"Experimental investigation of heat transfer enhancement by u-shaped Turbulator"

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Abstract - *An experimental investigation was carried out to find the optimum U shape turbulator array pattern on flat surface. The researchers have documented theories related to U shape turbulator array pattern (i.e. staggered array,inline array etc). Protrusions like U shape turbulator are mounted on internal flow passage to augment the heat transfer. This project work is related to investigation of the forced convection heat transfer over U shape turbulator array. The objective of work is to find heat transfer rate and frictional characteristics from test surface and results are compared with flat surface. The U shape turbulator array pattern is in V-angled shape (120°) with pitch is 40mm & 50mm , side by side with pitch is 25 mm & 30 mm, staggered arrangement with 30mm & 40mm.The Nusselt number, friction factor and thermal performance index was measured, U shape turbultor form are rectangle in shape with 45mm in width and 15mm in height, test plate is having dimensions of 600 x 100 x 6mm. The Reynolds number based on hydraulic diameter was varied from 17509.19 to 39153.5.*

Key Words: *Protrusion, heat transfer enhancement, turbulator array, Nusselt number, friction factor, thermal performance index.*

1. INTRODUCTION

The heat transfer rate in flow passage is increased by three techniques such as passive, active techniques and compound technique. Passive techniques are more preferable because of its simplicity in design and fabrication also the cost for fabrication is low in comparison with active and compound techniques.It includes the surface roughening or geometrical modifications in the flow channel which alters flow distribution and promotes more heat transfer. Another technique used is active technique, these techniques are complex to design and implement because it requires external input to disturb the flow. It includes mechanical aids, surface vibration, fluid vibration, electrostatic fields, and injection, suction and jet impingement.

A number of experimental investigations have done turbulator arrays and turbulator shapes. The flow disturbance is maximum for side by side arrangement of turbulators due to higher vortex generation than inline array . The oval shape turbulators are more efficient than spherical turbulator, with the peak end at upstream and round end at downstream produces more mixing of fluid which promotes the heat transfer rate. analytically analysed convective heat transfer in turbulised flow past a turbulatord surface. A parametric study is performed with k-ε turbulence model to determine the effects of Reynolds number, turbulator depth and Nusselt number on heat transfer enhancement, and have computed heat transfer coefficients in a channel with one side turbulatord surface. The Reynolds number based on the channel hydraulic diameter was varied from 200000 to 360000.They showed that more heat transfer was occurred downstream of the turbulators due to flow reattachment. Due to the flow recirculation on the upstream side in the turbulator, the heat transfer coefficient was very low. As the Reynolds number increased, the overall heat transfer coefficient was also increased.

This experiment includes study of heat transfer rate from angular, side by side ,staggered turbulator array plates of angle 120^0 & 30mm and 40 mm pitch.25mm&30mm pitch The Reynolds number ranges from 17509.19 to 39153.5. The heat transfer rate is given by the Nusselt number ratio i.e. ratio of practical Nusselt number and theoretical Nusselt number. The friction factor is calculated with help pressure drop across test section and compared with theoretical friction factor which is given by Dittus-Boelter equation. The obtained results are compared with flat plate results.

2. EXPERIMENTAL INVESTIGATIONS

Experimental Set Up An experimental set up has been designed to obtain the heat transfer rate in rectangular duct for turbulatord surface plate. Figure 1 shows the schematic diagram of experimental setup. This apparatus is open air flow duct that consist of blower (1) supplies air with Reynolds number ranges from 12000 to 30000, flow control valve (2) which is provided to adjust and control the flow,orifice meter (3) to measure the Reynolds number, entrance section (4), flow straightener (5) to convert the flow through circular pipe into rectangular duct, test section (6) within test plate is mounted having cross section area of 100 x 25mm with aspect ratio 4. The test section is fabricated with epoxy resin, the entrance and flow convertors are fabricated with PVC and sheet metal respectively. The exit section (7) of 250mm is used after test section to reduce the end effect. The uniform heat flux is supplied by the flat rectangular heater of 600 x100mm surface area, the heater is located below the test plate. The test section is insulated by bakelite sheet below the heater and silicon wool has wounded on test rig to prevent the radiative heat transfer. The calibrated k-type thermocouples are used to measure temperature at different locations [1]. The pressure drop across test section is measured by micro-manometer with double reservoir (ranges from 0.002-5 mbar) filled with benzyl alcohol and water.

