Dept. of Mech. Eng.
CUET

Published Online April 2017 (http://www.cuet.ac.bd/merj/index.html)

Mechanical Engineering Research Journal

Vol. 10, pp. 71-77, 2016

ISSN: 1990-5491

EXPERIMENTAL STUDIES ON HEAT TRANSFER AND FRICTION FACTOR IN A TURBULENT FLOW THROUGH SMOOTH SQUARE DUCT

A. K. M. Abdul Hamid[1*], A. Razzaq Akanda[2] and M. A. Taher Ali[3]

[1]Department of Mechanical Engineering, RUET, Rajshahi-6204, Bangladesh
[2]Sonargoan University, 71, Kawranbazer, Tejgaon, Dhaka-1215, Bangladesh
[3]Mechanical Engineering Department, BUET, Dhaka-1000, Bangladesh

Abstract: The forced convective heat transfer is studied experimentally and the measurements are presented as 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 of Reynolds number and location of positions of y/B across the duct on the distributions of local as well as average of mean heat transfer coefficient and friction factor are studied. Previously, for the analysis of results almost every investigator assumed that the Prandtl number as constant parameter but in the present experimental investigation the local data in the fully developed region are averaged and correlated with Prandtl 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 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. 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 equipment for engineering applications.

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

NOMENCLATURE

A = Area	m^2,	
B = Half of width of duct	M,	
C = Specific heat, Centre	W.s/kg^{0}C,	
D = Hydraulic diameter of duct	M,	
G = Mass flux	Kg/m^2s,	
h = Heat transfer coefficient	W/m^2 ^{0}C,	
k = Thermal conductivity	W/m ^{0}C,	
L = Length	m,	
P = Pressure	N/m^2,	
Pr = Prandtl number,	Dimensionless,	
Q = Heat transfer,	W,	
q = Heat flux,	W/m^2,	
u = Time mean velocity,	m/s,	
Nu = Nuselt number	Dimensionless,	
St = Stanton number	Dimensionless,	
Re = Reynolds' number,	Dimensionless,	
T = Mean temperature,	^{0}C,	
Pr = Prandtl number	Dimensionless,	
x, y, z = Coordinate system, Figs. 3 and 4.		

Greek Letters

ν = Kinematics viscosity	m^2/s
ρ = Density	kg/m^3

* **Corresponding author:** Email: abulkmdahamid@gmail.com,

Subscripts

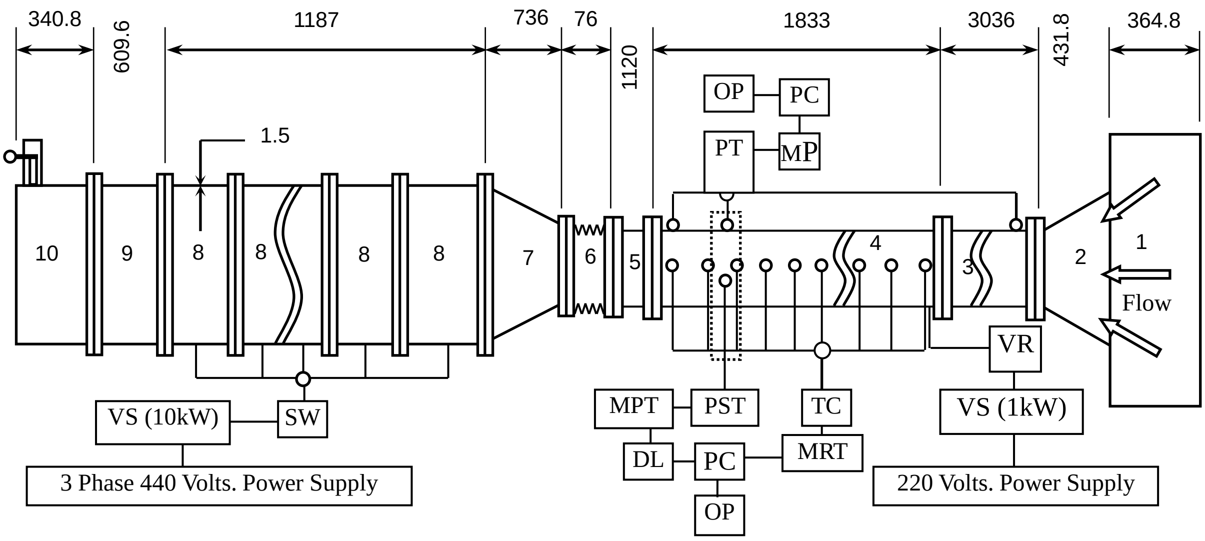
b = Bulk mean,	i = Inlet,
c = Centre, correct ,	o = Outlet,
L = Loss,	w = wall
m = Mean,	s = Smooth duct, surface,
LM = Log Mean	ln = log

