set-up is consisting of the components like centrifugal blower, flow control valve, orifice meter, exit length, test duct, entrance section, dimmer stat and temperature indicator. The main rectangular test section consist of three smooth and one rough surface. Underside of roughened plate heater plate is mounted whose wire is connected to dimmer stat. Under the heater a Bakelite plate is mounted. And whole assembly is assembled by two clamps. And whole test section is insulated from surrounding by wounding asbestos rope and putty. The schematic and picture of set-up is as shown in figure 1.

B. Components Of Experimental setup

- Blower.

- Flow control valve.
- Orifice-meter.
- U- tube manometer.
- Transition section.
- Rectangular test section.
- Temperature indicator.
- Ammeter.
- Voltmeter.
- Voltage stabilizer.

III.EXPERIMENTAL PROCEDURE

A. Experimental procedure for estimating heat transfer.

The flat plate (without turbulator) placed in rectangular channel followed by heater and external Bakelite plate assembled and fixed together by clamps and whole assembly insulated by asbestos rope and putty. This reduces wastage of heat to the surrounding during procedure.

- 1) Then connect thermocouple wires (copper and constantan) between plate and temperature indicator.
- 2) And check whether all wires are connected properly and take readings for ambient temperature.
- 3) Check the test section for air leakage.
- 4) Switch on the heater and set the value of heater power to 85W. Maintain the steady state condition of the plate by assuring readings of all thermocouple wires are constant.
- 5) Then switch on the blower and take the initial reading at all thermocouple on temperature indicator by turning the knob at respective thermocouples.
- 6) Then after each 10 min. take readings at each thermocouple. And take reading for friction factor at test section by taking reading by U-tube manometer whose ends are attached before and after test section.
- 7) Take the readings after every 10 minutes up to half an hour.
- 8) Then dismantle the test section and attach the flat plate with turbulator (pitch =10 mm).
- 9) Assemble the test section as describe in previous case.
- 10) Conduct same procedure for this plat as previous case.
- 11) Test the plates by same procedure with pitch P equal to 10mm, 20mm, 30mm for inline and staggered arrangement of the turbulator as shown in fig. 2.

B. Figures and Tables

Figure 1 shows CAD diagram of Experimental setup used for estimating heat transfer enhancement consisting various accessories.

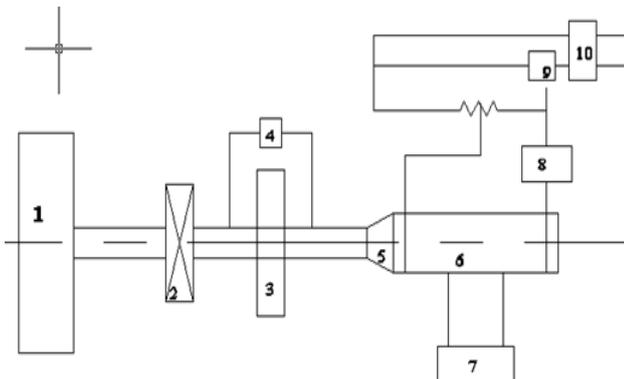


Fig. 1 Experimental set-up

Fig. 2 Shows flat rectangular test plate made up of aluminium of dimensions having length 600mm, width 100mm and 6mm thickness. It is having inverted U-shaped turbulators make 12° angle with the plate surface arranged in inline and staggered pattern having different pitches P equal to 10mm, 20mm, 30mm for respective patterns.

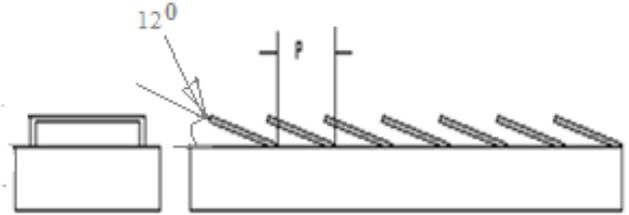


Fig. 2 Turbulator roughened test plate with pitch 'p'.

C. Test Plates



Fig.3 inline arrangement of inverted U-shaped turbulator having pitch P equals to 10mm.



Fig.4 Staggered arrangement of inverted U-shaped turbulator having pitch P equal to 10mm.



Fig.5 Inline arrangement of inverted U-shaped turbulator having pitch P equal to 20mm.



Fig.6 Staggered arrangement of inverted U-shaped turbulator having pitch P equal to 20mm.



Fig.7 Inline arrangement of inverted U-shaped turbulator having pitch P equal to 30mm.



Fig.8 Staggered arrangement of inverted U-shaped turbulator having pitch P equal to 30mm.