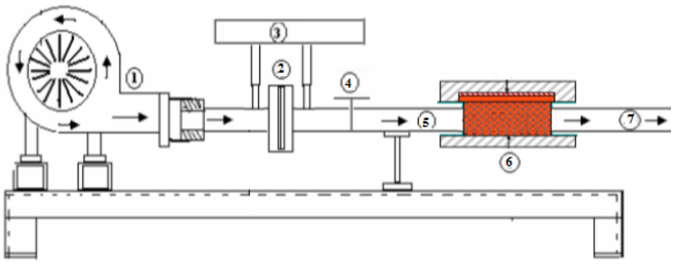


Fig. 1 Experimental set up

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 turbulatord surface. The netheat flux and the average test surface to bulk mean air temperature difference was determined over a testsection. Four values of flow rates were used for each set at same or fixed uniform heat flux. At each value offlow rate and the corresponding heat flux, system was allowed to attain a steady state before the temperaturedata were recorded. The pressure drops were measured when steady state is reached [1].

During experimentation the following parameters were measured:

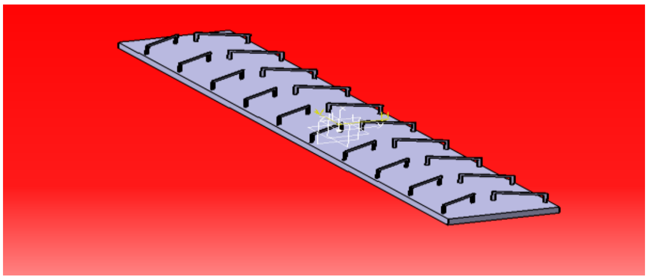
i) Pressure difference across the orifice meter.

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

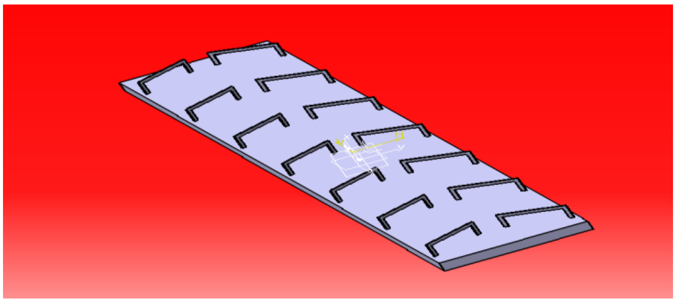
iii) Pressure drop across the test section.

TEST PLATES

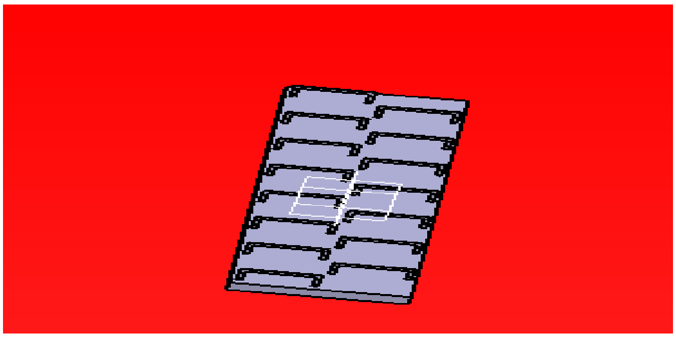
The figures below shows the schematics of turbulator pattern used, these test plate are having V-angled (60°, 90°, 120°) turbulator pattern with 18mm and 14mm pitch. The turbulators are made in staggered arrangement.



