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CHARECTERISTICS OF TIME MEAN VELOCITY AND TEMPERATURE FIELDS FOR TURBULENT FLOWS IN AN ASYMMETICALLY HEATED SMOOTH SQUARE DUCT

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ABSTRACT

This paper presents experimental results concerning a time mean velocity and temperature fields obtained for a fully developed turbulent flow through an asymmetrically heated smooth square duct with constant heat flux boundary condition for ten Reynolds numbers varying within the range of $5\times10^4<Re<1\times10^5$. The time mean air velocity profiles near the centre show parabolic form exhibiting little or no influence of secondary flow. 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, saddle like shape indicating the effects of secondary flow. These effects gradually increase with the increase of locations from centre towards the side walls. Near the side walls the secondary velocity on velocity profiles becomes prominent and the saddle shape form of profiles from its usual parabolic form characterizes its influence. 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 indicating that the heat transfer becomes almost complete and the duct behaves as a flat plate. The plots of Θ/Θ_c versus u/u_c fall on a straight line for each Reynolds number. This indicates that Θ/Θ_c correlates with u/u_c and suggests the similarity between temperature and velocity profiles. In the lower half of the duct cross-section near the bottom heated wall the mean universal velocity distribution and the mean universal temperature distribution confirm the validity of the inner law. The compact semi-empirical correlations for universal velocity and temperature distributions for fully developed flow are obtained. Previously all calculations were done assuming that the Pr is constant but in this experimental investigation it has been found that the it decreases with increase of both Re as well as locations from centre towards the side walls. Hence compact correlations obtained, include Pr as a variable parameter, can be used for improved design of the heat transfer equipments for engineering applications.

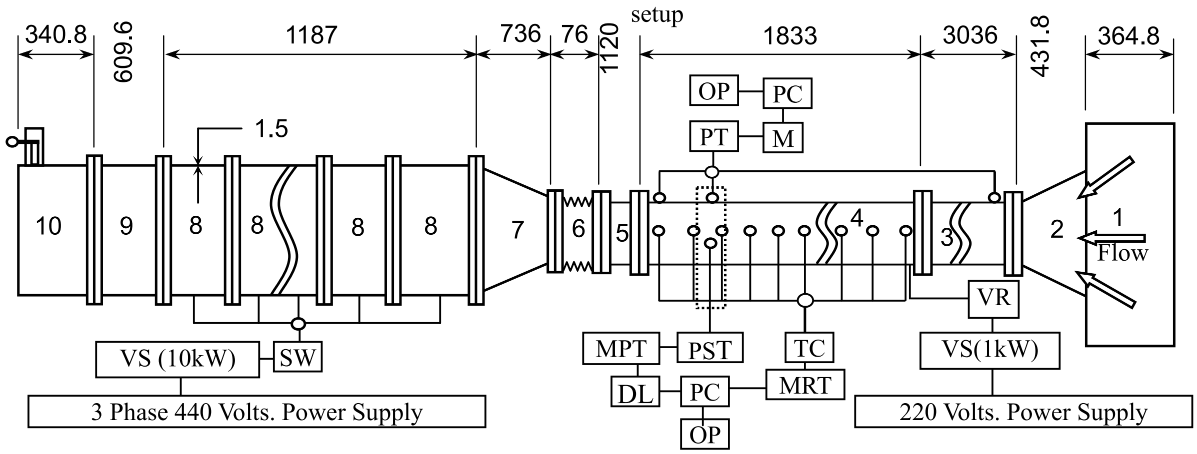
Keywords: Time Mean Velocity, Mean Temperature, Smooth Square Duct.

