Dept. of Mech. Eng.
CUET

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

Mechanical Engineering Research Journal

Vol. 10, pp. 78-89, 2016

ISSN: 1990-5491

CHARACTERISTICS OF TIME MEAN VELOCITY AND TEMPERATURE FIELDS FOR A FULLY DEVELOPED TURBULENT FLOW IN AN ASYMMETRICALLY HEATED SQUARE DUCT WITH RIBBED ROUGH BOTTOM WALL

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: This paper presents experimental results concerning time mean velocity and temperature fields obtained for a fully developed turbulent flow through an asymmetrically heated square duct with bottom wall ribbed rough at constant heat flux boundary condition for different Reynolds number. The air temperature decreases at a decreases rate from the heated wall towards the centre of the duct and beyond that the temperatures remain nearly constant i.e., approximately, equal to that of inlet temperature, indicating that the heat transfer becomes almost saturated and the top half of the duct behaves as a flat plate. The plots of Θ/Θ_c versus u/u_c fall on straight lines very close to each other. This demonstrates that Θ/Θ_c, correlates highly with u/u_c, suggesting the perfect similarity between the temperature and the velocity fields. The semi logarithmic plots of mean velocity and temperature show that they lay on a straight line indicating that the present experimental values obey the universal velocity and temperature distribution laws in the lower half of the duct cross-section near the heated bottom ribbed rough wall. The compact empirical correlations for universal velocity and temperature distributions for fully developed flow are obtained can be used for improved design of the heat transfer equipment for engineering applications.

Keywords: *Turbulent flow, Asymmetrically heated, Ribbed 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,
e = Ribbed height	Mm,
G = Mass flux	Kg/m^{2}s,
h = Heat transfer coefficient	W/m^2 ^{0}C,
k = Thermal conductivity	W/m ^{0}C,
L = Length	m,
P = Pressure	N/m^2,
p = Pitch	mm,
Pr = Prandtl number,	Dimensionless,
Q = Heat transfer,	W,
q = Heat flux,	W/m^2,
Re = Reynolds' number,	Dimensionless,
T = Mean temperature,	^{0}C,
T* = Friction temperature,	^{0}C,
T^+ = Universal temperature,	Dimensionless,
u = Time mean velocity,	m/s,
u* = Friction velocity	m/s,
u^+ = Universal velocity	Dimensionless,
Pr = Prandtl number	Dimensionless,
x, y, z = Coordinate system, Fig. 3 & 4.	

Greek Letters

Θ = Difference between wall temperature and air temperature	$(T_w - T)^0$C,
Θ_c = Difference between wall temperature and air temperature at centre of duct	$(T_w - T_C)^0$C,
δ = Aluminum wall thickness	mm,
Δ = Difference	Dimensionless,

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

Subscripts

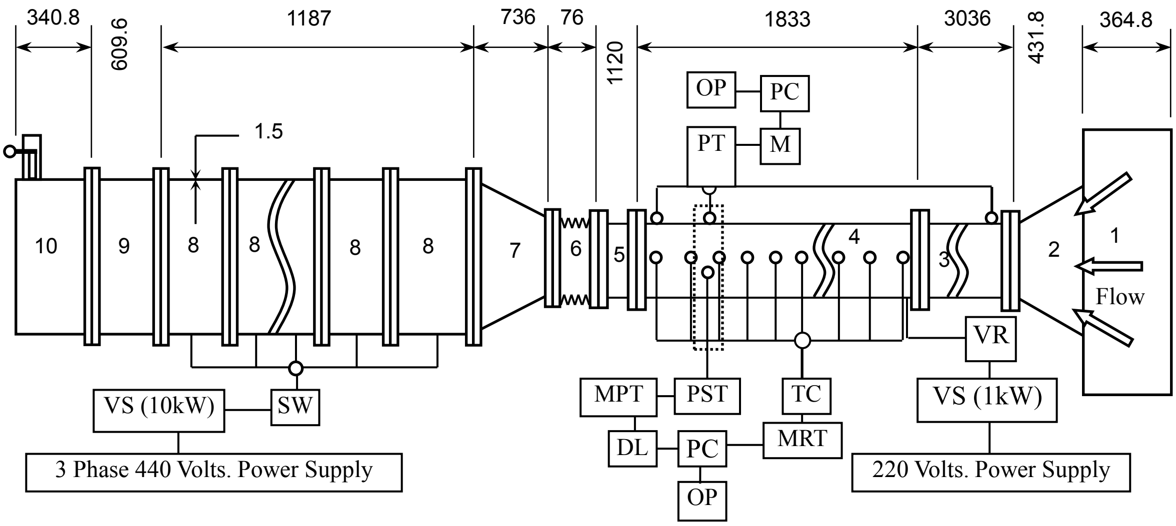
a = Ambient temperature,	f = Fluid,
b = Bulk mean,	i = Inlet,
C = Centre, correct ,	o = Outlet,
L = Loss,	r = Ribbed rough,
m = Mean,	s = Smooth duct,
LM = Log mean.	ln = log

1. INTRODUCTION

Forced convection from surfaces with large-scale roughness is encountered in many technological applications such as heat exchangers, advanced gas-cooled reactor fuel elements, and electric and electronic cooling devices. In the turbulent flow, the properties such as velocity, temperature and pressure are subjected to fluctuations both with location in the fluid and with time. A number of research investigators have published data on forced convective heat transfer and flow characteristic for different configuration of both duct and ribbed roughened elements as turbulent promoters. Amongst the wide variety of heat transfer mechanism, forced convective heat transfer is one which is used extensively in engineering fields. The study of forced convective heat transfer in a straight square duct with the bottom wall rib roughened as turbulent promoters gives the basic knowledge of fluid flow and temperature distribution characteristics, which will be helpful to identify and solve the various problems occur engineering in different flow systems and this knowledge will give better idea to design the different flow device and equipment required for heat transfer such as heat exchanger, gas turbine blade, chemical processing, nuclear reactors, electrical and electronic devices etc. Different types of surface roughness have been produced and their friction factors and heat transfer characteristics have been tested for possible use in heat transfer augmentation, such as: extended surfaces (i.e., Finns), enhanced devices (i.e., twisted taps), coiled tubes (i.e., curved pipes), roughened surfaces (i.e., random or periodic roughness element) etc. Surface roughened by periodic roughness elements is considered for the present experimental study.

The above discussion suggests the need to gain sufficient knowledge and clear understanding of heat transfer mechanism and the flow characteristics in the straight square duct with bottom wall periodically rib roughened as turbulent promoters and heated uniformly by electric current. Only the bottom wall of the duct with turbulent promoters is heated electrically. A similar experimental study has been reported by Hirota, et al., [8] but the heating of four sides of the duct was carried out by steam. The present research study presents both the characteristics of the velocity and temperature field in rough duct. The results of the mean velocity and temperature measurements in a straight square duct with a rib roughened bottom wall are compared with those obtained in a smooth square duct of similar configuration as well as with some well-known published data of similar nature. Air was working fluid; uniform wall temperature was the boundary condition.