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 i.e., increase in friction factor. Many investigators have developed correlations on heat transfer, friction factor, Prandtl number does not remain constant [1,2] but it rather decreases with increase of Reynolds number as well as with increase of location of positions y/B from centre towards the Stanton number etc., assumed constant Prandtl number. Hence the enough data and the effects of Reynolds number, locations across the duct, pressure drop including Prandtl number is very rare in literature. In view of the above discussion, a need therefore exist for an experimental investigation to incorporate Prandtl 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 is influenced by the existence of the secondary flow [5,7]. Though the 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 cannot be ignored, [5,12]. 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 [9]. 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 few reports on fundamental ducts [1,4,11]. The present reports are concerned with the turbulent flow in an asymmetrically heated smooth square duct with constant heat flux as a boundary condition. Due to lack of useful data on temperature measurements we conducted such measurements, the results of which are presented here in order to examine the effects of secondary flow on asymmetrically heated smooth square duct with constant heat flux as boundary condition. The existing correlations on heat transfer for turbulent flow of air through ducts were obtained assuming Prandtl number as a



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 experimental setup.

constant parameter. The present investigation shows that the Prandtl number drops with increase of both temperature of flowing air through the duct and the Reynolds number. It also drops with location of positions from the centre towards the side walls of the duct [2].

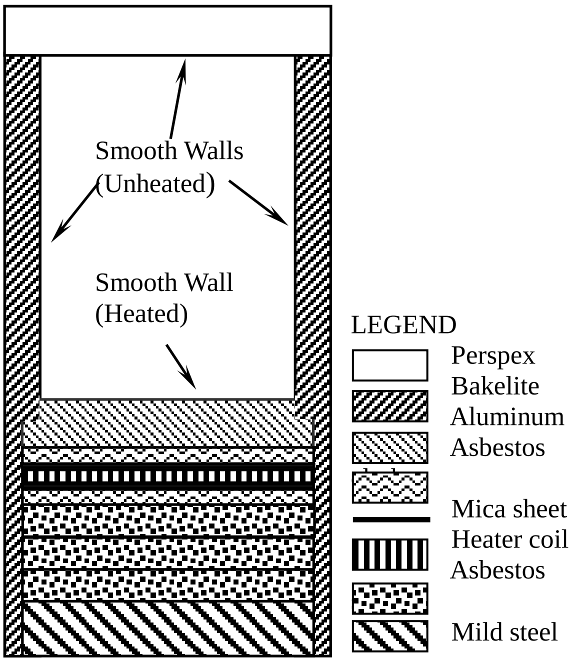


Fig. 2 Illustrating the Cross sectional View of the Duct.

3. EXPERIMENTAL SET UP AND METHODOLOGY

The experimental set up has been designed and instruments and probes are installed in it and these are connected with a high speed digital computer [1]. A schematic diagram of the straight experimental setup of length 9735 mm is illustrated in the Fig. 1. The test square duct of length 5989 mm having cross section area of 50 mm×50 mm consists of heated section of length 1833 mm (= 36.66D) and unheated section of 3036 mm (= 60.06D). The unheated section of the duct serves to establish hydro-dynamically fully developed flow at the entrance to the heated section. In order to minimize any possible end effects to be transmitted at the test sections, a smooth duct of 1120 mm long having the same cross sectional dimensions as that of test duct is attached at the downstream end of the heated test section. Fig. 2 illustrates the details of the cross sectional view of the test section. Two side walls of the entire test duct are made from Bakelite sheet of 12 mm thick to provide both the high strength and to ensure no leakage of current. The top wall is made from transparent Plexiglas plate of thickness 12 mm in order to provide optical access for observation and necessary adjustment of probes. The entire test duct except the top wall is enclosed in glass wool to minimize heat loss. The filtered air at room temperature is drawn into the straight square test duct through the air filter followed by inlet parabolic nozzle in order to establish uniform velocity. Only the bottom wall is heated electrically. To maintain a constant heat flux a voltage stabilizer followed by a voltage regulator, both having 1kW capacity is used for constant power supply to the heater. The flat nichrome wire of size 28 SWG having the resistance of 9.8097 Ω/m is used to achieve uniform wall eat flux conditions. As the thermocouple is attached intrinsically with the pitot static tube measurements of both the velocity and temperature of the air flowing through the duct is taken simultaneously. The bottom wall temperatures of the test section are measured by 8 copper constant thermocouples distributed along the entire length of the heated test duct. Thermocouples also are used to measure the bulk mean air temperature entering and leaving the test section. Two pressure tapings (one at x = 0.20D and the other at x = 94.56D i.e., x = 34.5D from the leading edge of the heated test section) are used for the static pressure drop measurement across the heated test section.

