Proceedings of the
International Conference on Mechanical Engineering 2007
(ICME2007) 29- 31 December 2007, Dhaka, Bangladesh

ICME07-TH-37

FORCED CONVECTIVE HEAT TRANSFER AND FRICTION FACTOR IN AN ASYMMETRICALLY HEATED SMOOTH SQUARE DUCT

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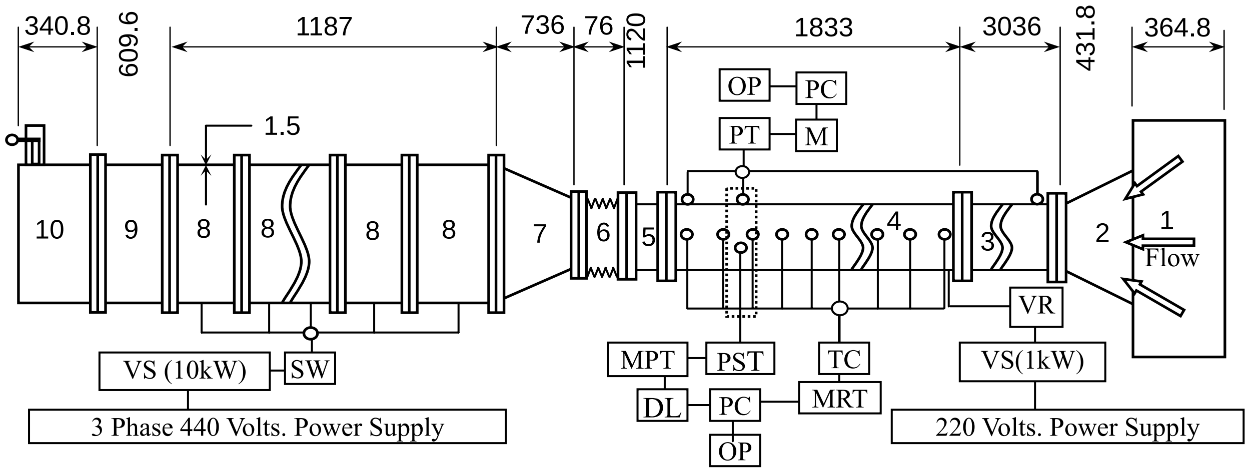
ABSTRACT
The forced convective heat transfer is studied experimentally and the measurements are presented of the distributions of local as well as average heat transfer coefficients and the friction factors for a fully developed turbulent flow in an asymmetrically heated smooth square duct at constant heat flux as a boundary condition. The Nusselt number, the friction factor and the Stanton number have been calculated for ten different Reynolds numbers over the range of $5\times10^4 < Re < 1\times10^5$. The effects Reynolds number and locations across the duct on the distributions of local as well as average heat transfer coefficient and friction factor are studied. Previously, it was assumed that the Prandl number is constant but in the present experimental investigation it has been found that it decreases with increase of both Re as well as locations across duct from centre towards the walls. The local data in the fully developed region are averaged and correlated with Prandl number as variable parameter. The results compared well with the published data for Nusselt number and Stanton number except friction factor. In the present investigation the friction factor obtained increases with the increase of Reynolds number instead of published data where it decreases or approaches a constant value for smooth noncircular ducts. The results are presented in their final concise form of compact correlations that involve dimensionless groups which represent the characteristics of heat transfer and friction factors. The correlations can be used for improved numerical analysis and for better design of heat transfer equipments for engineering applications. The secondary flow pattern in the duct is reflected in the local distributions of the Nusselt number, the friction factor and the Stanton number the values of which on the heated wall of smooth square duct are 1.035 to 1.225 (3.53% to 22.45%), 1.07 to 1.17 (7.18% to 16.78%), and 1.17 to 1.20 (16.81% to 20.24%) times higher than those of smooth circular duct

Keywords: Heat Transfer, Friction Factor, Smooth Square Duct.