IV.DATA REDUCTION

To find maximum flow rate of Blower

$$m_a = C_d \times \left(\frac{\pi}{4}\right) \times d_o \times \rho_a \times \left[\frac{2 \times g \times h_a}{1 - \beta^4}\right]^{\frac{1}{2}}$$

To find height of air column

$$h_a = \frac{\rho_w \times h_w}{\rho_a}$$

To find velocity of air

$$m_a = \rho_a \times V_a \times A_{duct}$$

To find maximum Reynolds number for Blower

$$V_a = \frac{m_a}{\rho_a \times A_{duct}}$$

Reynold's Number

$$Re = \frac{\rho_a \times V_a \times D_h}{\mu}$$

Heat transfer rate calculation

$$Q = m_a \times C_p \times (T_{out} - T_{in})$$

$$Q = h \times A_{plate} \times (T_{plate} - T_{bm})$$

Theoretical Nusselt Number

$$Nu_{th} = 0.239 \times Re^{0.8} \times Pr^{0.4} \dots \dots \text{Dittus Boelter eqn}$$

Pressure reading and friction factor

The height of water column in Micro-manometer is 0.003 m
So the pressure difference is calculated by following formula

$$\Delta P = g \times h \times \left[\rho_{benzyl\ alcohol} + \rho_{water} \times \left[\left(\frac{d_o}{d}\right)^2 - 1 \right] \right]$$

Calculation of friction factor

$$f = \frac{\Delta p \times D_h}{2 \times \rho_a \times L \times v^2}$$

Calculation for thermal enhancement factor

$$\eta n = \frac{(Nu/Nu_0)}{\frac{f}{f_0}^{\frac{1}{3}}}$$

V.RESULT & DISCUSSION

The Dittus - Boelter equation valid for Reynolds numbers above 5000 is given by:

$$Nu_{u0} = 0.023 Re^{0.8} Pr^{0.4}$$

Baseline Nusselt number is used to normalise the values of measured Nusselt number on dimpled surface. The Nusselt number distribution is asymmetric due to the rib induced secondary flow. The flow that separates from the rib surface reattaches to the endwall and forms a vortex, and the vortex moves along the rib installation direction. Due to the flow reattachment and the vortex, relatively high heat transfer coefficients are caused between ribs.

The fig 8 shows the variation of baseline Nusselt number with Reynolds number. It shows the decreasing trend with increase in Reynolds number.

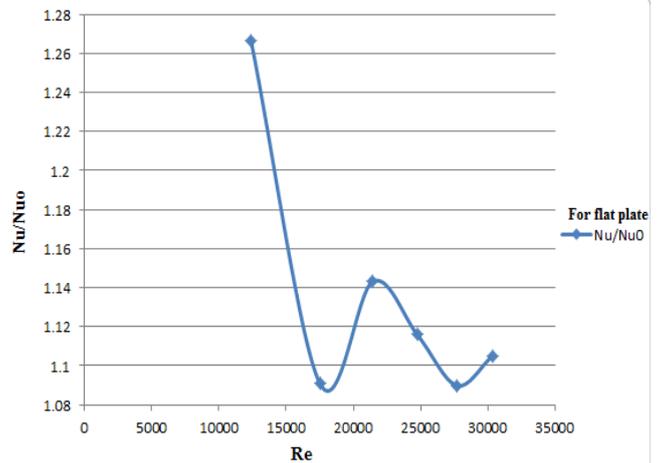


Fig. 9 Comparison of Nu/Nu0 as function of Reynolds number for flat plate.

In case of ribs, boundary layer get shared at a place only at its top-edge. Heat transfer rate will be higher if the edges are sharper and friction penalty will also increases. Inverted U-shaped turbulator is having more number of sharp edges to shear more the viscous boundary layer. Two horizontal edges form two free shear layers which either mix-up downstream with the accelerated flow (underway of turbulator) and it generates turbulences or form two reattachment points at the hot surface in order to enhance the heat transfer rate but that depends only on the Reynolds number. Four side edges create a secondary flow may also interrupt the growth of the boundary layer which ultimately enhances the heat transfer coefficient.