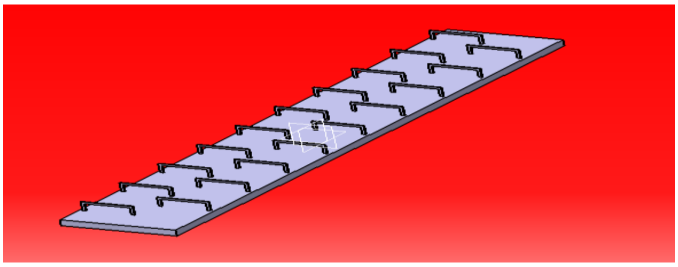
1)Test plate with angle 120° 40mm pitch



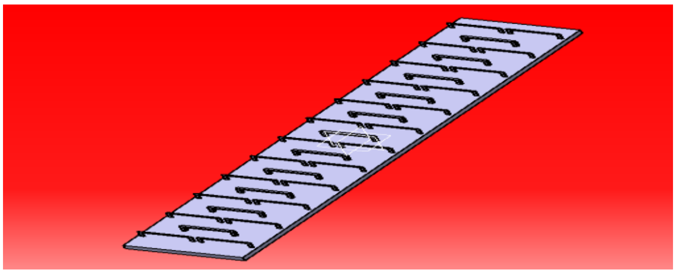
2)Test plate with angle 120° 50mm pitch



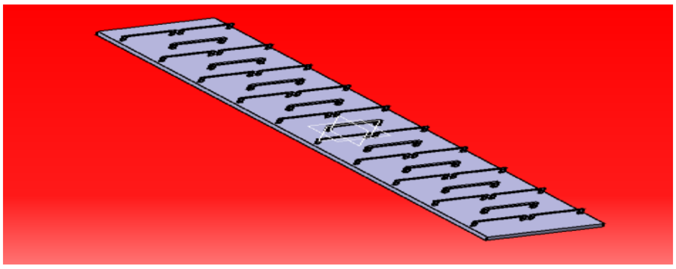
3)Test plate with side by side 25mm pitch



4)Test plate with side by side with 30mm pitch



5)Test plate with staggered with 30mm pitch



6)Test plate with staggered 40mm pitch

DATA REDUCTION

The values of performance parameter, mass flow rate of air at given heat flux can be calculated with steady state values of plate and air temperature.

A. Performance Parameters

Reynolds number based on hydraulic diameter

Re = 12300 to 40000

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

$$m_a = c_d \frac{\pi}{4} d_0 \rho_a \frac{2g}{1-} \qquad (1)$$

The useful heat gain can be calculated by:

$$Q_{useful} = m c_p (T_0 - T_i) \qquad (2)$$

$$Q_{useful} = Q - (Q_{conduction} + Q_{radiatic} \qquad (3)$$

Heat loss by conduction is given by:

$$Q_{conduction} = -kA \qquad (4)$$

Heat loss by radiation is given by:

$$Q_{radiation} = \sigma \epsilon A (T_s^{\ 4} - T) \qquad (5)$$

Convective heat transfer coefficient:

$$Q_{useful} = hA(T_s - T_{bm}) \qquad (6)$$

Nusselt number:

$$N_u = \frac{hD_h}{k} \qquad (7)$$

Friction factor from values of pressure drop across test rig:

$$f = \frac{\Delta p \times D_h}{2 \times \rho a \times L \times v^2} \qquad (8)$$

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.

RESULTS AND DISCUSSIONS

The boundary layer of the mainstream was separated by the turbulator and recirculation zone is created in the upstream side of the turbulator. The separated mainstream flow reattaches in the downstream surface and the reattached flow forms a twin vortex. As the flow comes out of the turbulator, it reattaches again at the downstream of the turbulator. These complex flow phenomena induce high and low heat transfer in and near the turbulator [7].

The heat transfer, thermal performance and friction factor characteristics of duct has given which is obtained from experimental data.

Baseline Nusselt Number

The Dittus - Boelter equation valid for Reynolds numbers above 3000 is given by:

$$N_{u0} = 0.023 Re^{0.8} Pr^{0.4} \qquad (9)$$

Baseline Nusselt number is used to normalise the values of measured Nusselt number on turbulator surface. The flow reattachment and the vortex produces relatively high heat transfer coefficients between ribs.For the turbulator only, relatively higher Nusselt numbers are observed at the flow reattachment region inside and downstream of the turbulator. A low Nusselt number region on the turbulator cavity is caused by the flow recirculation.

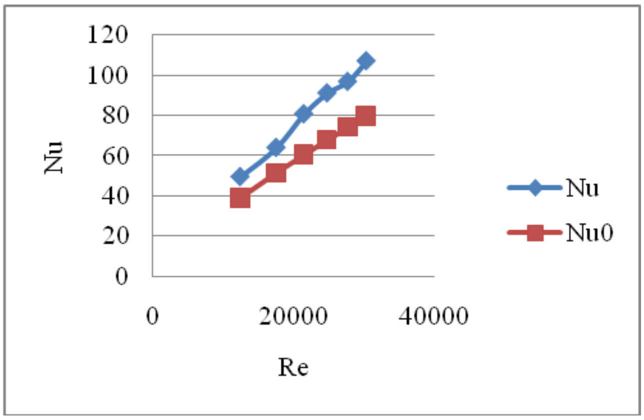


chart 1 Comparison of theoretical and practical Nusselt number as function of Reynolds number for turbulator plate

The variation of baseline Nusselt number with Reynolds number for turbulator plate. It shows the increasing trend with increase in Reynolds number.