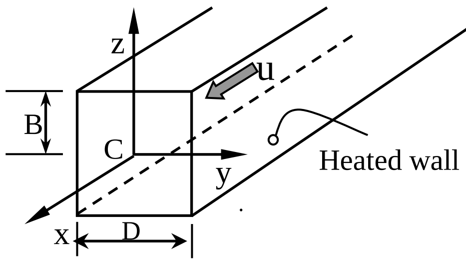


Fig. 3 Geometric Parameters and Wall Coordinate System.

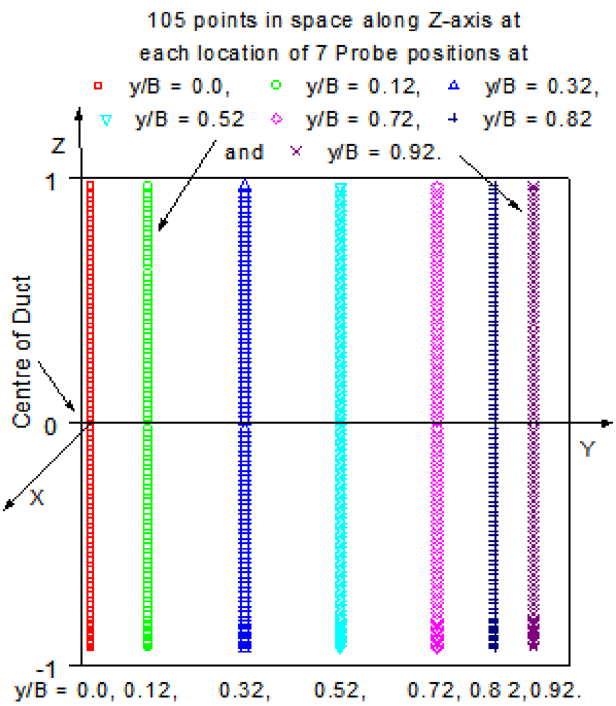


Fig. 4 Illustrating 7 Probe position along Y-axis for measuring Velocity and Temperature.

4. MEASUREMENT SYSTEM

The configuration, the dimensions of the test model, the flow direction, and the coordinate system are schematically shown in Fig. 3. Fig. 4 illustrates that the air velocities and temperatures were measured at 7 different positions of y/B = 0, 0.12, 0.32, 0.52, 0.72, 0.84, and 0.92 along direction of ± Z-axis from bottom wall to top wall i.e. perpendicular to bottom wall within the range of $-1 \leq Z/B \leq 1$.

The mean velocity and the temperature were calculated from the probability distribution function of the measurements recorded by data logger. There were typically 105 measurement locations within the range of $-1 \leq Z/B \leq 1$ at each measuring position. Thus at each Reynolds number typically 735 locations were selected for the measurement of both velocity and temperature of air flowing. The mean velocity and the static pressure are measured by the United Senser (USA) pitot static tube of 1.6 mm outer diameter with a Furnace Controls Ltd. (U.K.), pressure transducer (model MDC FC001 and FC012 and a Keithly (USA) digital micro-voltmeter with a data logger system (model 2426). The signals of the pitot static tube are transmitted to pressure transducer through 1.4 mm bore flexible tygon tubing. The signals of the digital micro-voltmeter correspond to the velocity head of the pitot static tube. Before starting the experiment, the output voltage for the pressure transducer for different range of scale is calibrated in the calibration rig with a Dwyer (USA) slack vertical water tube manometer for the velocity and with an Ellison (USA) inclined manometer, using Kerosene of specific gravity 0.7934, for static pressure measurement. The output voltage is found to vary linearly with pressure in the measurement range. For the measurement of all signals with micro-voltmeter, integration times of about 30 seconds are used. Thermocouples are used to measure both wall and air temperature.

