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Experimental Heat transfer study of Turbulent Square duct flow through V type turbulators

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Abstract: *This paper describes the experimental study of square duct with V type turbulators. The effects of inserted V-type turbulators, their attack angle on heat transfer rate, pressure drop and friction factor characteristics have been reported. All turbulators were tested under turbulent flow regime, for Reynolds numbers between 10000 and 30,000. Experiments are conducted for air with uniform heat flux condition.In the present work experimental study is carried out three different velocity flow rate results of turbulators were placed in square duct. Assorted turbulators inside square duct channel are extensively used to enhance heat transfer rate. The reason behind is that turbulators increase the fluid flow turbulence near the wall, disturb the boundary layer and also increase the heat transfer area. Details for pitch distance between turbulators (P), rib height(e) and turbulators angle are similar to experimental reference. The enhancement in heat transfer and friction factor of this turbulators inserted square duct is also compared with plain square duct under similar flow condition. The results show that 50^0 inclined V type turbulators yields about 3.6times enhancement in Nusselt number and thermal performance is 26% higher than the plain square duct.*

Keywords: *friction factor, turbulator, squareduct,heat transfer coefficient , Reynolds number, Nusselt number*

I. Introduction

The necessitate of high performance thermal system in various engineering applications need to stumble on an assortment of techniques to improve heat transfer in system. In convention area heat transfer improved by means of various augmentation techniques. It means increase the heat transfer area by protrusion, ribs and roughness. However, the thermal/hydraulic performance of the ribs can be affected by many factors including the holes in a rib, size and spacing. Saha and Mallik[1] (2005) reported an experimental investigation of the heat transfer and pressure drop characteristics of the laminar flow of an viscous oil through Square and horizontal Rectangular plain ducts and ducts inserted with full-length twisted tapes, short length twisted tapes, and frequently spaced twisted-tape elements, under constant heat flux boundary conditions. Pramanik and Saha [2](2006) studied heat transfer and the pressure drop characteristics of laminar flow of viscous oil through the square and rectangular ducts with internal transverse rib tabulators on two opposite surfaces of the ducts and integral with twisted tapes under constant heat flux conditions. Channel.S.Skullong et al.**[**3]experimentally investigated airflow friction and heat transfer features in a square channel fitted with different rib heights turbulators for the turbulent management, Reynolds number of 4000-40,000. It was found that the use of in-line ribs provides significant heat transfer enhancement, $Nu/Nu_0 = 2.6$. Nusselt number improvement tends to increase with the rise of Reynolds number. Most of past works were mainly deals with heat transfer development through circular tube maintained at a uniform wall temperature or heat flux conditions, with twisted tapes, helical screw tape inserts etc. Han *et al*.[4] carried out combined effects of the rib angle-of-attack and the channel point ratio on the local heat transfer distributions in square and rectangular channels with two opposite rib-roughened walls for Reynolds numbers from 10,000 to 60,000. The rib angle-of-attack was varied from 90^0 , 60^0, 45^0, and 30^0, whereas the corresponding channel width-to-height ratio can be varied from 1 to 2 and to 4 respectively. It was accomplished that the highest heat transfer and the accompanying highest pressure drop can be obtained at $\alpha = 60^0$ in the square channel, the highest heat transfer and the highest pressure drop occur at $\alpha = 90^0$ in the rectangular channel with a channel aspect ratio of 4. Liu *et al[5]* present an investigational study on Heat transfer characteristics in steam-cooled rectangular channels with two opposite rib-roughened walls. Most of the study have been focused on heat transfer characteristics for rib height and spacing ratio for angled, transverse, discrete or broken, square rib. In this present work, the geometric computation for 3-D turbulent flow over the W-discrete thin ribs mounted oftenly on one side of heated wall of square duct. As far as ducts of non-circular cross sections are concerned, Sekuli_c [6] and S¸ ahin [7] have presented analyses of irreversibility's associated with ducts of various shapes (namely, circular, square, equilaterally triangular, rectangular and sinusoidal) for laminar flow conditions, Heat Transfer behavior in a Square Duct with Tandem Wire Coil Element Insert .The full-length wire coil provides higher friction factor and heat transfer than the tandem wire coil elements under the same operating conditions. Khan Rezaul karim , M.A.T. Ali ,Md. Morad Hossian Mollah, M.A. R.Akhananda, and Md.A .R. Sarkar:[8] This paper deals with The turbulent flow characteristics obtained in forced convection heat transfer with asymmetric heating has been carried in developed region of square duct. The experiments have been performed for different Reynolds number varies from 50000 to 560000.It reveals that Nusselt number increases by 25.75percent varies in saw tooth backward, 27.43 percent in saw tooth forward,26.71 percent in triangular 24.43percent in trapezoidal ribbed wall over non ribbed wall with an increase of Reynolds number up to 12 percent. Cho et al. (2003) researched on the purpose of angle of attack and number of discrete ribs in a square duct and observed that the gap region between the inclined separate ribs accelerates the flow and enhances the local turbulence, which will results in an improvement in the heat transfer. They also reported that the inclined rib arrangement with a downstream gap position shows higher augmentation in heat transfer compared to that of continuous inclined rib arrangement.