1. INTRODUCTION
In the field of fluid mechanics and heat transfer many physical properties of any fluid along with boundary conditions viz. velocity, temperature, density, viscosity, pressure, conductivity, heat transfer coefficient, surface geometry, etc. are involved at a time to represent a single case of fluid flow or heat transfer or both. During the dynamic process of heat flow these properties vary accordingly to keep the process going. The several variables as some of them mentioned above are combined into a few dimensionless parameters and the results are presented in the form of empirical equations that involve both individual variable parameters and dimensionless groups. This technique considerably reduces the number of variables involved in the experimental. An increase in heat transfer is accompanied by an increase in the pressure drop of the air flow. Many investigators have developed correlations on heat transfer and friction factor dependence on Reynolds number and many other parameters but they have assumed constant Prandl number. It has been recently found that in exactly the same experimental set up and configuration with constant heat flux boundary condition the Prandl number does not remain constant [1] and[5] but it rather decreases with increase of Reynolds number as well as with increase of location from centre towards the side wall. Hence the enough data and the effects of Reynolds number, locations across the duct, pressure drop etc. on Prandl number are very rare in literature. In view of the above discussion, a need therefore exist for an experimental investigation to incorporate Prandl number as an important variable parameter to obtain improve correlations for correct analysis of heat transfer problems in engineering applications.

2. LITERATURE SURVEY
The turbulent flow as well as the temperature field in non-circular ducts are influenced by the existence of the secondary flow [7] , [8], [10] and [12]. Though the



LEGEND All Dimension are in mm

1. Air Filter	8. Fans 9. Silencer	DL = Data Logger
2. Inlet Contractor	10. Butter Fly	PC = Personal Computer
3. Unheated Duct	TC = Thermo-couple	OP = Out-put
4. Heated Test Duct	DTR = Digital Temperature Recorder	PT = Pitot Tube
5. Unheated Duct	PST = Pilot Static Tube	SW = Switch
6. Bellow	MPT = Micro Pressure Transducer	VS = Voltage Stabilizer
7. Diffuser	M = Manometer	VR = Voltage Regulator

Fig: 1 Schematic Diagram of the Setup

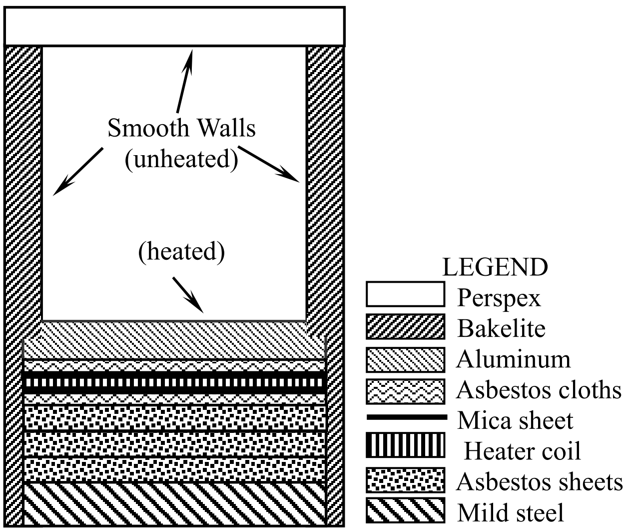


Fig: 2 Illustrating the Cross sectional View of the Duct

velocity of this secondary flow is a small percentage of the primary flow velocity, of the order of 2 to 3 percent, its influence on the flow and temperature fields in the duct can not be ignored, Ali, [7] [8] and [19]. This is the reason why these flows and temperature fields have attracted interest not only for the light they shed on fluid dynamics, but also in relation to the augmentation of heat transfer [11] [14] and [15]. Although, in ducts, secondary flow would affect the characteristics of the forced convective heat transfer, as has already been found by many investigators, there are only a a few reports on fundamental ducts [1], [5], [6], [8], [17]. The present reports is concerned with the turbulent flow in an asymmetrically heatd smooth square duct with constant heat flux as a boundary condition. Due to lack of useful

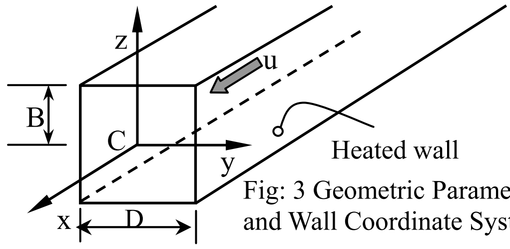


Fig: 3 Geometric Parameters and Wall Coordinate System

the heat transfer and the friction factor in the asymmetrically heated smooth square duct with constant heat flux as boundary condition.

3. EXPERIMENTAL SET UP AND METHODOLOGY

A schematic diagram of the straight experimental setup of length 9735 mm is illustrated in the Fig: 1. For detail description refer to [1].

4. MEASUREMENT SYSTEM

The configuration, the dimensions of the test model, the flow direction, and the coordinate system are schematically shown in Fig: 3. For detail description refer to [1]and [4].

