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Experimental Investigation on Turbulent Heat Transfer, Nusselt Number and Friction Factor Characteristics of Square Ducts

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ABSTRACT: This paper presents the results of an experimental investigation of heat transfer to the airflow in the square duct. The main purpose of this study is influence of square ducts on the convective heat transfer mechanism. Under this investigation friction factor and the Nusselt Number were calculated. The reason that square ducts increase the fluid flow turbulence near the wall and also increase the heat transfer area. The results of the heat transfer are characterized in terms of the non-dimensional parameters: Nusselt number (Nu), Reynolds number (Re), and Prandtl number(P_r). The experimental measurements are carried out for Reynolds number range of10,000-50000.The heat transfer, friction factor and the pressure drop characteristics of turbulent flow of air (10,000 < Re < 50,000) through square ducts have been studied experimentally. The study shows significant improvements can be achieved with the square duct particularly in high prandtl number.

KEYWORDS: Friction factor; Nusselt number; Reynolds number; Square ducts

I. INTRODUCTION

The process of improving the performance of a heat transfer system or increase in heat transfer coefficient is referred to as heat transfer augmentation or enhancement. This leads to reduce size and cost of heat exchanger. An increase in heat transfer coefficient generally leads to additional advantage of reducing temperature driving force, which increases second law efficiency and decreases entropy generation. General techniques for enhancing heat transfer can be divided in two categories. One is passive method such as twisted tapes, helical screw tape inserts, rough surfaces, extended surfaces, additives for liquid and gases. The other is active method, which requires extra external power, for example mechanical aids, surface fluid vibration, use of electrostatic fields. Passive methods are found more inexpensive as compared to other group. Twisted tape is one of the most important members useful in laminar flow from this group .In a turbulent flow, the dominant thermal resistance is limited to a thin viscous sublayer. The wire coil insert is more effective in a turbulent flow compared with a twisted tape, because wire coil mixes the flow in the viscous sublayer near the wall quite effectively, whereas a twisted tape cannot properly mix the flow in the viscous sublayer. For a laminar flow, the dominant thermal resistance is limited to a thicker region compared with a turbulent flow. Thus, a wire coil insert is not effective in a laminar flow because it cannot mix the bulk flow well, and the reverse is true for a twisted tape insert. Hence, twisted tapes are generally preferred in laminar flow. Several techniques have been proposed in recent years and are discussed in the following sections.

A variety of duct and twisted tape geometries will be designed, developed, modeled, and compared using computer software. Ducts will be modified by changing perimeter shape, angle of twist, and tapering exit geometries. Twisted tape inserts will be modeled in a sample circular he problem of turbulent heat and fluid flows in a straight square duct (SSD) is fundamental in thermal science and fluid mechanics. The turbulence in the SSD has a remarkable change in flow structure due to the existence of the so-called Prandtl's second kind secondary flows [1]. The secondary flow has a significant effect on the