A . Effect of Reynolds numbers on Nusselt number ratio

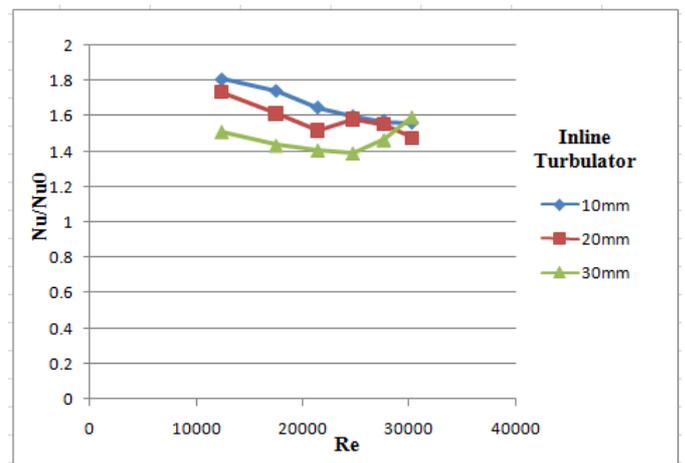


Fig. 10 Enhancement ratio as a function of Reynolds number for different plates having inline arrangement of turbulators of different pitch.

Figure 10 shows the graph between Nusselt number ratio Vs Reynold's number. It is shown that as compare to Nusselt number that of flat plate plate, inline arrangement of turbulators of various pitch have higher performance. That means the plate with obstructions on the surface have greater value of Nusselt number ratio than that of flat plate. From the graph we can see that the plate having inline arrangement of turbulator of pitch distance 10 gives higher value of nusselt number as compared to other plates having turbulator pitch 20 and 30.

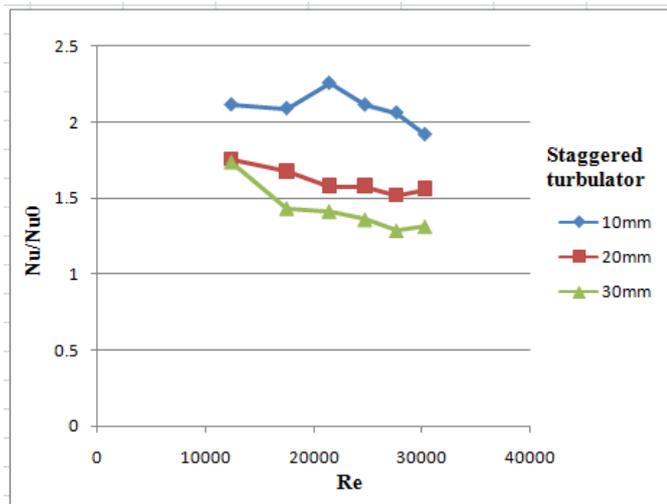


Fig. 10 Enhancement ratio as a function of Reynolds number for different plates having staggered arrangement of turbulators of different pitch.

Figure 10 shows the graph between Nusselt number ratio Vs Reynold's number. It is shown that as compare to Nusselt number that of flat plate, staggered arrangement of turbulators of various pitch have higher performance. That means the plate with obstructions on the surface have greater value of Nusselt number ratio than that of flat plate. From the graph we can see that the plate having staggered arrangement of turbulator of pitch distance 10 gives higher value of nusselt number as compared to other plates having turbulator pitch 20 and 30.

B. Effect of Reynolds number on Friction Factor

The value of friction factor ranges from 1.8 to 2.9 at Reynolds number from 10000 to 40000. It is observed that, as Reynolds number increases friction factor also increases for all patterns. Friction factor is higher for a 90° V-angled plate with 18mm pitch, because of strong recirculation of the flow and lowest. As depth increases vortices becomes stronger and stronger increase the friction factor.

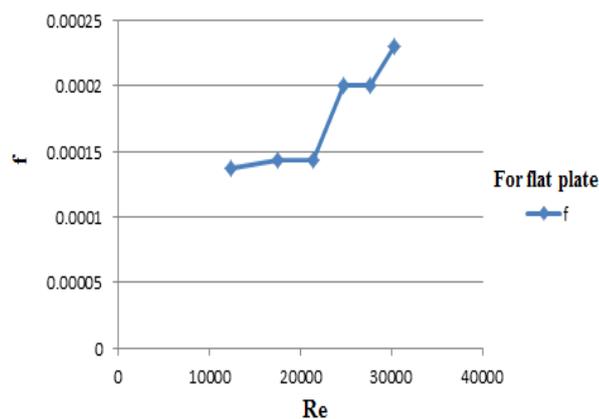


Figure 11 shows variation of friction factor with Reynolds number. The friction factor increases with increase in Reynolds number.