5. DATA REDUCTION

The mean values of time mean velocity and temperature are calculated by integration of the local time mean velocity and temperature profile curve divided by the total length of the curve along the abscissa [1] and [2]. The net heat transfer rate can be calculated from

$$q = Q/A_c = C_p G(\Delta T_{ln}) \qquad (1)$$

where, $\Delta T_{ln} = (T_{bo} - T_{bi}) / \ln[(T_{wc} - T_{bi}) / (T_{wc} - T_{bo})]$ (2)

The Log Mean temperature difference of air, Eqn. (2) is used in Eqn. (1) to obtain the net heat transfer rate. The local outer wall temperature T_w is read from the thermocouple output. The corrected local inner wall temperature, T_{wc} for Eqn. (1) is calculated by one dimensional heat conduction equation as:

$$T_{wc} = T_w - (Q\delta/k A_s) \quad (3)$$

The average value of local heat transfer coefficient h is evaluated from:

$$h = q/(T_{wc}-T_b) \quad (4)$$

The coordinate y indicating the location of probe position for measurement are nondimensionalized by the half width of the duct, B=D/2 as y/B. The flow velocity recorded by data logger in milliovlts is converted to velocity in (m/s) and pressure drop in (N/m^2) by calibration equations. Thermal conductivity depends on temperature. Since the air velocity and temperature varies along the duct, all the air properties and related parameters are calculated at the bulk mean air temperature, $T_b = ½(T_o + T_i)$ and bulk mean air velocity, $u_b = ½(u_o + u_i)$ [1], [3] and [15].

The local Nusselt number, Nu, friction factor, and Stanton umber are calculated from the following relations as:

$$Nu_s = hD/K_f \quad (5)$$

$$f_s = (\Delta p/\Delta L)\times D/[(\rho\ \bar{u}^2)/2] \quad (6)$$

$$St_s=h/[\rho c_p u_b] \quad (7)$$

6. DATA ANALYSIS

The duct is heated asymmetrically at the bottom wall. It is symmetric about z-axis but asymmetric about the y-axis. So the measurements are taken only in the one half of the cross section of the duct about the symmetrical axis as shown in Fig: 4, which represent the flow characteristics of the entire duct. Axial time mean velocity and temperature are measured at x = 34.5D downstream from the heated test duct entrance where both the velocity and the temperature are fully developed, [15]. Thus, at the section x=34.5D downstream from the Fig: 4 Dots represents probe positions in space for measurements of velocity and temperature from leading edge of the heated section i.e. x=94.56D from the uniform than that of the square ducts due to symmetrical unheated section. In this position both velocity and temperature fields can be considered to be fully developed, [6]; [15] and [16]. The time mean velocity and temperatures of air are measured within the region of $-25 < z < 25$ and $0 \le y \le 23$ at 7 different locations of y = 0, 3, 8, 13, 18, 21, and 23 in the cross section. The time mean velocity and the temperature are calculated from the probability distribution function of the measurements recorded by the data logger. There are typically 105 measurement points in space at each measuring location i.e., 105×7=735 points in space for half of the cross section of the duct which represents the data for the entire duct cross section [8] and [15].

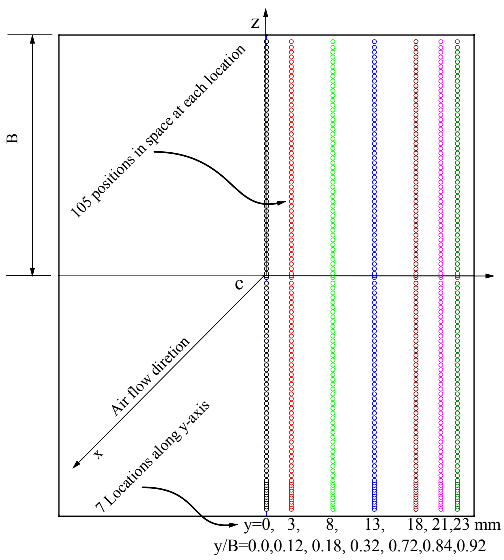


Fig 4: Dots represents probe positions in space for measurements of velocity and temperature.

The corresponding statistical error is between 0.5 to 2 percent in the time mean velocity and between 1.3 to 2.2 percent in the temperature. The scattering of the wall temperature measurement is found to be between 2.1 to 3.4 percent and the uniformity of the wall temperature distribution is considered to be satisfactory, [18]. The time velocity measurements are repeated whenever error or doubtful situations occurred to ensure that the measured results are repeatable.