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]. The net heat transfer rate can be calculated from,

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

where,

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

The log mean temperature difference of air, Eq. (2) is used in Eq. (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 En. (1) is calculated by one dimensional heat conduction equation as:

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

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

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

The coordinate y indicating the location of probe position for measurement are non-dimensionalized 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,

$u_b = ½(u_o + u_i)$.

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

$$Nu_s = hD/K_f \tag{5}$$

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

$$St_s = h/\,[\rho c_p u_b] \tag{7}$$

6. DATA ANALYSIS

Since the duct is heated asymmetrically at the bottom wall only, it is symmetric about z-axis but asymmetric about the y-axis. The measurements are taken only in one half of the cross section about the symmetrical axis as shown in Fig. 4, which represent the flow characteristics of the entire duct. Measurements are made at the section x = 34.5D downstream from the leading edge of the heated section i.e. x = 94.56D from the unheated section. In this position both velocity and temperature fields can be considered to be fully developed, [10]. The time mean velocity and temperatures of air are measured within the region of $-25<z<25$ (i.e. $-1<z/B<1$) and $0\leq y\leq 23$ (i.e.$0\leq y/B\leq$) at 7 different locations of y = 0, 3, 8, 13, 18, 21, and 23 (i.e., y/B = 0.0, 0.12, 0.32, 0.52, 0.72, 0.84, and 0.92) 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 and a total of 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 [1] or [2]. The measurements are taken for 10 different Reynolds number varying between $5\times10^4<Re<1\times10^5$. The measurements are taken only in one half of the cross section about the symmetrical axis as shown in Fig. 4, which represent the flow characteristics of the entire duct. Measurements are made at the section x = 34.5D downstream from the leading edge of the heated section i.e. x = 94.56D from the unheated section. In this position both velocity and temperature fields can be considered to be fully developed [10]. The time mean velocity and temperatures of air are measured within the region of $-25<z<25$ (i.e. $-1<z/B<1$) and $0\leq y\leq 23$ (i.e.$0\leq y/B\leq$) at 7 different locations of y = 0, 3, 8, 13, 18, 21, and 23 (i.e., y/B = 0.0, 0.12, 0.32, 0.52, 0.72, 0.84, and 0.92) 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 and total 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 [2]. The measurements are taken for 10 Reynolds number varying between $5\times10^4<Re<1\times10^5$.

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. 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 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 [3,4,5]. The experimental results concerning Nusselt number, friction factor and Stanton number obtained for a turbulent flow through an asymmetrically heated smooth square duct with constant heat flux as the boundary condition are discussed briefly.

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 Nusselt number with location of positions of y/B is similar to that of local variation of mean air temperature, Fig. 7(a) [2]. The correlations obtained are expressed as follows:

$$\overline{Nu}_s = 175.35+47.82(y/B)^2 - 78.15(y/B)^3 \quad (8)$$

$$\overline{Nu}_s = 44.024 + 19\times10^{-4}\ Re \quad (9)$$

$$\overline{Nu}_s = 0.046\ Re^{0.753}\ Pr^{0.4} \quad (10)$$

With increase of Reynolds number both the Nesselts number curve shifts up in a similar manner as that of temperature of air [2]. At constant pressure drop local Nusselt number decrease linearly with increase of the local Reynolds number near the side walls in the region y/B ≥ ±0.52, but near the centre in the region y/B ≥ ±0.52 the local Nusselt number decreases sharply with Reynolds number remaining almost constant. This reflects the effect of secondary flow in the region 0.72≥y/B≥0.12. Also with increase of pressure drop the curve shifts up towards right. This fact is clearly displayed in the Fig. 5(b) where both the local and average of the local Nusselt number increases linearly with increase of Reynolds number and with change of location of positions y/B from centre towards the side walls the curve shifts up towards right. Fig. 5(b) displayed the distributions of local Nusselt number excluding the prandtl number across the test duct. In the circular pipe 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 Reynolds number as well as locations y/B across duct from centre towards the walls [2]. The local data in the fully developed region are averaged and correlated with Prandl number as variable parameter is shown in Fig. 6. The results compared well with the published data for Nusselt number and Stanton number except friction factor.