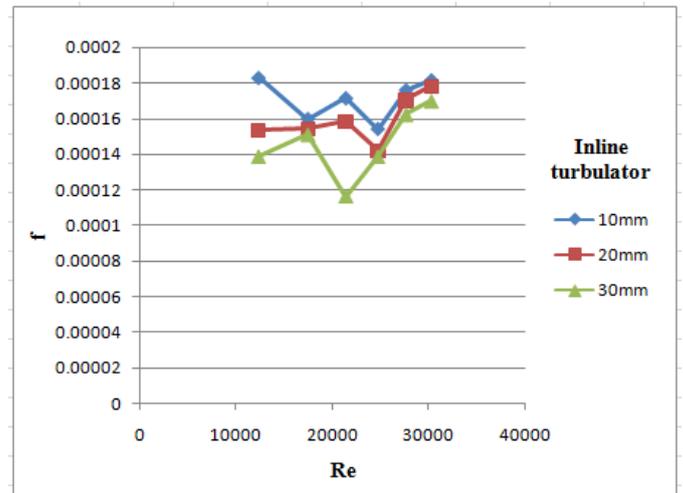


Fig. 11 Friction Factor as a function of 'Re' for plate having inline arrangement of turbulators of different pitch.

Figure 11 shows the graph between Friction factor Vs Reynold's number for the plate having turbulators arranged in inline pattern having different pitch. It shows that friction factor for plate having turbulators of 10 mm pitch distance has higher friction factor than that of plates having pitch distances 20 mm and 30 mm. And it gives lowest friction factor for plate having pitch distance equal to 30 mm.

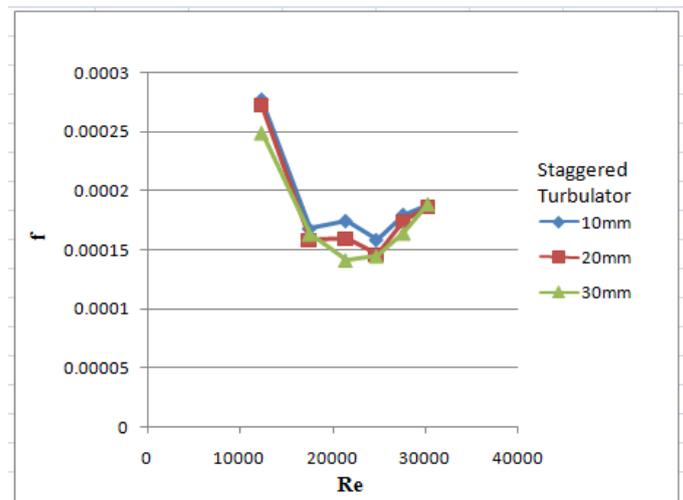


Fig. 12 Friction Factor as a function of 'Re' for plate having staggered arrangement of turbulators of different pitch.

Figure 12 shows the graph between Friction factor Vs Reynold's number for the plate having turbulators arranged in staggered pattern having different pitch. It shows that friction factor for plate having turbulators of 10 mm pitch distance has higher friction factor than that of plates having pitch distances 20 mm and 30 mm. And it gives lowest friction factor for plate having pitch distance equal to 30 mm.

VI.CONCLUSION

An experimental study of the air flow in a rectangular duct with one wall roughened with 'inverted U-shaped turbulators', subjected to uniform heat flux boundary condition has been performed. The remaining three walls were insulated by insulating material. These conditions were correspond to the flow in the application of solar air heater duct. The effect of

Reynolds number, relative roughness pitch and the heat transfer coefficient and friction factor has been studied. Results have been compared with those of a smooth duct under similar flow conditions to estimate enhancement in heat transfer coefficient and friction factor. Investigations have been carried out at Reynolds numbers in between ($Re=12000-30000$). Following conclusions have been drawn from the experiment:

- 1) Roughness pitch strongly affects the flow and hence also affects performance of the duct. The inverted U-shaped turbulator geometry shows appreciable heat transfer enhancement even at lower Reynolds number of Re equal to 15000 where it is not possible in case of ribs.
- 2) The maximum enhancement in Nusselt number and friction factor values compared to smooth duct. It is found that staggered arrangement having turbulator pitch distance 10 mm gives maximum heat transfer rate since it gives greater values of Nusselt number ratio than that of other pitch distances of turbulators.
- 3) Due to turbulences generated only in the viscous sub-layer region of boundary layer adjacent to hot surface results in better thermo-hydraulic performance that means maximum heat transfer enhancement at affordable friction penalty.
- 4) As from the graph in figure 10 and figure 11 we conclude that Nusselt number ratio for staggered geometry ($Nu/Nu_0=1.58-2.2$) for given range of Reynolds number ($Re=12000-30000$) of the turbulator of 10 mm pitch distance is having higher value than that of inline geometry ($Nu/Nu_0=1.45-1.8$). Also in case of plate having staggered turbulator geometry has affordable friction penalty, and having almost same friction factor value closed to inline geometry.

ACKNOWLEDGEMENT

This work has been carried out at Sinhgad Academy of Engineering, Pune. The author acknowledges Sinhgad Academy of Engineering and Savitribai Phule Pune University for grant.

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