7. RESULTS AND DISCUSSIONS

The experimental results concerning a time mean velocity and temperature fields obtained for a turbulent flow through an asymmetrically heated smooth square duct with constant heat flux as the boundary condition are discussed briefly. The longitudinally constant heat flux boundary condition of the present investigation, thermally fully developed region is characterized by wall and air temperature that increases linearly as a function of longitudinal positions, [5], [6], and [7]. The measurements are taken for 10 Reynolds number varying between $5\times10^4<Re<1\times10^5$.

7.1 Nusselt Number

Fig: 5(a) displays the dependence of local Nusselt number on both Reynolds number and locations of y/B increasing from centre towards the side walls. The nature of variation of Nu with y/B is exactly the same as that of local variation of mean air temperature [5]. The correlations obtained are expressed as follows:

$$\bar{N}u_s =175.35+47.82(y/B)^2-78.15(y/B)3 \quad (8)$$

$$Nu_s = 0.046\ Re^{0.753}\ Pr^{0.4} \quad (9)$$

With increase of Reynolds number both the Nesselts number curve shifts up in a similar manner expected as that of temperature of air [4]. The variation of local Nusselt number across the test duct is shown in Fig: 5(b). At constant pressure drop local Nusselt number decrease linearly with increase of local Reynolds number near the side walls in the region $y/B \ge \pm0.52$ but, near the centre in the region $y/B \ge \pm0.52$ the local Nusselt number decreases with Reynolds number remaining almost constant. This reflects the effect of secondary flow in the region $0.72\ge y/B\ge 0.12$. Also with increase of pressure drop the curve shifts up as wells right. This fact is clearly displayed in the Fig: 6 where both the local and average Nusselt number increases linearly with increase of Reynolds number and with change of location of probe

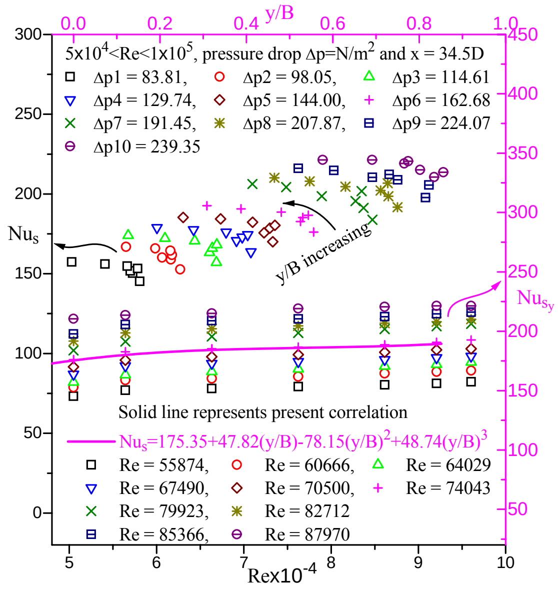


Fig 5(a): Effects of local Reynolds number on local Nusselt Number across duct at.

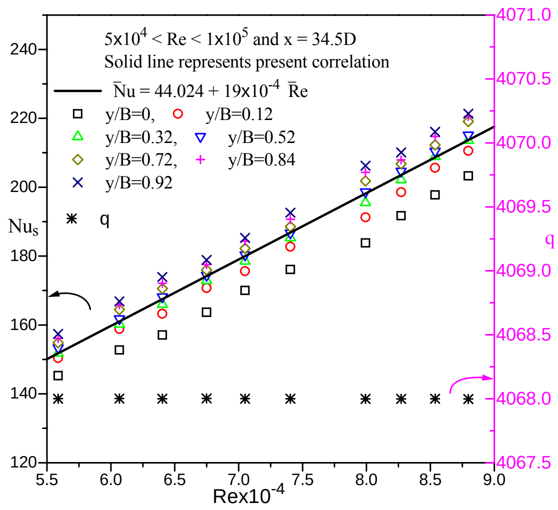


Fig 5(b): Effects of Reynolds number on local Nusselt number and heat flux across duct.

position from centre towards the side walls the curve shifts up. In the circular pipes the flow is much more boundary conditions and absence of secondary velocity, where as in the square ducts used in the present experiments the anisotropic boundary conditions due to the presence of corners of the ducts produce secondary velocity in the main flow field increasing the mixing up of air and the turbulent intensity which intern enhance the heat transfer rate. Previously, it was assumed that the Prandl number is constant but in the present experimental investigation it has been found that it decreases with increase of both Re as well as locations across duct from centre towards the walls. The local data in the fully developed region are averaged and correlated with Prandl number as variable parameter. The results compared well with the published data for Nusselt number and Stanton number except friction factor. In the present investigation the friction factor obtained