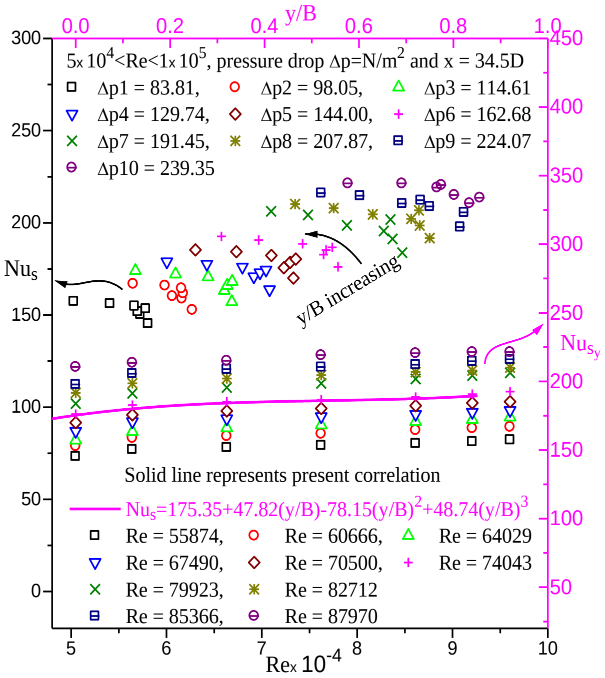


Fig. 5(a) Effects of Reynolds number on local Nusselt number and heat flux across duct.

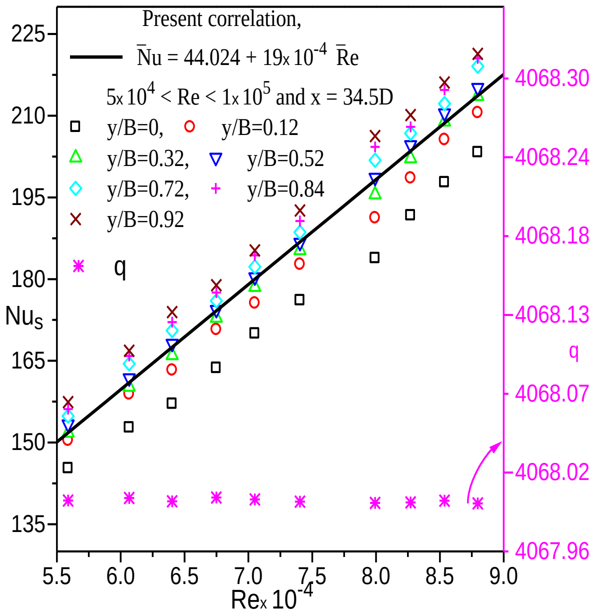


Fig. 5(b) Effects of Reynolds number on local Nusselt number and heat flux across duct.

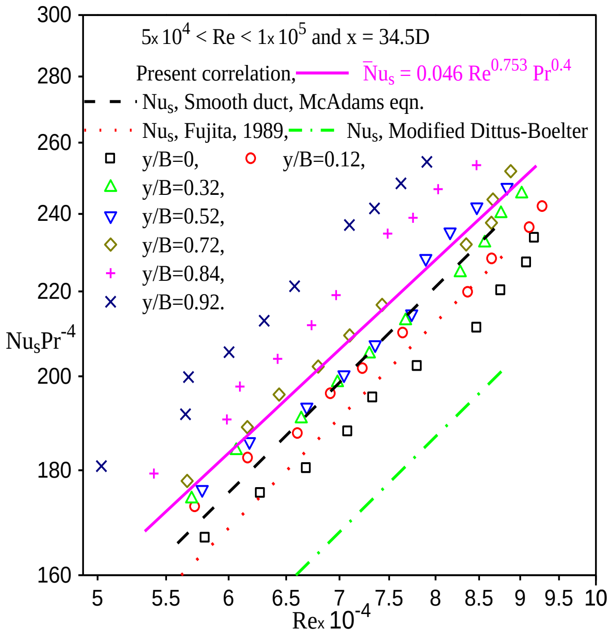


Fig: 6 Distributions of local and average Nusselt number across duct.