The experimental results could be used to examine the results of numerical analysis and to check the validity of existing



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.

and future turbulent models. It is expected that this study will provide relevant information to solve forced convective fluid flow problems of non-circular ducts for the heat transfer designers.

The objectives of this present study are to investigate experimentally the flow characteristics of forced convective heat transfer in a straight square duct and to grasp the characteristics of both times mean velocity and temperature distribution of the entire duct cross section. The objectives are summarized below:

(i) The measurement of the mean primary velocity, mean static pressure, and mean temperature along the transverse direction at different downstream location will be taken with the help of available instrument and the raw data will be fed into the computer for data analysis.

(ii) With the analyzed data graphs will be plotted to study at the effects of various parameters. Correlations relating different parameters will be established.

(iii) The correlations obtained from the present research investigation will be compared with published data of some well-known scientists and research investigators.

2. LITERATURE SURVEY

Over decades the forced connective heat transfer in a noncircular duct with turbulent promoters has been the subject of many research investigators. One well-known method of enhancing heat transfer on a surface is to roughen the surface by the use of repeated ribs as turbulent promoters. These turbulent promoters break the laminar sub layer and / or buffer layer and create local wall turbulence due to flow separation and reattachment between the rib which greatly enhances the heat transfer and increases the pressure drop (drag). In the past decades, several researchers measured the primary flow velocity and got some useful results [3,6,12]) found that the length, required for full flow development in a pipe may exceed 140 diameters. Studies of similar detail are very few in more complicated geometries with dissimilar boundary conditions, curvature and heating systems of different processes. A brief literature survey indicates the lack of reliable experimental data on related variable parameters both velocity and temperature profiles in a square duct heated asymmetrically with constant heat flux boundary condition. A very few published data are available on straight square duct with bottom wall periodically rib roughened as turbulent promoters. In most of the research studies ribs were attached or glued onto the exposed surfaces of the walls, which causes thermal resistance. An experimental study on forced convective heat transfer from the square duct with bottom wall ribbed rough and heated asymmetrically. The present duct configuration is a simple straight square duct with one sided ribbed roughened by machining the aluminum plate (bottom wall) for understanding the flow characteristic and forced convective heat transfer mechanism. From this point of view, the flows in a duct of combination of rough and smooth walls have attracted interest [15]. Hence, the reason of conducting experiments for accurate measurements of turbulent flows in square and rectangular ducts with one rough wall [19].

Many investigators show that the turbulent flows as well as the temperature field in non-circular ducts are influenced by the existence of the secondary flow perpendicular to the stream wise flow direction [4,5]. 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 can not be ignored [6,11]. The secondary flow produces an increase in the wall shear stress towards the corners and significant influence on heat transfer at the walls [4]. The secondary flow not only induces a reduction in the volumetric flow rate, but it also causes the axial velocity field to be distorted with an outward shift of the contours of constant velocity. According to this viewpoint, only in a few experiments of fully developed flow has been achieved. Although there are a few studies of the latter aspect, there are a considerable number of experimental and analytical studies on the flows in straight square ducts which have a simple cross-section [14]. The effects of rib height and rib spacing on turbulent heat transfer and pressure drop in rectangular channel were reported by [16] and [18]. 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. Until recently, experimental investigation of these cases was confined to comparatively simple cases with symmetrical boundary conditions. For longitudinally uniform temperature boundary condition of the present investigation, the thermally fully developed region is characterized by air temperature that increases linearly as a function of longitudinal position.

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 hydrodynamically 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 seconds

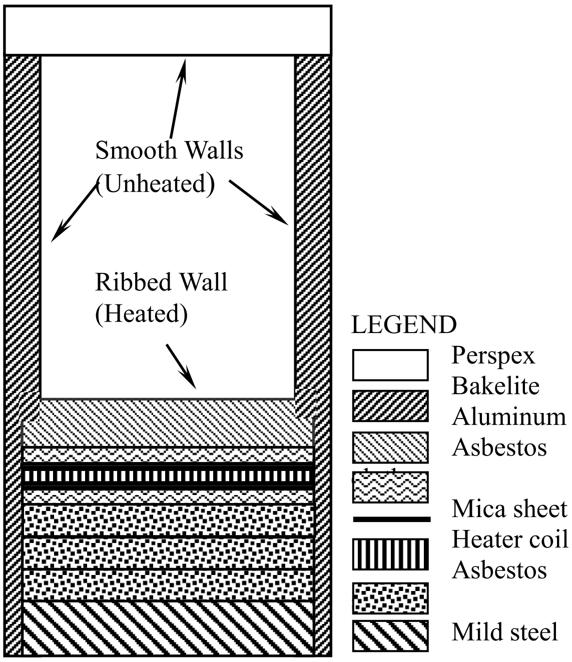


Fig. 2 Illustrating the Cross sectional View of the Duct.

are used. Thermocouples are used to measure wall and air temperature. 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 chrome wire of size 28 SWG having the resistance of 9.8097 Ω/m is used to achieve uniform wall heat flux conditions. As the thermocouple is attached intrinsically with the pilot static tube measurements of both the velocity and temperature of the air flowing through the duct are 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. Measurements of temperature and velocity were recorded at the location of x = 26.5D downstream from the leading edge of the heated section where both the temperature and velocity are said to be fully developed.

4. MEASUREMENT SYSTEMS

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≤ Z/B ≤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≤ Z/B ≤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 united Senser (USA) pilot 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 pilot static tube are transmitted to pressure transducer through 1.4 mm bore flexible tygon tubing.

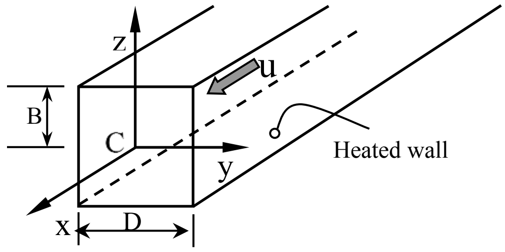


Fig: 3 Geometric parameters and wall coordinate system.

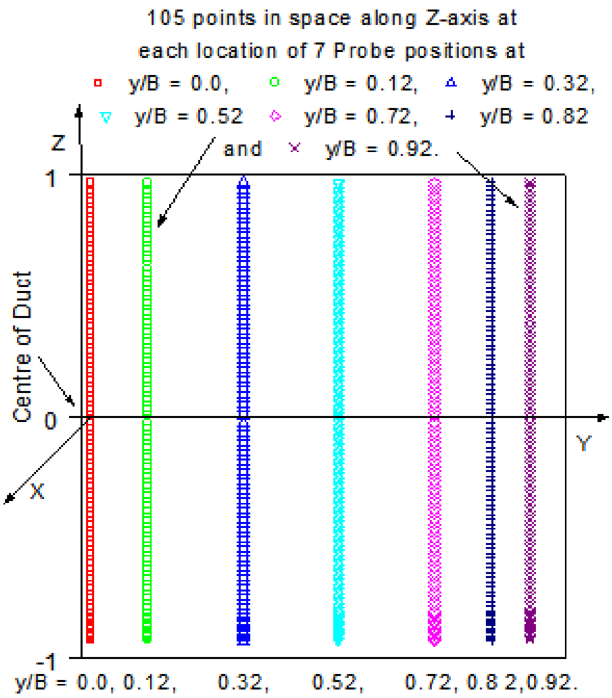


Fig. 4 Illustrating 7 Probe position along Y-axis for measuring Velocity and Temperature.