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transport of heat and momentum as uncovered by the recent large eddy simulation (LES) [2]. Extensive studies have been pursued. Examples are the work based on the algebraic stress model of turbulence [3], those emphasizing the effect of rib roughened wall [4–7], the effect of a square bar detached from the wall [8], the effect of periodic array of cubic pin–fins in a channel [9], and theeffect of inside tubes with helical fins [10]. An optimal design for ducts and twisted tape inserts that maximize heat transfer abilities will be determined for separately.Ray and Date[11] (2001) used a numerical method for predicting the heat transfer characteristics in the square duct with twisted tape insert for laminar flow region under constant heat flux conditions. Ray and Date[12] (2003) presented a numerical method for predicting the heat transfer characteristics in a square sectioned duct inserted with a twisted tape, whose width equals the length of the duct side for laminar and turbulent flow region under constant heat flux conditions. Saha and Mallik[13] (2005) reported an experimental investigation of the heat transfer and pressure drop characteristics of laminar flow of viscous oil through horizontal rectangular and square plain ducts and ducts inserted with full-length twisted tapes, short length twisted tapes, and regularly spaced twisted-tape elements, under constant heat flux boundary conditions. Pramanik and Saha [14](2006) studied heat transfer and the pressure drop characteristics of laminar flow of viscous oil through rectangular and square ducts with internal transverse rib tabulators on two opposite surfaces of the ducts and fitted with twisted tapes under constant heat flux conditions. Saha and Langille[15] (2002) reported heat transfer and pressure drop characteristics of laminar flow through circular tube with longitudinal strip inserts under uniform wall heat flux. Sarma *[16]* (2003) studied Laminar convective heat transfer with twisted tape inserts in a tube, new method is postulated to predict heat transfer coefficients with twisted tape inserts in a tube. Yadav[17] (2009) studied influence of the half length twisted tape insertion on heat transfer and pressure drop characteristics in a U bend double pipe heat exchanger An experimental investigation on turbulent heat transfer and friction loss behaviors of airflow through a constant heat-fluxed channel fitted with different heights of triangular ribs was conducted by Thianpong et al [18] with AR = 10 and height H = 30 mm with three uniform rib heights, e = 4, 6 and 8 mm and one non-uniform rib height e = 4, 6 mm alternately for a single rib pitch p = 40 mm. Comparatively the uniform rib height performs better than the corresponding non-uniform one. Md.Julker,N [19] conducted an experiment and numerical analysis in a rectangular channel with semicircular ribs with uniform height 3 and 5 mm on one wall with four different rib pitches 28, 35, 42 and 49 mm turbulence and pressure drop behaviors. Friction factor is greatly influenced by rib pitch to rib height ratio where with increasing the number of *(P/e)* ratio decreases additional pressure loss at the same Reynolds number. Caliskan and Baskaya [20] discuss which the heat transfer measurement over a surface with V-shaped ribs and Convergent-divergent shaped ribs by a circular impinging jet array was investigated using thermal infrared camera. During the experiments the Reynolds number was varied from 2000 to 10,000. channel.S.Skullong et al.**[**21]experimentally investigated airflow friction and heat transfer characteristics ina square channel fitted with different rib heights turbulators for the urbulent regime, Reynolds number of 4000-40,000. It was found that the use of in-line ribs provides considerable heat transfer augmentations, Nu/Nu0 = 2.6. Nusselt number augmentation tends to increase with the rise of Reynolds number. Most of earlier works were mainly deals with heat transfer enhancement through circular tube maintained at a uniform wall temperature or heat flux boundary conditions, with twisted tapes, helical screw tape inserts etc.

The limited work is available on heat transfer studies through square duct maintained at constant heat flux conditions. To date no attempt has been made on heat transfer enhancement through a square duct maintain at nearly uniform wall temperature conditions. Hence, this work deal with the enhancement of heat transfer studies at nearly uniform wall temperature conditions in square duct for Turbulent flow with different inserts using air as a working fluid.

II EXPERIMENTAL SETUP

Materials of the Experimental setup : Air Blower (1.5KW), GI pipe, Orifice plate, U-Tube water manometer(to measure pressure drop) ,flanges, Control valve, test section, Aluminium Square duct, Thermocouples (No:16), Voltmeter, and Ammeter .

A schematic diagram of the Experimental set up is shown in Fig.1. The Aluminum square duct has an internal size of 4500 mm x 60 mm x 60 mm, which consists of an entrance section, a test section and an exit section of length 1500 mm ($20D_h$), 1500 mm ($20D_h$) and 750 mm ($10D_h$) respectively. The exit end of the duct is connected to 50 mm internal diameter G. I. (galvanized iron) pipe provided with a calibrated orifice plate through a square to circular transition piece. The outside of entire set-up from inlet to the orifice plate, were covered with 20 mm thick thermocol (foamed polystyrene) sheet, so that the heat losses from the test section can be minimized. The plates are heated by means of separate heaters assembly, thus subjected to uniform heat flux of 0-1500 W/m^2. To measure mass flow rate of air through the duct, a calibrated orifice-meter is used which is connected with an U-tube manometer with a control valve in the pipeline. Ten thermocouples are affixed in

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test section to record the plate temperature at different locations. The airflow rate was varied to give the flow having Reynolds number in the range of 10000 to 50,000. Initially air is passed through test section and measure the temperature, voltmeter and ammeter readings calculate Reynolds number, Nusselt number , friction factor,.The wall temperature is obtained by taking average of all thermocouples installed in the duct. A groove of 1 mm depth is made on outer wall of the duct, temperature measuring probe inserted in to the groove and adhesive was applied around the groove to fix up the thermocouple to the wall. This is particularly essential to get inner wall temperature. Relevant data were noted under the steady state condition for constant surface heat flux, which was assumed to have reached when the plate and air temperatures shows negligible variation for about 10- minute duration. The steady state for each test run was obtained in about 1.5 to 2 hours. After ensuring steady state, temperature of the inlet and outlet , hot air and wall temperature were recorded throughout the experiments and isothermal pressure drop was also measured by U-tube vertical manometers as shown in Fig1.