7.2 Friction Factor And Stanton Number

The Reynolds number and the location of positions of y/B across the duct dependence of both local friction factor and Stanton number for fully developed flows with constant heat flux are depicted in Figs. 7(a) and 7(b). Fig. 7(a) shows the distributions of local friction factor and Stanton number at constant pressure drop in a semi logarithmic plot. Both the friction factor and the Stanton number drop linearly with increase of locations y/B from centre towards the side walls because of the corresponding curves of air velocity decrease (Fig. 7(a) of [4]). The Fig. 7(a) also shows that with the increase of the pressure drop both the local friction factor and Stanton number curves shifts up towards the right but the corresponding values of friction factor and Stanton number gradually increasing and decreasing respectively with increase of pressure drop Δp. 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 Fig. 7(a) also demonstrates 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:

$$\bar{f}_s = 0.12\times10^{-4}\,Re^{0.037} \tag{11}$$

$$\overline{St}_s = 1.342\times10^{3}\,Re^{1.353} \tag{12}$$

Finally the effect of Reynolds number on average of local friction factor and Stanton number 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 friction factor increases with the increase of Reynolds number. Exactly for the same experimental set up and boundary condition 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 [2] the increased heat transfer in the asymmetrically heated duct is achieved at the expense of increased friction due to increase of both primary as

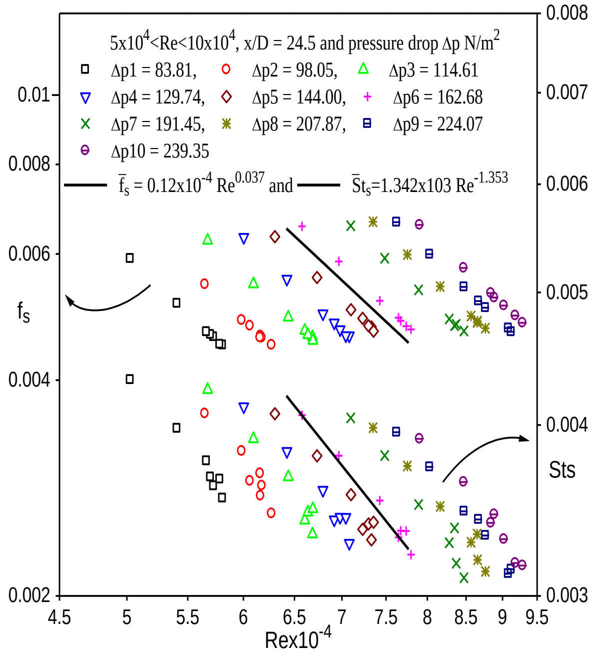


Fig. 7(a) Effects of local Reynolds number on friction factor and Stanton number across duct.

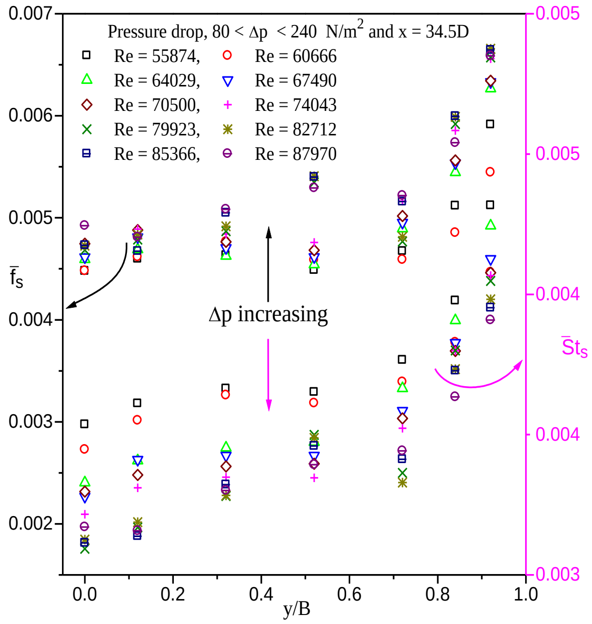


Fig. 7(b) Comparisons of local friction factor and Stanton number across duct.