The signals of the digital micro-voltmeter 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 wall and air temperature, for detail explanation [1].

5. DATA REDUCTION

The net heat transfer rate can be calculated from

$$Q = \rho_f u_m A_c C_p (T_o - T_i) = G A_c (T_o - T_i) \quad (1)$$

$$q = Q/A_s \quad (2)$$

The local outer wall temperature T_w is read from the thermocouple output. The corrected local inner wall temperature, T_{wc} is calculated by one dimensional heat conduction Eq. (3) as:

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

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

$$h = q / \Delta T_{lm} \quad (4)$$

Where, $\Delta T_{LM} = (T_o - T_i) / \ln [(T_{wc} - T_i) / (T_{wc} - T_o)]$ (5)

Since the air velocity and temperature varies along the duct [1] and [8], all the physical properties of air and the related parameters are calculated at the local bulk mean air temperature, $T_b = ½(T_o + T_i)$ and local bulk mean air velocity, $u_b = ½(u_o + u_i)$ for each data recorded in space. The time mean velocity u is normalized by the mean velocity at the centre u_c as u/u_c and the temperature T is normalized as θ/θ_c.

where, $\theta = (T_{wc} - T)$ and $\theta_c = T_{wc} - T_c$. (6)

Also the time mean velocity and the mean temperature are nondimensionalized by the friction velocity, u^* and by the friction temperature, T^* as:

$$u^+ = u/u^* \text{ and } T^+ = (T_{wc} - T)/T^* \quad (7)$$

where, $u^* = \sqrt{(\tau_w/\rho)}$ and $T^* = q/\rho C_p u^*$ (8)

For any flow field, these nondimensional parameters are important criteria to indicate its characteristics. Similarly the distance measured from the bottom wall surface, Z is also nondimensionalized as:

$$Z^+ = u^* Z/\nu \quad (9)$$

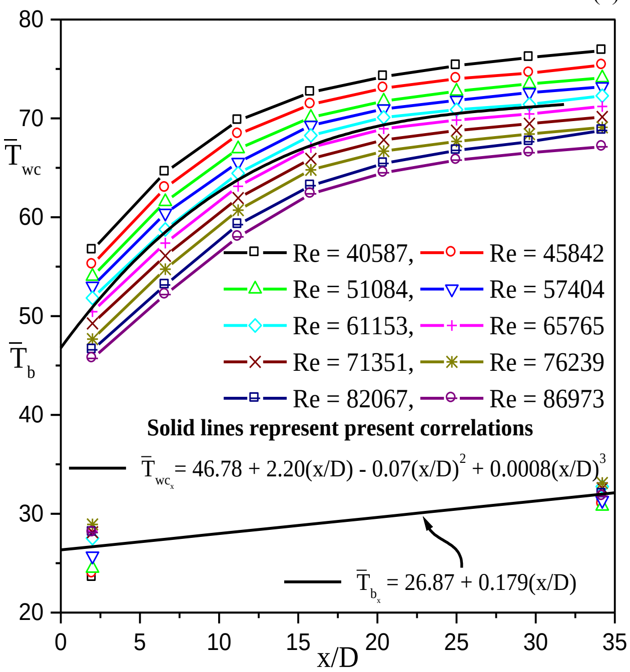


Fig. 5 Wall and Air temperature distributions.

6. DATA ANALYSIS

The top and two sides having walls and the bottom ribbed rough wall make the duct asymmetric in nature. Also the heated bottom ribbed rough wall creates flow field asymmetric. Thus duct is symmetric about z-axis but asymmetric about the y-axis. Hence the measurements are taken only in one half of the cross section about the symmetrical z-axis as shown in Fig. 4. Measurements are made at the sections x = 2D and x = 34.5D downstream from the leading edge of the heated section i.e. at x = 60D and x = 94.56D respectively from the unheated section. At position x = 94.5D both velocity and temperature fields can be considered to be fully developed, [8] and [9]. The time mean velocity and temperature of air are measured within the region of $0 \le y/B \le 1$ at 7 different locations of y/B = 0.0, 0.12, 0.32, 0.52, 0.72, 0.84, and 0.92 in the cross section, Fig. 4. There are typically 105 measurement points in space at each measuring location and there is a total of 105×7 = 735 points in space for the one half of the cross section of the duct which represents the data for the entire duct cross section. The time mean velocity and the temperature are analyzed from the probability distribution function of the measurements recorded by the data logger. The measurements are taken for 10 different Reynolds number varying between $4\times10^4<Re<9\times10^4$. In the analysis of data the heat transfer surface area increased by 30.37 percent is included unlike the previously published data where they ignored the surface area of ribs. In ignored the surface area of ribs. In the present investigation ribs are cut down from the surface of the wall instead of attaching ribs by glow which causes thermal resistance [8]. The present analyzes are carried out with the local values physical properties and all related variable parameters for each points in space [2]. Hence it can be said that the present experimental results obtained are more accurate and reliable. The corresponding statistical error is between 0.51 to 2.05 percent in the time mean velocity and between 1.23 to 2.15 percent in the temperature. The scattering of the wall temperature measurement is found to be between 2.11 to 3.24 percent and the uniformity of the wall temperature distribution is considered to be satisfactory [10]. The time velocity measurements are repeated whenever error or doubtful situations occurred to ensure that the measured results are repeatable.

7. RESULTS AND DISCUSSION

The experimental results concerning a time mean velocity and temperature fields obtained for a turbulent flow through an asymmetrically heated ribbed rough square duct with constant heat flux as the boundary condition are investigated. To get the over view regarding the magnitude and the distribution of these two important parameters of the flow field the profiles are drawn and discussed briefly for the lowest and the highest Reynolds numbers only.

7.1 Wall and Air Temperature Distributions along Length of Duct

Fig. 5 illustrates the distributions of local bottom wall temperature, T_w supplied along the length of the duct and the bulk mean air temperature, T_b in the duct as a function of longitudinal positions. The longitudinal distribution of the bulk mean air temperature is assumed to increase linearly as stated by many investigators [3,7,8]. 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. The figure shows the linear increase of bulk mean air temperature measured from the inlet to the outlet. At constant Reynolds number, the wall temperature increases at a decreasing rate approaching the asymptotic condition near x = 15D and after this the longitudinal distribution of wall temperature increases linearly up to the last station of measurement at x = 34.5D. The Figure 5 also shows continuous shifting of the temperature profile curve downwards with the increase of Reynolds number confirming increased heat extraction from the heated wall with the increase of flow velocity. Typically, at downstream distances from x = 15D from the start of leading edge of the heating section Fig. 5, the wall temperature become parallel to aforementioned bulk mean air temperature straight line. The average of the bulk mean temperature distribution and the corrected wall temperature can be expressed by the equations obtained as given bellow respectively:

$$\overline{T}_b = 26.87 + 0.179(x/D) \quad \text{Fig. 5} \tag{10}$$

$$\overline{T}_{wc} = 46.78 + 2.20(x/D) - 0.07(x/D)^2 + 0.0008(x/D)^3 \quad \text{Fig. 5} \tag{11}$$

Fig. 6 illustrates the effect of Reynolds number on local corrected mean wall temperature, $\overline{T}_{wc}$ and on heat flux, q. With the increase of Reynolds number the wall temperature drops linearly at each location, y according to correlation obtained as follows:

$$\overline{T}_{wc} = 79.48 - 0.00023Re \quad \text{Fig. 6} \tag{12}$$

Also the curve shifts downwards with increase of y/B.