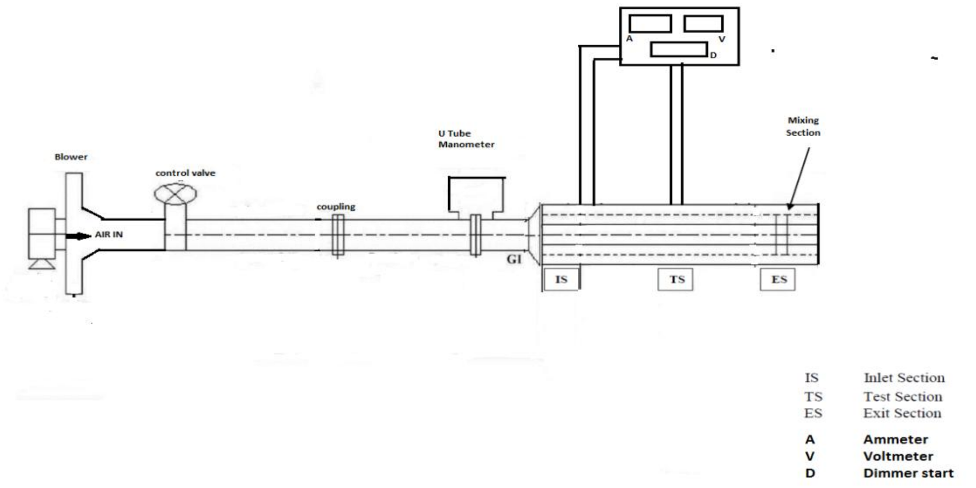

Fig 1 Schematic diagram of Experimental setup

II. EXPERIMENTAL RESULTS

. The Fig no 2 represents Comparison of experimental and predicted values of Nusselt number for Square Duct. It shows Reynolds number increases with increase of Nusselt number. The enhancement efficiency tends to increase with the increase of Reynolds number The Fig no 3 represents. Comparison of experimental and predicted values of Friction for Square Duct..It is observed that Experimental values are closer to the predicted values.

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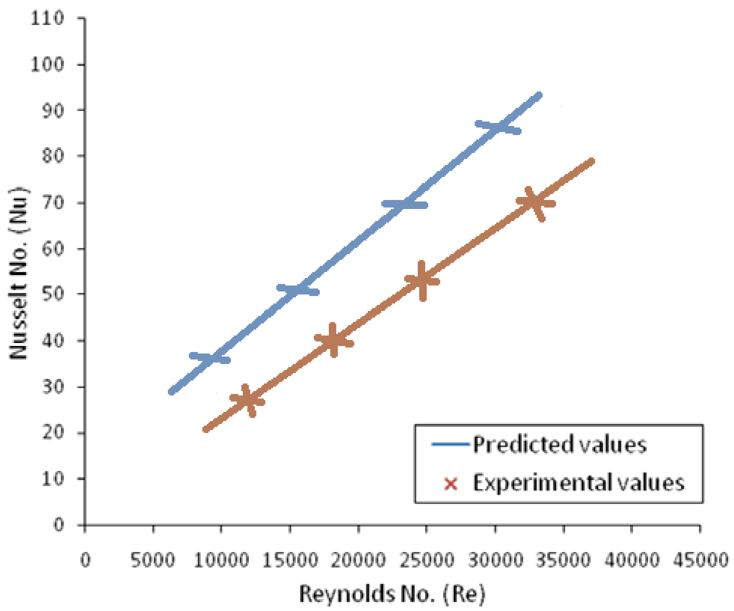

Fig.2.