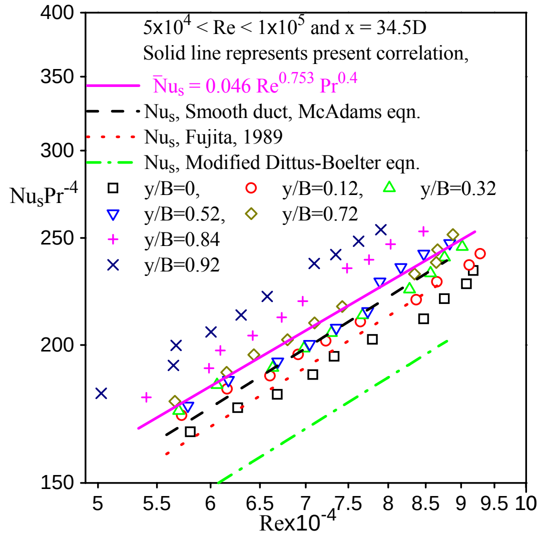


Fig 6: Distributions of local and average Nusselt number across duct.

increases with the increase of Reynolds number instead of published data where it decreases or approaches a constant value for smooth noncircular ducts. The results are presented in their final concise form of compact correlations that involve dimensionless groups which represent the characteristics of heat transfer and friction factors. The correlations can be used for improved numerical analysis and for better design of heat transfer equipments for engineering applications. The secondary flow pattern in the duct is reflected in the local distributions of the Nusselt number, the friction factor and the Stanton number the values of which on the heated wall of smooth square duct are 1.035 to 1.225 (3.53% to 22.45%), 1.07 to 1.17 (7.18% to 16.78%), and 1.17 to 1.20 (16.81% to 20.24%) times higher than those of smooth circular duct.

7.2 Friction Factor and Stanton number

The Reynolds number and the locations across the duct dependence of both local friction factor and Stanton number for fully developed flows with constant heat flux are depicted in Fig: 7(a) and 7(b). Fig: 7(a) shows the distributions of local friction factor and Stanton number at constant pressure drop in a logarithmic plot. Both the friction factor and the Stanton number drop linearly with increase of location from centre towards the side walls because of the corresponding decrease in air velocity [4]. The figure also shows that with the increase of the pressure drop both the local friction factor and Stanton number curves shifts towards right but the corresponding values of friction factor and Stanton number gradually increasing and decreasing respectively with increase of y/B. This trend is clearly visible in Fig: 7(b), where both the friction factor and Stanton number increase with the increase of y/B at constant Reynolds number. The graph also shows that with the increase pressure drop the curves of friction factor and Stanton number shift up words and down wards respectively. These variations of friction factor and Stanton number can be expressed as follows:

$$\bar{f}_s = 0.12\text{x}10^{-4}\ \text{Re}^{0.037} \quad (9)$$

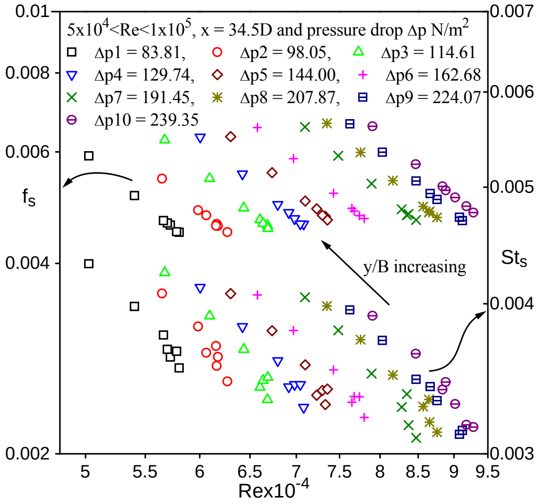


Fig 7(a): Effects of local Reynolds number On friction factor and Stanton number across duct.