This phenomenon is due to the combined effects of the secondary flow as well as the turbulence intensity increases with the increase of Reynolds number which in turn increases wall friction. Hence the increase of heat extraction by flowing air causes the wall temperature to drop.

7.2 Mean Velocity Profiles

The velocity profiles in the region 0≤y<8, Fig. 7(a) show the usual parabolic form having the maximum near the centre towards the unheated wall instead of occurring at the axis of symmetry z = 0. And this peak simultaneously becomes less prominent and vanishes as the profile moves towards the sidewalls for y>8 and the profiles near the side walls show their saddle shape form. The secondary flow carries the high velocity flow from the center towards the corner increasing the velocity there, thereby creating this shape of the profiles. The velocity profiles locations y<8 are different from those near the sidewall

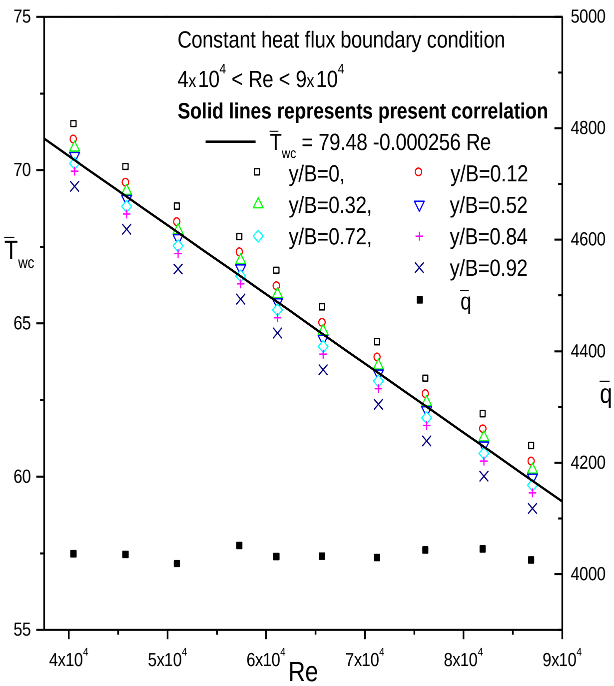


Fig. 6 Effects on local wall temperature and heat flux.

because of the fact that there is no bulging in these profiles indicating little influence of secondary velocity on mean axial velocity in the central region of the duct. Thus the stream wise flow velocity profiles, bulges towards the corner along the corner bisectors and depression from duct centre towards the side walls along wall bisectors producing saddle like shape indicating the effects of secondary flow. Over decades the forced connective heat transfer in a noncircular duct with turbulent promoters has been the subject of many research investigators. One well-known method of enhancing heat transfer on a surface is to roughen the surface by the use of repeated ribs as turbulent promoters. These turbulent promoters break the laminar sub layer and or buffer layer and create local wall turbulence due to flow separation and reattachment between the ribs which greatly enhance the heat transfer and increase the pressure drop (drag). In the past decades, several researchers measured the primary flow velocity and got some useful results i.e., [3], [6] and [12] found that the length, required for full flow development in a pipe may exceed 140 diameters. Studies of similar detail are very few in more complicated geometries with dissimilar boundary conditions, curvature and heating systems of different processes. A brief literature survey indicates the lack of reliable experimental data on related variable parameters both velocity and temperature profiles in a square duct heated asymmetrically with constant heat flux boundary condition. An experimental study on forced convective heat transfer from the square duct with bottom wall ribbed rough and heated asymmetrically. A very few published data are available on straight square duct with bottom wall periodically rib roughened as turbulent promoters. In most of the research studies ribs were attached or glued onto the exposed surfaces of the walls, which causes thermal resistance. The present duct configuration is a simple straight square duct with one sided ribbed roughened by

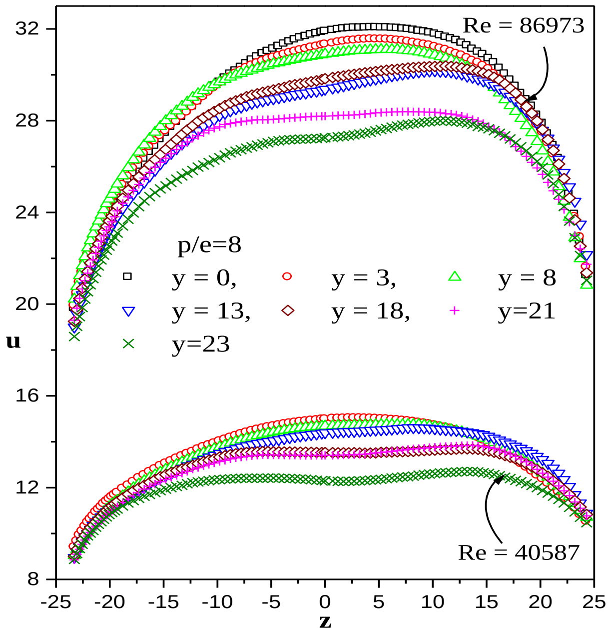


Fig. 7(a) Distribution of local velocity profiles.

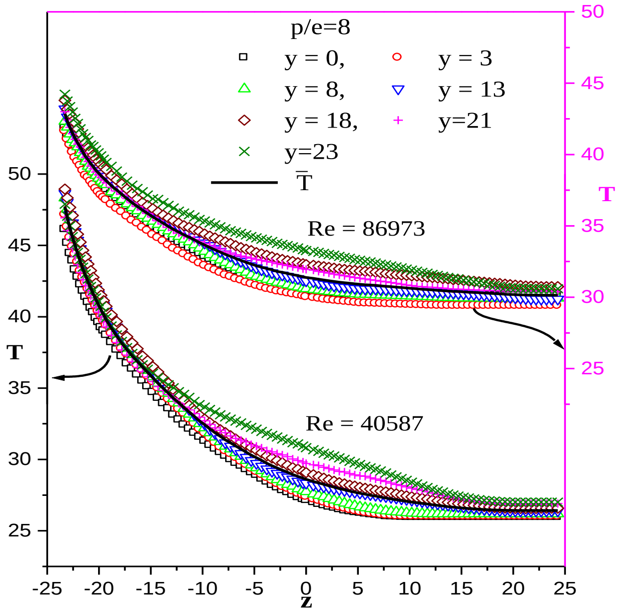


Fig. 8(a) Distribution of local temperature profiles.