EFFECT OF REYNOLDS NUMBER

It is observed that baseline Nusselt number increase with increase in Reynolds number and the rate of increase for plate is higher than for smooth plate. The measured Nusselt numbers are normalised by theoretical value of Nusselt number. The maximum value of Nusselt number obtained is 133.07 for side by side 25pitch pattern.

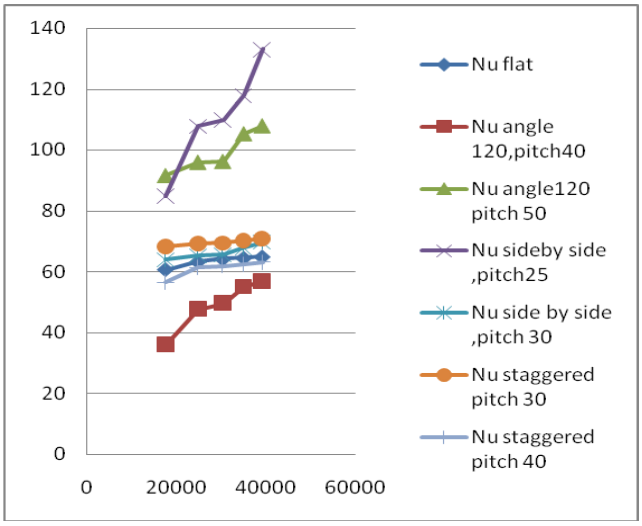


chart 2 Variation of Nusselt number with Reynolds number

The experimental Nusselt numbers are compared with theoretical Nusselt number.Effect of turbulator depth and number of turbulators can be found by the Nusselt number ratio which gives the normalised values of heat transfer rate.

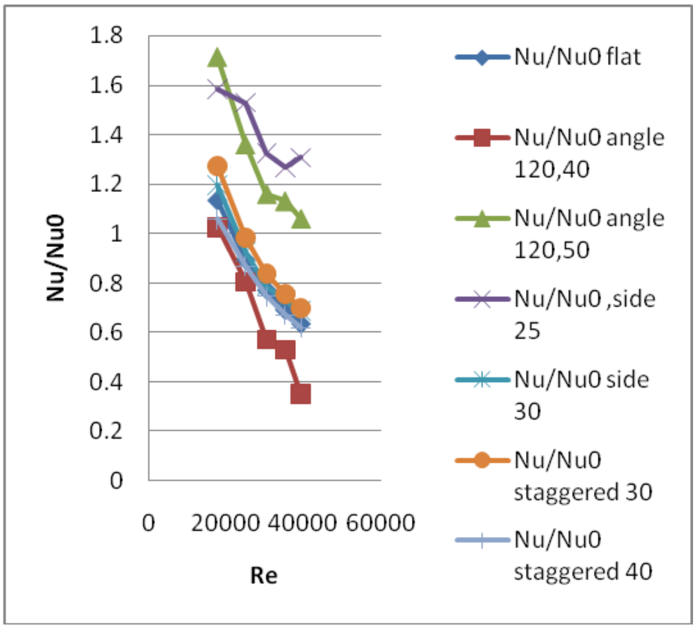


chart 4 Variation of Nusselt number ratios with Reynoldsnumber plates.

The normalised Nusselt number plot with increasing Reynolds number, the Nusselt number ratio increases with Reynolds number the maximum value is obtained is 1.3071 at39153.5 Reynolds number .

FRICTION FACTOR

The value of friction factoris 4.06 at Reynolds number from 17509.19 to 39153.5. From results it is observed that friction factor increases with Reynolds number. Friction factor is higher for a side by side plate with 25mm pitch, because of strong recirculation of the flow and lowest.