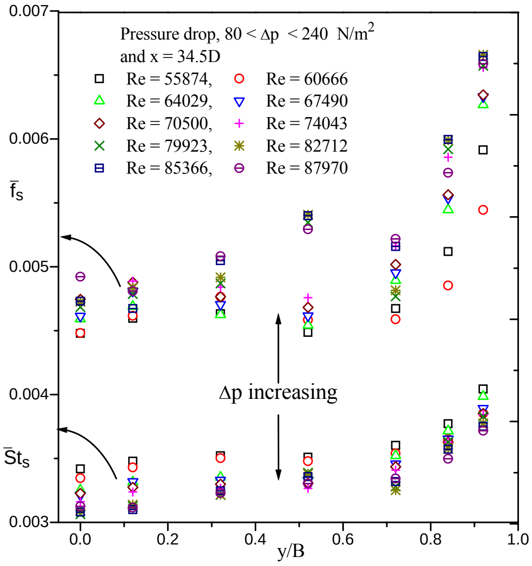


Fig 7(b): Comparisons of local friction factor and Stanton number across duct.

$$\text{and } St_s = 1.342 \times 10^3\, Re^{1.353} \qquad (10)$$

$$\bar{f}_s = 0.0032\, Re^{0.234} \qquad (11)$$

$$\text{and } \bar{St}_s = 0.0055\, Re^{-0.22} \qquad (12)$$

Finally the effect of Re on average of local f and St is shown in Fig: 8, where the well known published data also incorporated for comparison. Comparing with the published data results obtained in the present experimental investigation show that instead of decreasing f increases with the increase of Re. Exactly the same experimental set up and boundary conditions the results obtained for both Nusselt number and Stanton number shown in Fig: 6 and Fig: 8 respectively agree well with published data but the results obtained for friction factor does not agree. With the increase of Reynolds number the viscosity of air increases [5] the increased heat transfer in the asymmetrically heated duct is achieved at the expense of increased friction due to increase of both primary as well as secondary flow of air

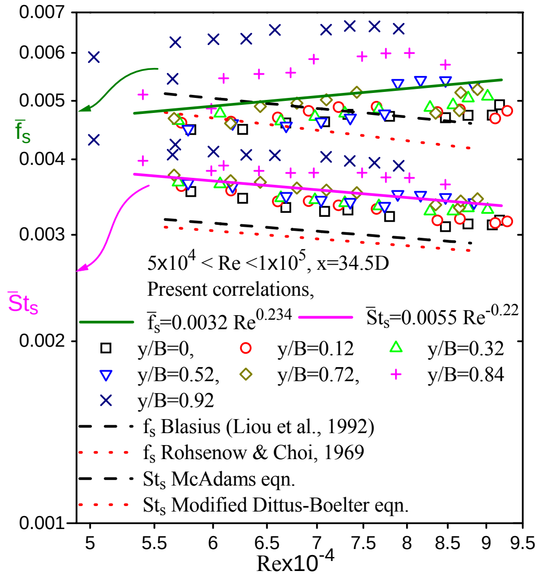


Fig 8: Distributions local and average friction factor and Stanton number across duct.

flow. Since the viscosity is increasing with the increase of Reynolds number the friction factor must also increase. Also it can be seen in the Darcy's formula, $f=((\Delta p/\Delta L)D)/(\frac{1}{2}\rho\upsilon^2)$, $f \propto \Delta p/u^2$, assuming all other parameters are constant, the ratio of $\Delta p/u^2$ increases with the increase of pressure drop. Thus the tendency of friction factor to decrease with the increase of Reynolds number is not a valid statement and hence not acceptable for the improved analysis and for better design of heat transfer equipments for engineering applications.

8. ACKNOWLEDGMENT

This is a part of PhD woks carried out at BUET, Dhaka, by the first author under the guidance of the second and third authors. The first author is grateful BUET authorities and staffs, the panel expert referees, especially the second author for their comments and suggestions, which led to substantial improvement of this work.

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10. NOMENCLATURE

Symbol	Meaning	Unit
A	Area	m^2
B	Half of width of duct	m
C	Specific heat, Centre	$W.s/kg^{O}C$
D	Hydraulic diameter of duct	m
f	Friction factor	Dimensionless
G	Mass flux	Kg/m^2s
h	Heat transfer coefficient	$W/(m^2, {}^0C)$
K	Thermal conductivity	$W/(m, {}^0C)$
L	Length	m
Nu	Nusselt number	Dimensionless
P	Pressure	N/m^2
Q	Heat transfer	W
q	Heat flux	W/m^2
Re	Reynolds' number	Dimensionless
St	Stanton number	m/s
T	Mean temperature	${}^0C$
u	Time mean velocity	m/s
u^+	Time meanuniversal velocity	Dimensionless
x, y, z	Coordinate system defined in Fig: 3 and 4	
Greek Symbols		
τ	Shear stress	N/m^2
ν	Kinematics iscosity	m^2/s
ρ	Density	kg/m^3