1. INTRODUCTION

Amongst the wide variety of heat transfer mechanism, forced convective heat transfer is one which is used extensively in engineering fields. Many investigators show that the flow in noncircular ducts is accompanied by secondary flow perpendicular to the stream wise flow direction. This secondary flow not only reduces volumetric flow rate but also distorts the axial velocity field which in turn distorts the temperature field, [6] and [8]. The secondary flow produces an increase in the wall shear stress towards the corners and significant influence on heat transfer at the walls. Until recently, experimental investigation of these cases was confined to comparatively simple cases with symmetrical boundary conditions. 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 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 for smooth duct has been carried out for ten Reynolds numbers varying from $5x10^4$ to $1x10^5$. The gathered data will be useful to those pursuing the task of numerical prediction in this area of research and development. The object is to provide a good understanding and compact correlations obtained, include Pr as a variable parameter, can be used for improved numerical analysis and for better design of equipments for engineering applications.

2. LITERATURE SURVEY

In the past decades, several researchers measured the primary flow velocity and got some useful results (i.e., [6], [8], [11], [12], and [20]), found that the length,



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

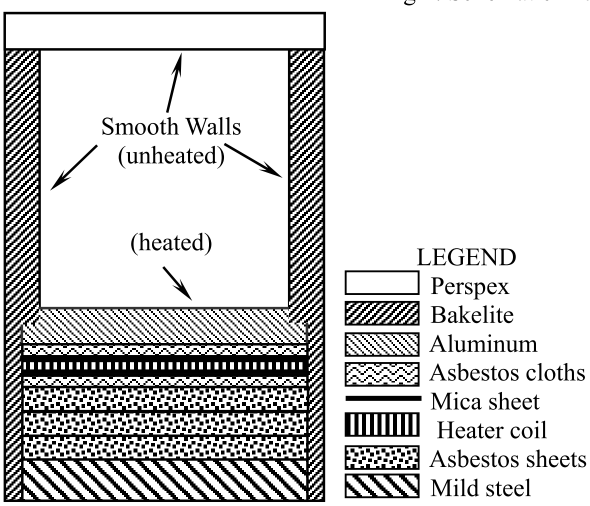


Fig: 2 Illustrating the Cross sectional View of the Duct

required for full flow development in a pipe may exceed 140 diameters. According to this viewpoint, only in a few experiments of fully developed flow has been achieved [5], [6], [7], and [11]. The turbulent flow as well as the temperature field in non-circular ducts is influenced by the existence of the secondary flow, [8], and [10]. 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], [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, [5] and [12].

3. EXPERIMENTAL SET UP AND METHODOLOGY

A schematic diagram of the straight experimental

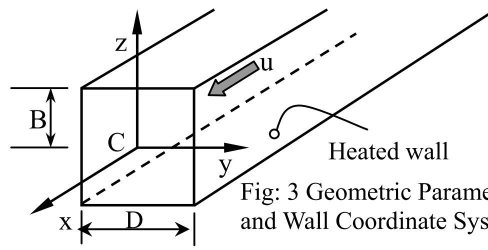


Fig: 3 Geometric Parameters and Wall Coordinate System

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].

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

$$q = Q/A_c = C_pG(\Delta T_{ln}) \quad (1)$$

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] and [15]. 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

$$\text{where, } \theta = (T_{wc}-T) \text{ and } \theta_c = T_{wc}-T_c. \quad (5)$$

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

$$u^+ = u/u^* \text{ and } T^+ = (T_w-T)/T^* \quad (6)$$

$$\text{where, } u^* = \sqrt{\tau_w/\rho} \quad (7)$$

$$\text{and } T^* = q/\rho C_p u^* . \quad (8)$$

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 [1] and [15] as:

$$Z^+ = u^*Z/\nu, [1] \quad (9)$$

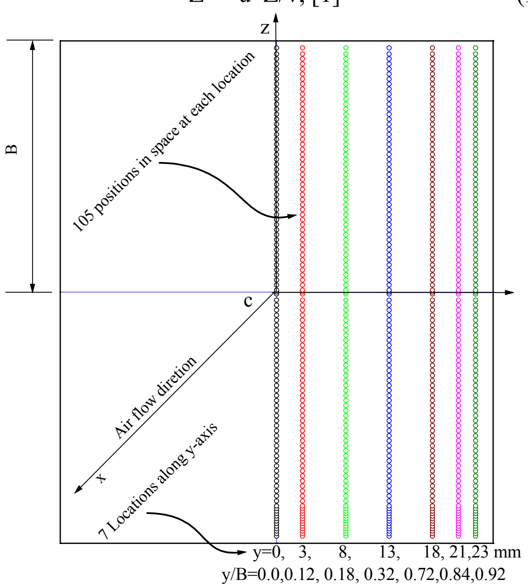


Fig: 4 Dots represents probe positions in space for measurements of velocity and temperature.