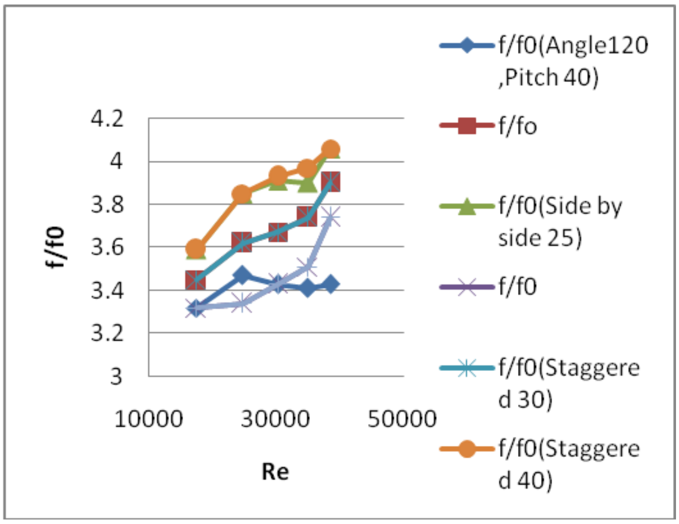


chart 3 Variation of friction factor ratios with Reynolds number for 25mm pitch turbulator plates.T

The friction factor ratio variation with Reynolds number ranges from 17509.19 to 39153.5. The maximum friction is observed4.06 for side by side with 25mm pitch plate

THERMAL PERFORMANCE INDEX

The thermal performance index parameter provides the quality of heat transfer augmentation (Nu/Nuo) and friction factor augmentation (f/fo)^(1/3). It increases with increase in number of turbulator

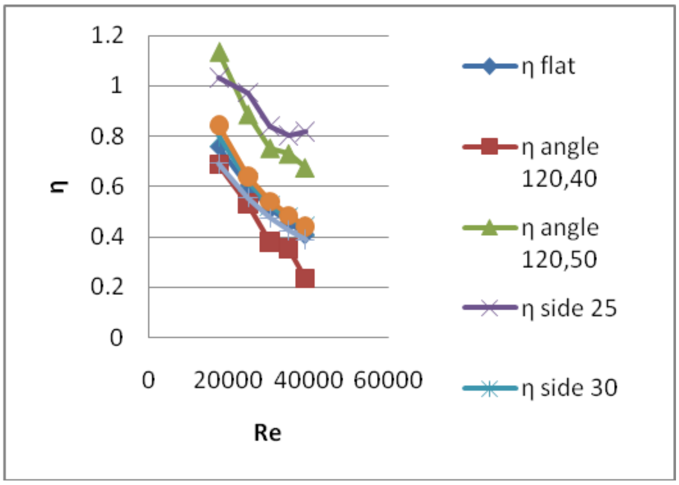


chart 5 Variation Thermal performance with Reynolds number

The variation of thermal performance with Reynolds number. Thermal performance decreases with Reynolds number, the higher thermal performance obtained is 0.8190 for side by side having pitch25mm array pattern

3. CONCLUSIONS

An experiment is carried out to study of the flow of air in a rectangular channel with u-shaped turbulator surface with side by side array patternwith 25mm pitch plates subjected to uniform heat flux with uniform boundary conditions.The effect of array pattern andTurbulator pitch on the heat transfer coefficient and friction factor has been studied. Experimental results i.e. heat transfer rate, friction factor and thermal performance are obtained and compared with smooth surface. Reynolds number varied from17509.19 to 39153.5.

1) The best plate from above is side by side having pitch25mm.

2) The The maximum heat transfer rate observed is 1.3071 for side by side arrangement of pitch 25mm.

3) Thermal performance index is 0.81906 for same arrangement.

4) The Nusselt numbers for all plate are considered to increase as the number of turbulator on plate increases. The higher value of Nusselt number is observed for plate with side by side with pitch 25mm.

ACKNOWLEDGEMENT

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NOMENCLATURE

A	-Effective heat transfer surface area, m2
Cp	-Specific heat at constant pressure for the air, Jkg-1K-1
Cd	-Coefficient of discharge for orifice
D	-Pipe diameter, m
d	-Orifice Diameter, m
g	-Acceleration of gravity, msec-1
H	-Height of test plate, m
h	-Convection heat transfer coefficient, Wm-2K-1
K	-Thermal conductivity of gas, Wm-1K-1
L	-Characteristics length of plate, mm
m	-Mass flow rate, kg/s
Nu	-Nusselt number
Q	-Heat transfer rate, W/m2
Re	-Reynolds number
T	-Temperature, °C
Tb	-Mean bulk temperature, °C
Tin	-Air inlet temperature, °C
Tout	-Air outlet temperature, °C
Ts	-Average surface temperature, °C

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