The present research has been carried out to validate the results of V type turbulators to that of plain square duct for heat transfer and friction factor characteristics in a square duct with two osite walls. This study will help to show that, the V type turbulators shows better Performance than the plain square duct.

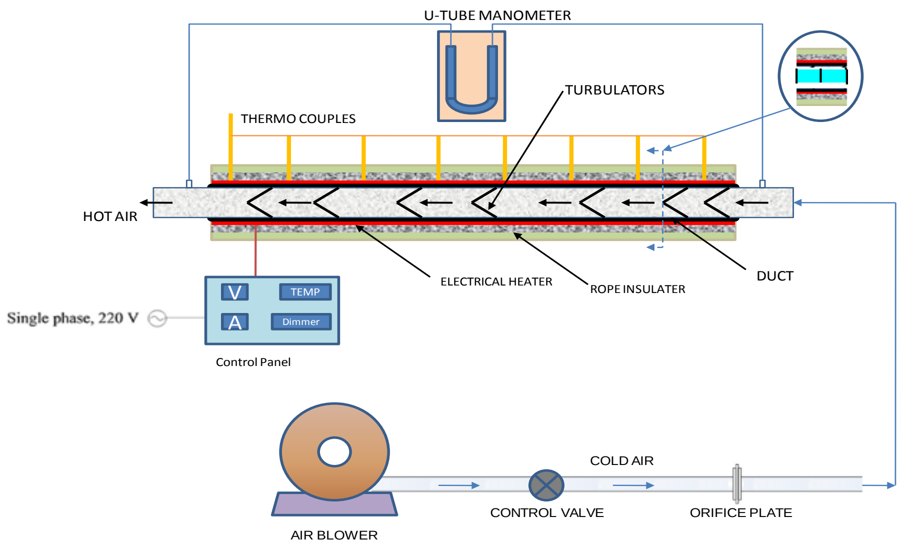

Fig 1 Schematic diagram and Experimental system

Nomenclature

Q -heat transfer rate, W
T- temperature, K
U - mean axial flow velocity, m/ s
V -voltage, V
A- Convection heat transfer area of duct, m^2
A_c -Cross-sectional area, m^2
D_h -hydraulic diameter of duct , m
C_p -specific heat capacity, J/kg K
e- turbulator height, m
p -turbulator pitch spacing
f- friction factor
H- duct height, m
h -average heat transfer coefficient, W/m^2K
k -thermal conductivity of air, W/mK
L -length of test duct, m
$\dot{m}$ -mass flow rate of air, kg/s
Nu -Nusselt number
ΔP -pressure drop P_a
Pr -Prandtl number
Re -Reynolds number

Greek letters
η -thermal enhancement factor,
α -attack angle of tape, degree (^{0})
ν -kinematics viscosity, m^2/s
ρ_a -density of air, kg/m^3

II. Experimental Program and Range of Parameters

The experimental set up consists of a square duct with entrance, test and exit sections, a blower, control valve, orifice meter, electrical heating arrangement and temperature measuring device. A schematic diagram of the test set-up is shown in Figure 1. The Aluminum square duct has an internal size of 3000 mm x 65 mm x 65 mm, which is comprises of an entrance section, a test section and an exit section of length 750 mm , 1500 mm and 750 mm respectively. The duct was characterized by the inner duct width (W) of 65 mm, height (H) of 65 mm with overall length of 3000 mm, including the test section (L) of 1500 mm. The v type turbulator elements made of a 0.5 mm thick aluminum strip were placed. In the present experimental setup a high-pressure blower (1 kW) was the source of supplying the working fluid (air) and was connected to a settling tank by a circular pipe at which an orifice plate built as per ASME standard [45] was mounted to measure the airflow rate. To maintain a constant surface heat-flux, the nichrome electrical heater was employed for heating all duct walls.