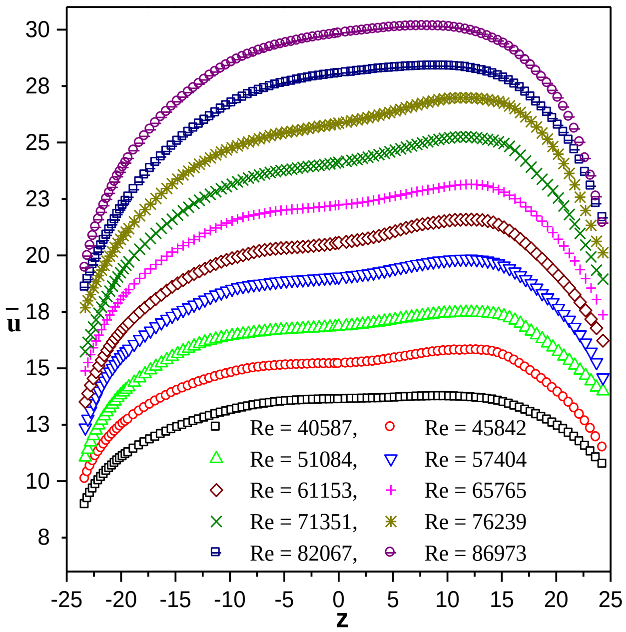


Fig. 7(b) Comparison of mean velocity profiles.

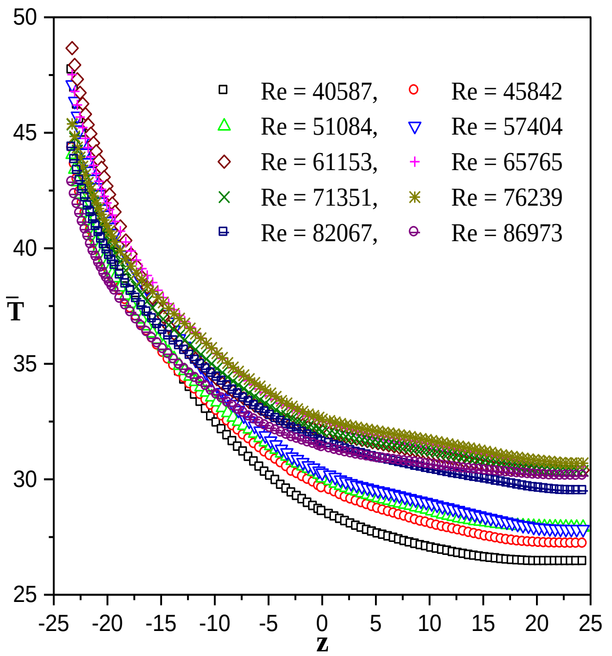


Fig. 8(b) Comparison of mean temperature profiles.

machining the aluminum plate (bottom wall) for understanding the flow characteristic and forced convective heat transfer mechanism. From this point of view, the flows in a duct of combination of rough and smooth walls have attracted interest, [15]. Hence, the reason of conducting experiments for accurate measurements of turbulent flows in square and rectangular ducts with one rough wall [19].

The bulging of velocity profile at each location for y>8 is greater near unheated top smooth wall than that of the heated bottom ribbed rough wall. This difference in peak values on velocity profile is because of increase in turbulence intensity due to existence ribbed roughen wall surface as well as the effects of heat transfer from the bottom heated wall increasing the wall friction considerably.

This phenomenon combined with secondary flows on the primary velocity profiles becomes more and more prominent and the saddle shape form of profiles from its usual parabolic form characterizes their influence. Also the velocity gradients near the heated bottom ribbed rough wall are higher than those near the unheated top smooth wall. The temperature difference changes the viscosity of air thereby reshaping the profile and at the same time the presence secondary flow however small it is in that region, greatly influence the mean velocity profile to its final shape as found in the experiment. Fig. 7(b) shows the Comparison of average of time mean velocity profiles at constant Reynolds number. The profiles shift upward with increase of Reynolds of number. It is clearly seen that the velocity gradient is steeper near the heated bottom ribbed rough wall than that near the unheated top smooth wall as well as near the bottom wall the velocity gradients are more steeper at higher Reynolds number than that at the top wall.

7.3 Mean Temperature Profiles

As it can be seen from Fig. 8(a) that the air entering the heated test section the temperature remains almost constant in the top half region of the duct i.e., z>0, which means that in this region the duct behaves like a flat plate. In the lower half of the duct i.e., –z<0 the temperature of air starts increasing at the increasing rate until the heated bottom ribbed rough wall where the maximum temperature is reached. The Fig. 8(a) also shows that temperature profiles in the region -20<y<0 are distorted because of the combined effects on flow fields at each location of y due to strong influence of the secondary flows, enhancing turbulence intensity because of existence of ribbed roughen wall surface and the effects of heat transfer. This phenomenon increases with the increase of y>0 towards the side walls and hence more heat extraction taking place in this region. Also this effect is greater at higher Reynolds number. Fig. 8(b) shows the comparison of average of mean temperature profiles for 10 different Reynolds number. It can be seen from these plots that although the variation of uncontrolled (room temperature i.e., 24.11 °C to 30.34 °C) entry temperatures is wide, the temperature profiles near the heated bottom wall merge together, which confirms the heat extraction by air from wall is saturated. At higher Reynolds number both the time mean velocity and the temperature profiles shift upwards indicating enhance transfer of momentum and energy, and hence increase of convective heat transfer coefficients with the penalty of increase in surface friction. This is because the viscosity increases with the increase of the Reynolds number and location of positions, y/B from the centre of the duct towards side walls.

7.4 Normalized Mean Velocity Profiles

The mean temperature profiles are also normalized for the reasons described above for normalized velocity profiles. The normalized mean temperature profiles at different y/B locations are shown in Fig. 10(a) for the lowest and the highest Reynolds number. Normalized mean temperature profiles of smooth duct show that Θ/Θ_c increase with decreasing rate up to the center of the duct and beyond that the Θ/Θ_c of air approach to constant values to that of inlet air temperature showing negligible effect of heating. Thus it can be said that the duct behaves like a flat plate showing the little or no effect of unheated side and top smooth walls on their shapes. Fig. 10(b) shows the comparison of average of the normalized mean temperature profiles in a water fall view.