well as secondary flow of air 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, $f = ((\Delta p/\Delta L)\,D)/(\tfrac{1}{2}\rho\upsilon^2)$, $f \propto \Delta p/u^2$, 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 true and hence not acceptable for the better the analysis and design of heat transfer equipment for engineering applications. The improved correlations obtained are as follows:

$$\bar{f}_s = 0.0032\ Re^{0.234} \tag{13}$$

$$\bar{St}_s = 0.0055\ Re^{-0.22} \tag{14}$$

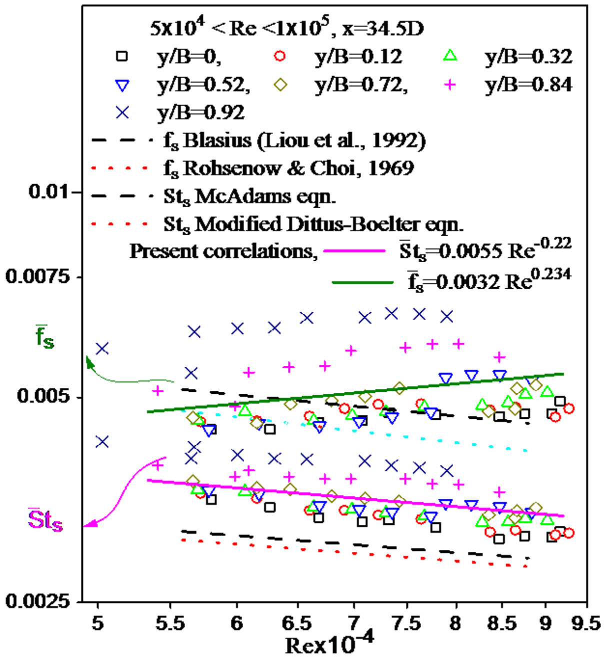


Fig. 8 Distributions local and average friction factor and Stanton number across duct.

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 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. 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 equipment for engineering applications.

8. ACKNOWLEDGMENTS

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 to BUET authorities and staffs, the panel of expert referees, especially the second author for their comments and suggestions, which led to substantial improvement of this work.

REFERENCES

[1] A .K. M. Abdul Hamid, “Experimental Study on Convective Heat Transfer with Turbulence Promoters”, Ph.D. thesis, Bangladesh University of Engineering and Technology, Dhaka, Bangladesh, 2004.

[2] A. K. M. Abdul Hamid, M. A. Taher Ali and M. A. R. Akhanda, "Characteristics of Air Properties And Variable Parameters For Turbulent Flow In An Asymmetrically Heated Smooth Square Duct", 7th International Conference of Mechanical Engineers, 2007.

[3] A. K. M. Abdul Hamid, M. A. Taher Ali and M. A. R Akhanda, “Characteristics of Time Mean Velocity and Temperature Fields for Turbulent Flows in an Asymmetrically Heated Smooth Square Duct", 7th International Conference of Mechanical Engineers, 2007.

[4] M. A. R. Akhanda, “Enhanced heat transfer in forced convective boiling,” Ph.D Thesis, University of Manchester, Institute of Science and Technology,1985.

[5] M. T. Ali, “Flow through square duct with rough ribs,” Ph.D. Thesis, Imperia College, University of London, U.K, 1978.

[6] H. Fujita, "Turbulent flows in square ducts consisting of smooth and rough planes”, Research Report of the Faculty of Engineering, pp. 3 11-25, Mie University, 1978.

[7] F. B. Gessner, "Turbulence and Mean-flow Characteristics of Fully Developed flow in Rectangular Channels," Ph.D. Thesis, Dept. Mech. Engg. Purdue University, 1964.

[8] J. C. Han, "Heat Transfer and Friction in Channels with Two Opposite Rib-Roughened Walls," ASME Journal of Heat Transfer, Vol. 106, No. 4, pp. 774-781, 1984.

[9] K. Hanjalic, and B. E. Launder, "Fully developed asymmetric flow in a plane channel", J. Fluid Mech.,Vol. 51, pp. 301-335, 1972,

[10] M. Hishida, Y. Nagano and A. Shiraki, “Structure of Turbulent Temperature and Velocity Fluctuations in the Thermal Entrance Region of a Pipe,” Trans. Japan Soc. Mech. Eng. (in Japanese), Vol. 44, No. 385, pp. 3145. 1978.

[11] W. M. Kays and M. E. Crawford, “Convective17 Heat and Mass Transfer”, McGraw-Hill, New York, 1980.

[12] K. Komori, A. Iguch and R. Iguni, "Characteristics of fully developed Turbulent flow and Mass Transfer in a Square Duct.”, Int. Chem, Eng., Vol. 20 (2), pp. 219-225, 1980.