Water was used in U-tube manometer to ensure reasonably accurate measurement of the pressure. Temperature indicators are used to record the temperature of inlet and outlet test section. A calibrated orifice-meter is connected with U-tube manometer used to measure the volume air flow rate through the square duct and control valve is provided in the pipe line to connect the blower to control the flow rate of air. Square duct is made up of aluminum all walls of inner duct. In order to maintain isothermal heating condition and with high accuracy, wall temperature distributions were measured over the heated part with 16 thermocouples with 0.1°C resolution. The outer surface of the test duct was well insulated to reduce convective heat loss to surroundings, and required precautions were taken to prevent leakages from the system. The test section was heated by continuously by winding flexible electrical wire connected to the electrical power supply unit. Wood bars with low thermal conductivity, were fitted around the square duct to function as thermal barriers at the inlet and exit of the test section. Experimental data is collected under the steady- state condition for different velocity flow rate of air to give the Reynolds number in the range of 10,000 - 30,000. The illustration of the geometry of V-shaped turbulators are used in the present study is shown in Figure 1

III. Data reduction

During the test, air in the test duct received heat (Q_{air}) from the electrical heat wire mainly via the convective heat transfer mechanism. Thereby, the Q_{air} is assumed to be equal to the convective heat transfer within the duct wall which can be expressed as:

$$Q_{air} = Q_{conv} = \dot{m}\, Cp(T_O - T_i)$$

The present experiment was conducted to investigate the heat transfer augmentation in a Square Duct inserted with the V shape turbulator inserts. The experimental results on the heat transfer and friction characteristics in a square duct are first analyzed in terms of Nusselt number and friction factor.

To determine useful heat gain Q_u, heat transfer coefficient h, Reynolds number Re, Nusselt number Nu, and friction factor f, the following procedure is adopted

The mass flow rate, m, of air through the duct has been calculated from pressure drop measurement across the orifice plate.

$$m = C_d A_o [\, 2\rho_a (\Delta P)_o / 1-\beta^4]^{0.5}$$

where, C_d is the coefficient of discharge which is equal to .610.

The pressure drop $(\Delta P)_o$ across the orifice plate is given by

$$(\Delta P)_o = 9.81 \, X \, (\Delta h)_o \, X \, \rho_m \, X \sin\theta$$

The results obtained are displayed in dimensionless terms of Nusselt number and friction factor. The average heat transfer coefficients are determined by using the experimental data through the following equations:

$$h = Q_{conv} \,/\, A\, (\tilde{T}_s - T_b)$$

In which $T_b = (T_0 + T_i)/2$

$$\tilde{T}_s = \sum T_s / 14$$

The mean inner wall surface temperature (T_w) of the test duct is determined from surface temperatures of 14 stations located between the inlet and the exit of the test duct, using the following equations

$$Q_{air} = Q_{conv} = \dot{m}\, Cp(T_0 - T_i)$$

Where, A is the heat transfer surface area of duct, Ts is the local surface temperature along the duct length, and T_s is the average surface temperature.

Thus, the average Nusselt number is written as

$Nu = h\, D_h / k$

The Reynolds number based on the duct hydraulic diameter (D_h) is given by

$Re = U\, D_h / \nu$

The square duct hydraulic diameter is given by $D_h = H$

The friction factor is evaluated by

$$f = \frac{2(\Delta p)_d}{(l/D_h)/\rho U^2}$$

Where, $(\Delta P)_d$ is the pressure drop across the test duct and U is the mean air velocity in the duct. All properties of air are calculated at the bulk air temperature.

where, $(\Delta P)_d$ is the pressure drop across the square duct and is given by

$(\Delta P)_d = 9.81 \, X \, (\Delta h)_d \, X \, \rho_m$

IV. Result and Discussions

The values of friction factor and Nusselt number determined from the experimental data were compared with the values obtained with the standard Blasius equation for the friction factor (Bhatti and Shah, 1987).and Dittus-Boelter equation for the Nusselt number. The assessment of the experimental and estimated values of friction factor and Nusselt number as a function of Reynolds number is shown in figure 2 and figure 3 respectively.