7.5 Normalized Mean Temperature Profiles

The mean temperature profiles are also normalized for the reasons described above for normalized velocity profiles. The normalized mean temperature profiles at different y/B locations are shown in Fig. 10(a) for the lowest and the highest Reynolds number. Normalized mean temperature profiles of smooth duct show that Θ/Θ_c increase with decreasing rate up to the center of the duct and beyond that the Θ/Θ_c of air approach to constant

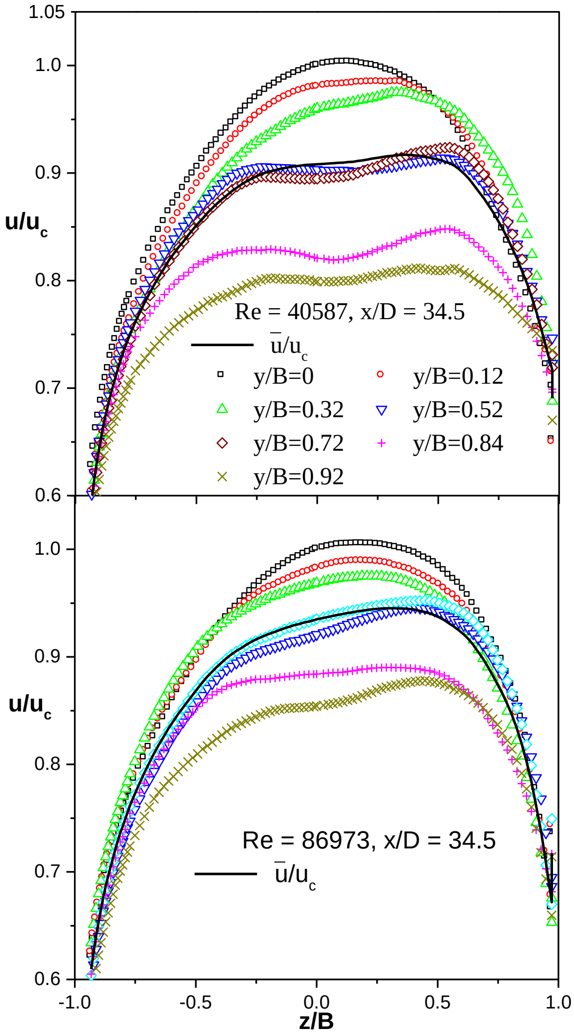


Fig. 9(a) Normalized local time mean velocity profiles.

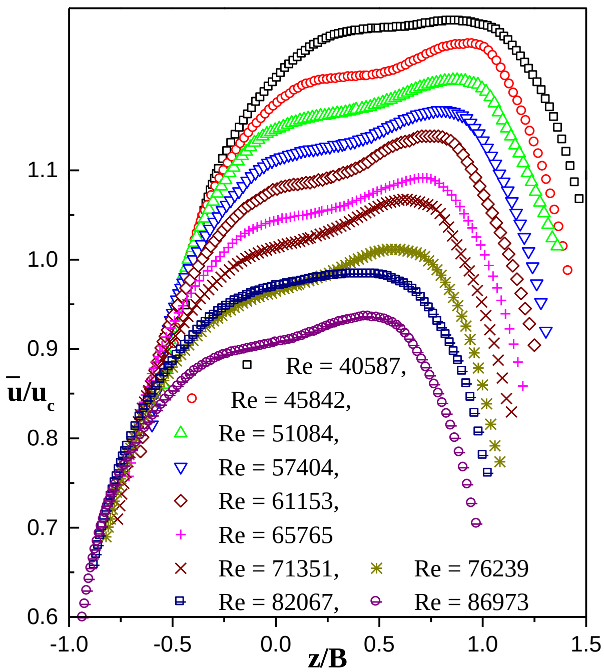


Fig. 9(b) Comparison of normalized velocity profiles.

show that Θ/Θ_c increase with decreasing rate up to the center of the duct and beyond that the Θ/Θ_c of air approach to constant

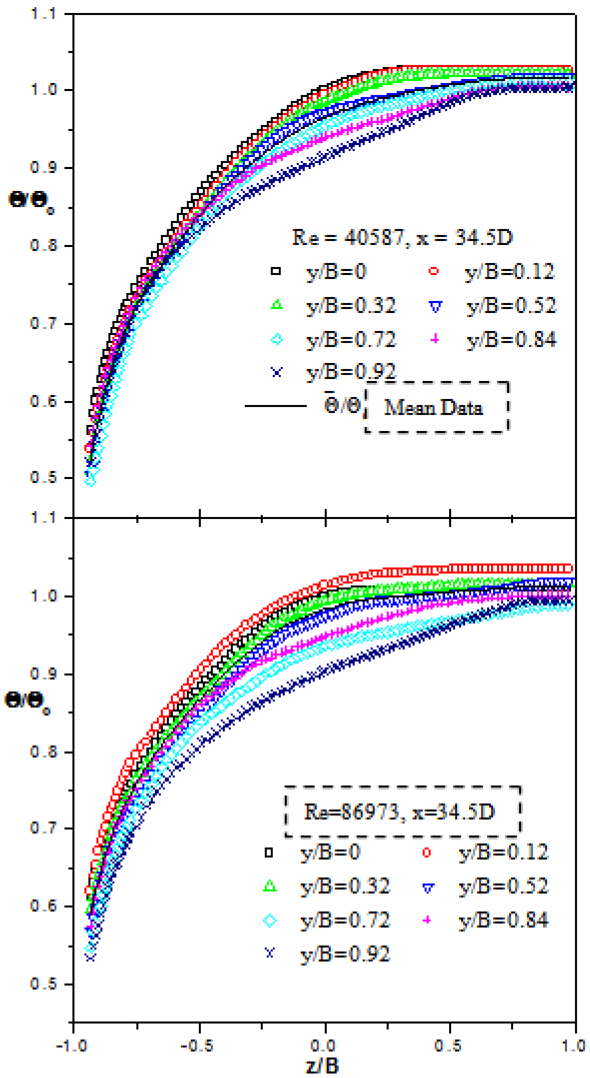


Fig. 10(a) Normalized local temperature profiles.

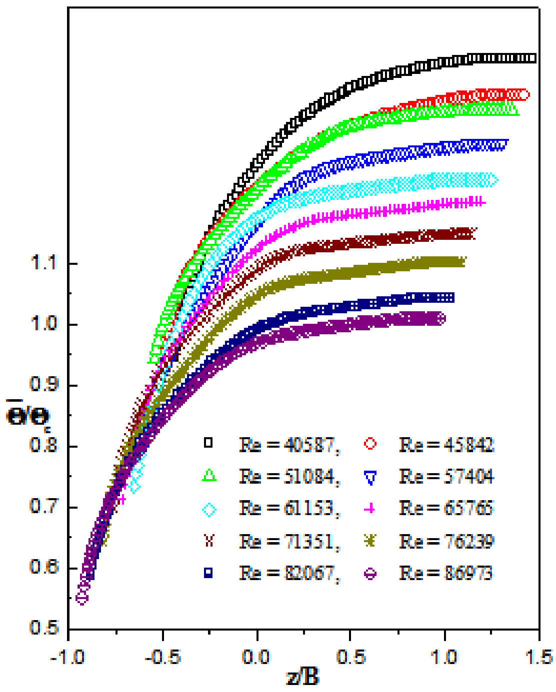


Fig. 10(b) Comparison of normalized.