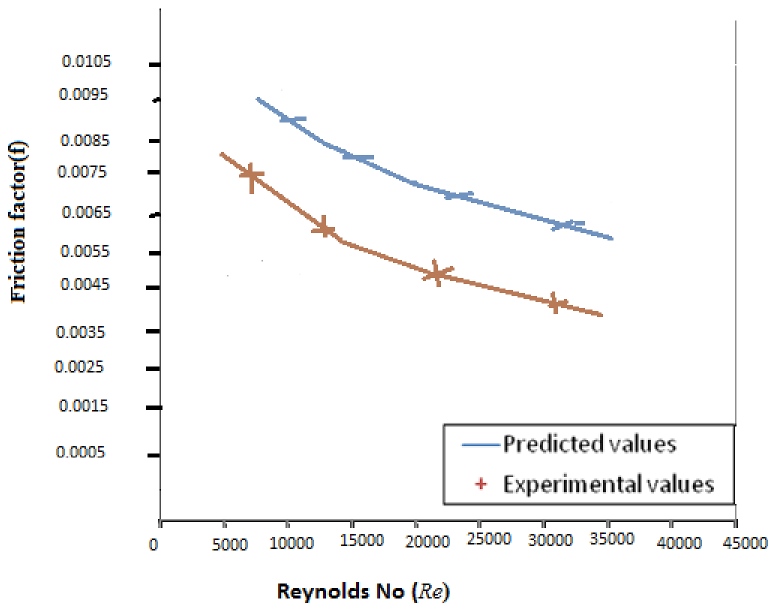

Fig. 3.

IV DATA REDUCTION

The objective of this experiment is to investigate the Nusselt number and friction factor in the square ducts. The Reynolds number is given by $\mathbf{R_e} = \rho_a \mathbf{V} \mathbf{D_h} / \mu_a$

where, $\mathbf{V} = \mathbf{m} / \rho_a\, WH$

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

$\mathbf{m} = C_d A_o[2\rho_a c_o)/1-\beta^4]^{0.}$

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The pressure drop **(ΔP)o** across the orifice plate is given by $(\Delta P)_o = 9.81 \times (\Delta h)_o \times \rho_m$ (4)
The average heat transfer coefficient, **h** is evaluated from the measured temperatures and heat inputs.

$$Q_u = m\, C_p\, (T_o - T_i)$$

$$h = Q_u / A_p (T_p - T_f)$$

$$T_f = (T_i + T_o)/2$$

$$T_p = \sum_{n=1}^{10} T_n/10$$

Then, average Nusselt number, N_u, is written as $N_u = hD_h / k_a$
The friction factor is determined from the measured values of pressure drop, $(\Delta P)_d$ across the test section length, between the two points $(\Delta P)_d = 9.81 \times (\Delta h) \times \rho_m$

$$f = 2(.(\Delta P)d\, D_h \;/\; 4\, \rho_a\, L_f\, V^2$$

V. NOMENCLATURE

D Diameter, m
D_h Hydraulic diameter, m
N_u Nusselt number
R_e Reynolds number
p_r Prandtl number
V Velocity, m/s
AR Aspect ratio
C_p specific heat at constant pressure (kJ/kg/K)
Gr Grashof number,
g gravitational acceleration (m/s^2)
f friction factor
I current, amp
k thermal conductivity of fluid,W/mK
U average axial flow velocity,m/s
V voltage, V
Qu Useful Heat gain, W
T_f Bulk mean air temperature, 0C or K
T_i Air inlet temperature, 0C or K
T_o Air outlet temperature, 0C or K
Tp Mean plate temperature, 0C or K
V Velocity of air, m/s
W Width of duct, m
H Depth of duct, m
h Convective heat transfer coefficient, W/m2 K
ka Thermal conductivity of air, W/m K
L Test section length, m
L_f Duct length for calculation of friction factor, m
L/D*h* Test length to hydraulic diameter ratio
m mass flow rate, kg/s
N_{uo} Nusselt number for smooth circular duct
N_{ur} Nusselt number for roughened duct
N_{ur}/N_{uo} Nusselt number ratio
(ΔP)d Pressure drop in the test channel, N/m2
(ΔP)o Pressure drop across the orifice plate, N/m2
Greek Symbols
β Ratio of orifice diameter to pipe diameter
μ_a Dynamic viscosity of air, N s m-2

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ρ_a Density of air, kg m-3
ρ_m Density of manometer fluid, kg m-3
η heat transfer enhancement efficiency

VI .CONCLUSIONS

The mean Nusselt number increased at about 112% when compared with those from correlations of Dittus–Boelter.. The boundary layer along the duct wall would be thinner with the increase of radial swirl and pressure resulting in more heat flow through the working medium. Furthermore, the swirl enhances the flow turbulence, which led to even better convection heat transfer. Thus, the higher Reynolds numbers the greater Nusselt number. Based on the calculations of the errors in the experimental Measurements by various instruments used, the uncertainties in the calculated values of Reynolds number, Nusselt number and Friction factor are estimated as ±2.45%, 2.82 %, ± 3.67 % respectively .

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