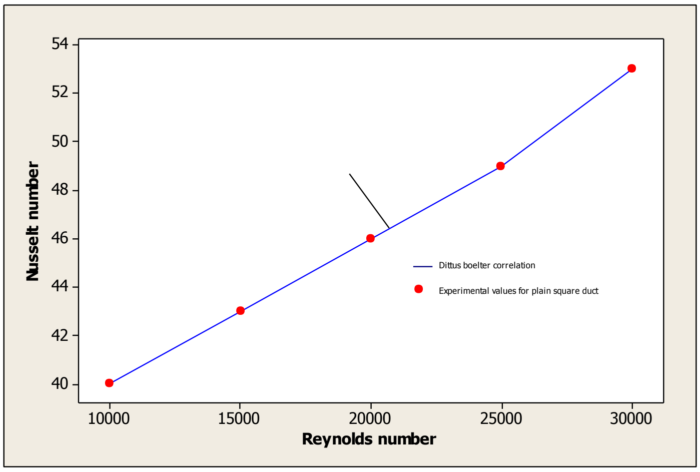

Figure 2.a

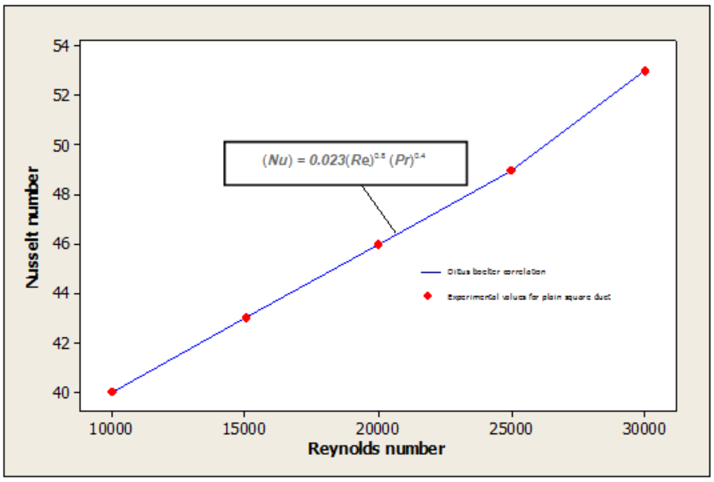

Figure 2.b

Fig 2 verification of (a) Nu and (b)f for plain square duct

Fig 2 reveals that plain square duct experimental results are correlated with standard equations of Square duct. The Nusselt number is comparatively higher at the start of heating section due to the steeper expansion of the thermal boundary layer. It is noted that *f* tends to decrease with rise of Reynolds number.

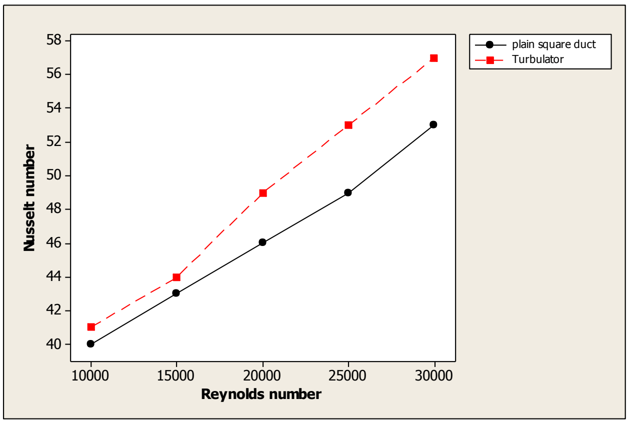

Fig3: Variation of Nu with Re for plain square duct and turbulator arrangement

It is clear from the above figure that the Nusselt number increases as Reynolds number increases and the heat transfer enhancement is about 2. 6 to 3.7 times than that of plain square duct. The structure of secondary flow by the Turbulators is mainly responsible for the increase in heat transfer coefficient.