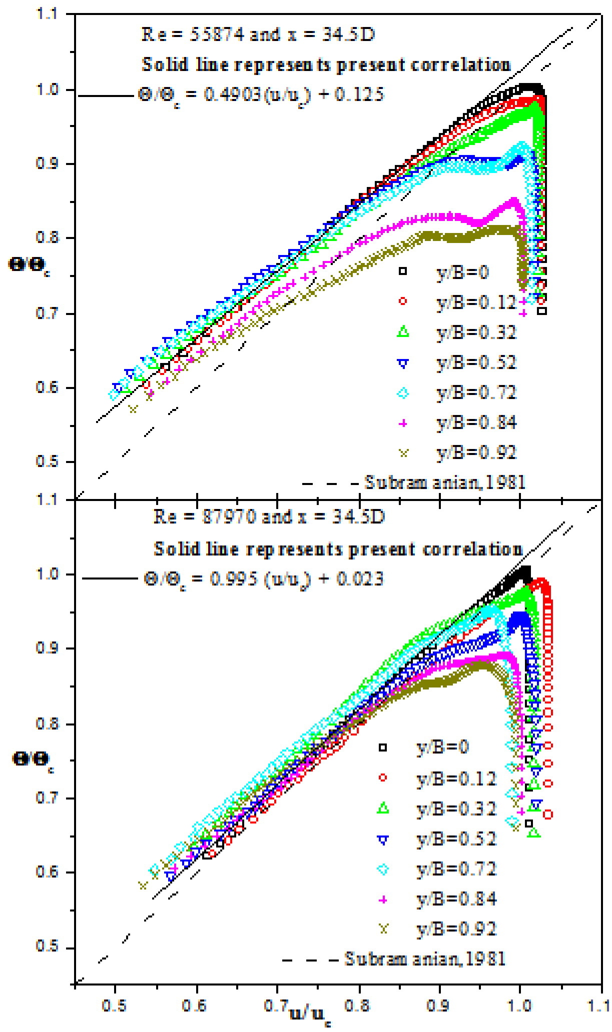


Fig. 11(a) Correlations between non-dimensional Local velocity and mean temperature.

values to that of inlet air temperature showing negligible effect of heating. Thus it can be said that the duct behaves like a flat plate showing the little or no effect of unheated side and top smooth walls on their shapes. Fig. 10(b) shows the comparison of average of the normalized mean temperature profiles in a water fall view.

7.6 Correlations between Mean Temperature and Velocity Profiles

The similarity between the non-dimensional temperature and the velocity fields for the lowest and the highest Reynolds number are shown in Fig. 11(a). If the temperature profile is in perfect agreement with that of velocity then the correlation follows the straight line, [13]. Accordingly, it is plausible that the more the temperature Θ/Θ_c, correlates with the velocity u/u_c, the more complete is the similarly between temperature and velocity fields, [8]. Fig. 11(a) shows that the correlations curves fall on straight lines very close to each other and demonstrating high correlation between temperature and velocity fields for the range of Reynolds number studied.

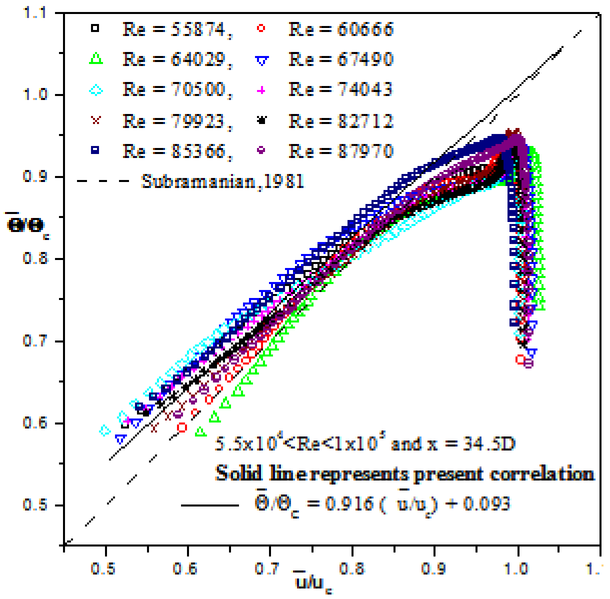


Fig. 11(b) Comparison of correlations between Normalized mean velocity.

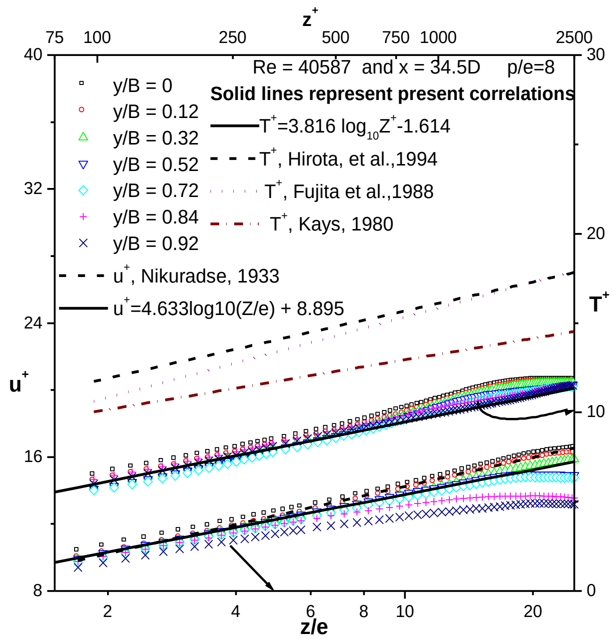


Fig. 12(a) Universal local mean velocity and temperature.

This demonstrates that Θ/Θ_c, correlates highly with u/u_c, suggesting the similarity between the temperature and the velocity fields. The corresponding ratios of Θ/Θ_c are greater than those of u/u_c for $u/u_c \leq 0.85$. Fig. 11(a) also shows that the degree of similarity decreases with increase y/B from the centre towards side walls. Fig. 11(b) shows the comparison of correlations between the temperature and velocity fields. In the present investigation the slopes of the curves ranges lie between 0.49 and 0.99. These slopes indicate the degree of similarity between the temperature and velocity fields. In a similar experiment [8] found the slope of 0.47 of similar plot for a single Reynolds number $6.5x10^4$. The present values are very close to that found by [7], [8] and [13].

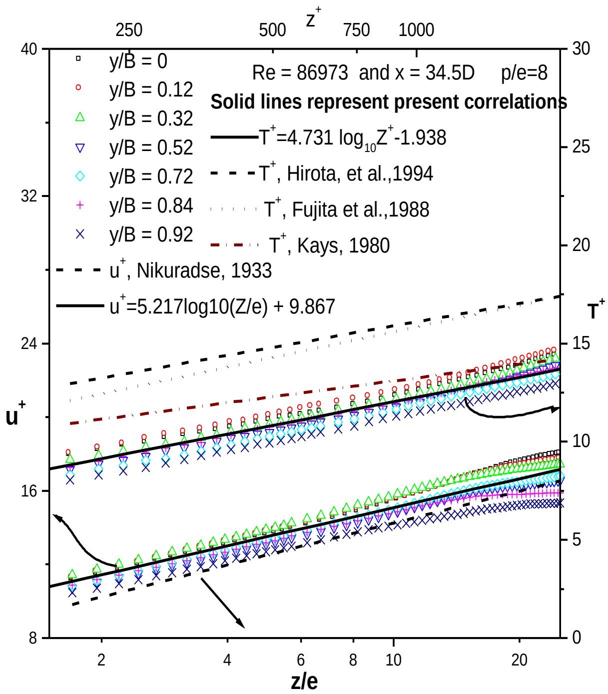


Fig. 12(b) Universal local mean velocity and temperature.