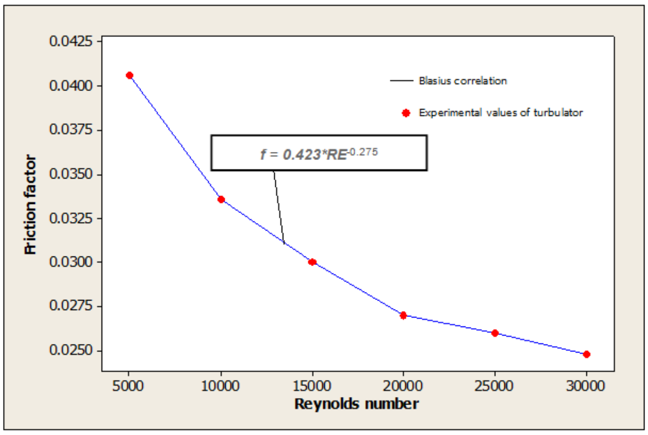

Fig 4 Effect of f with Re for Turbulator arrangement.

Fig4. Illustrates the variation of friction factor with Reynolds number for plain square duct. It is observed from the above graph that the friction factor decreases as Reynolds number increases and it remains almost constant at higher Reynolds numbers.

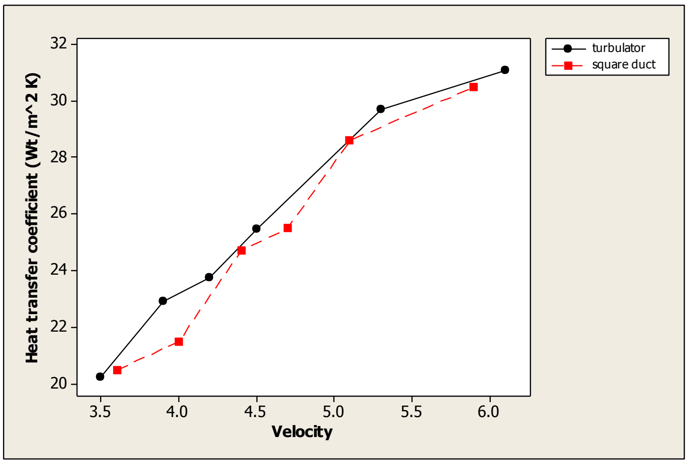

Fig: 5 Effect of turbulator angle and plain square duct at PR = 0.65 on Heat Transfer coefficient vs. Velocity.

Fig 5 reveals that velocity of air flow increases along with the heat transfer coefficient. for plain square duct. Similarly by inserting V type turbulators into the Square duct heat transfer coefficient further increased as compared to the plain square duct.

The present results on the heat transfer coefficient and velocity in uniform heat-fluxed square duct with V type turbulator inserts are presented respectively. In the above graph V type turbulator inserts Nusselt number is increased more than the plain Square duct which means higher heat transfer rate has been obtained inV type turbulator inserts rather than the plain Square Duct. Therefore, inserted duct yields the considerable heat transfer enhancement with similar trend pattern in comparison with the plain square duct.

V. Conclusions

The effect of various flow parameters on the heat transfer and friction factor characteristics in the case of square duct has been investigated. Results have also been compared with those of plain smooth square duct under similar experimental conditions to find out the augmentation in the heat transfer coefficient and friction factor. On the basis of results of heat transfer and friction characteristics of the different cases, it is observed that an enhancement in heat transfer is always accompanied with friction penalty.

An experimental investigation has been carried out with the aim to compare V type turbulators for a square duct gives maximum heat transfer with high value of thermo hydraulic performance.

The following conclusions can be drawn from this work:

1. The maximum enhancement in Nusselt number and friction factor values compared to plain square duct are of the order of 3.6 and 8.2 respectively.
2. The maximum values of Nusselt number, friction factor and thermo-hydraulic performance parameter are observed for a gap in an inclined ($\alpha = 50^0$) turbulator in the entire range of Reynolds number.
3. The Turbulator attack angle plays an important role for desirable fluid flow characteristics. The maximum value of Nusselt number is obtained for 50° due to stronger secondary flow.
4. In general, Nusselt number increases and friction factor decreases with an increase of Reynolds number. Nusselt number and friction factor are considerably higher as compared to those obtained for plain square duct. This is due to the distinct change in the fluid flow characteristics as a result of V shape turbulators that causes flow separations, reattachments and the generation of secondary flows.

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