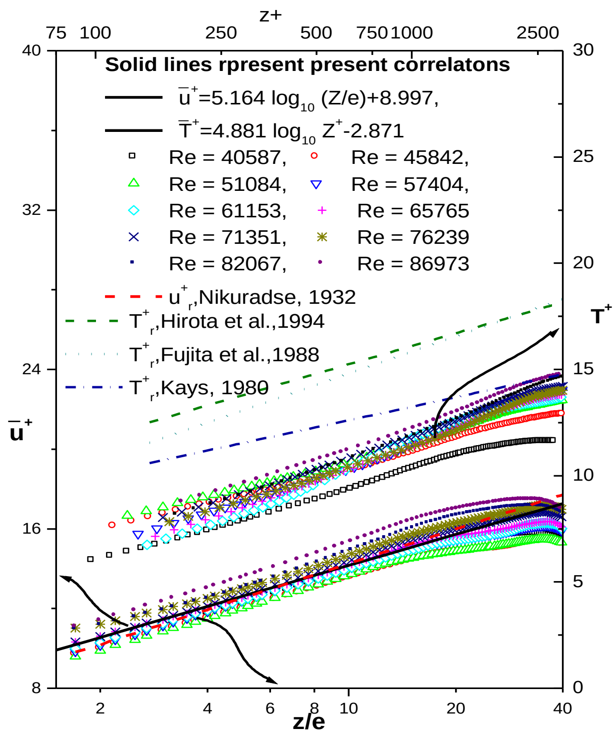


Fig. 12(c) Comparison of menu universal velocity and menu temperature.

Though in the experiment there is no systematic increase or decrease of the slope with Reynolds number but it can be concluded from the trend of variation of slopes the temperature and velocity fields approach to perfect similarity condition. In the present investigation the correlations obtained for the lowest and the highest Reynolds number are given in the respective plots and only the average of the mean correlations for the range of Reynolds number studied is quoted as follows:

$$\bar{\Theta}/\Theta_c = 0.916(\bar{u}/u_c) - 0.093 \quad \text{Fig. 11(b)} \quad (13)$$
for 40587<Re<86973

7.7 Universal Velocity and Temperature Distributions

As both the time mean velocity and temperature of a heated flow field are of some polynomial function of the wall distance and wall temperature, their plots in simple linear scale shows in general, the characteristic of the flow field. The shape of the velocity profile in any ribbed rough wall flow field is greatly influenced by the wall shear stress, τ_w and the air density, ρ. This is because with the increase of air temperature the viscosity of air increases which intern increases the shear stress and the density of air decrease. Thus for the same pressure difference the velocity of the air continues to accelerate with increase of air temperature. The universal velocity and temperature distributions are shown in Fig. 12(a) and Fig. 12(b) for the lowest and highest Reynolds number respectively of the present investigation. In these figures experimental values for different y/B locations are, plotted and their mean values are shown by the solid lines while the broken lines represent the published data by some well-known investigators as mentioned in the figures. As shown in these figures the turbulent part of the wall region $30<Z^+<250$, the mean velocity profiles are in good agreement with the logarithmic law of velocity profiles similar to published data by investigators. The generalized correlation obtained in the present investigation can be expressed by the following equation as follows:

$$u_r^+ = 5.164 \log_{10}(Z/e) + 8.997 \quad \text{Fig. 12(c)} \quad (14)$$

Fig. 12(c) shows the comparison of the universal distributions of mean velocity and temperature for the range of Reynolds numbers studied. The semi logarithmic plots of mean velocity and temperature show that they lay on a straight line indicating that the present experimental values obey the universal velocity and temperature distribution laws. The figure shows the little effect of their distribution due to the variation of Reynolds number. The following generalized logarithmic law is obtained and is represented by the solid line in Fig. 12(b) as follows:

$$\bar{T}_r^+ = 4.881 \log_{10} Z^+ - 2871 \quad \text{Fig. 12(c)} \quad (15)$$

8. CONCLUSIONS

The conclusions of the present experimental research studies on forced convective heat transfer in a rib roughened specimen (p/e = 8) are as follows:

I. The bulk mean temperatures of fluid increase with increase of Reynolds numbers i.e., the fluid temperature increase by only 16% as against 61% increase in Reynolds numbers for smooth duct whereas for ribbed duct (p/e = 8) fluid temperatures increase by only 10% as against 85% increase in Reynolds numbers. This is because roughened wall surfaces increase the turbulent intensity and hence the increase in heat transfers from the heated walls surfaces. Thus the heat transfer capability is higher for the square duct roughened with ribs as turbulent promoters than that of the smooth square duct.

II. The ratio of $[u/u_c]_{max}$ lie in the region $0.074 \le z/B \le 0.494$ for rib roughened duct, p/e=8. The $[u/u_c]_{max}$ values are 1.006 at 8.71×10^4, and 1.078 at 7.16×10^4, for the smooth duct and the rib roughened ducts of p/e=8 respectively.

III. For the smooth duct the values of u/u_c are higher than the corresponding values of θ/θ_c for the ratios of $\bar{\theta}/\theta_c<0.9$.

IV. For the rib roughened duct with p/e = 8, the values of θ/θ_c are higher than the corresponding values of u/u_c over the range of Reynolds numbers, $4.36\text{x}10^4 < Re < 8.07\text{x}10^4$. The results fall on a straight line which demonstrates that θ/θ_c correlates highly with u/u_c and suggests fairly complete similarly between temperature and velocity fields.

V. The universal velocity distribution fall on a line in the turbulent part of the wall region 1.85 <(Z /B)< 6 where the inner law is generally valid. The universal velocity distributions agree well with published data in the region 0 <(y / B) < 0.52 at higher Reynolds number i.e., Re > 6×10^4.

The relations obtained are compared with that of the smooth duct are as follows:

Rib roughened square duct, p/e = 8	Smooth square duct [1]
$u^+_{1r,8} = 5.164 \log_{10}(Z/e) + 8.997$	$u^+_s = 3.941 \log_{10}(Z/e) + 6.77$
$T^+_{1r,8} = 4.881 \log_{10} z^+ - 2.871$	$T^+_s = 5.352 \log_{10} z^+ + 0.071$

9. ACKNOWLEDGMENTS

This is a part of the experimental research works carried out by the first author leading to PhD degree under the guidance of the second and third authors in the Department of Mechanical Engineering at the Bangladesh University of Engineering and Technology (BUET), Dhaka, Bangladesh. The first author is grateful to BUET authorities and staffs for all the financial supports, the panel of expert referees, especially the second and third authors for their comments and suggestions, which led to substantial improvement of this work. This paper was presented in 4th BSME – ASME International Conference on Thermal Engineering, Paper:151: pp683 – 694; 2008, Dhaka, Bangladesh.

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