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 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 [4] and [15].

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 DISCUSSION

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, [4], [6], and [13]. The measurements are taken for 10 Reynolds number varying between $5\times10^4<Re<1\times10^5$.

7.1 Mean Velocity Profiles

To get the over view regarding the magnitude and the distribution of these two important parameters of the flow field, profiles are drawn for the lowest and the highest Reynolds numbers only. The presence of the bottom heated wall creates asymmetric flow field Fig: 5(a). The velocity profiles in the region $0 \le y \le 8$ show little or no influence of secondary flow on axial time mean velocity fields. The velocity profiles of this region show the usual parabolic form having the maximum near the centre slightly towards the heated wall. The velocity gradients in this region and also in other region near the bottom heated wall are higher than those near the top unheated wall. But as the locations, y moves towards side wall the secondary velocity on the primary velocity profiles becomes more and more prominent and the saddle shape form of profiles from its usual parabolic form characterizes its influence. Near the sidewall y>13 the profiles show their saddle shape form in the region -20<z<12.5. This saddle shape velocity profile is due to the generation of secondary velocity in the square duct. 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. It is also noticed that with the increase of Reynolds number, the

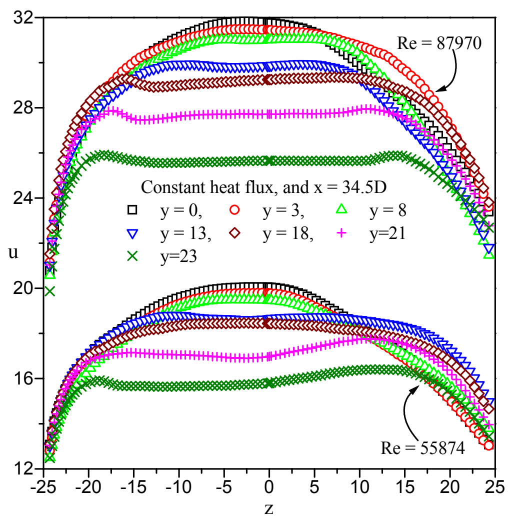


Fig 5(a): Distributions of local time mean velocity profiles in asymmetrically heated smooth duct.

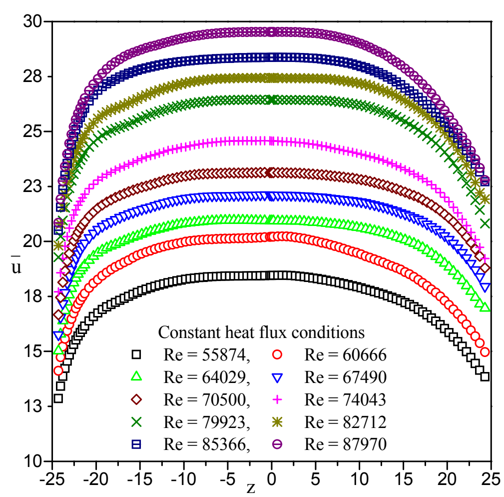


Fig 5(b): Comparison of mean velocity profiles in asymmetrically heated smooth square duct.

degree of saddle shape of the said velocity profiles increases. This is due to the fact that with the increase of Reynolds number the secondary velocity increases carrying more high velocity fluid from the center towards the corner of the duct creating pronounced bulging of the velocity profiles there. The velocity profiles at locations y <8 are different from those near the sidewall 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. It is noticed that velocity peaks of these profiles at y<8 occur slightly towards the heated 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 sidewall. This phenomenon is due to the combined effect of differential wall friction, the temperature gradient of the flow field and the presence of secondary flow in the square duct. The temperature difference changes the viscosity of air thereby reshaping the profile and at the same time the presence of

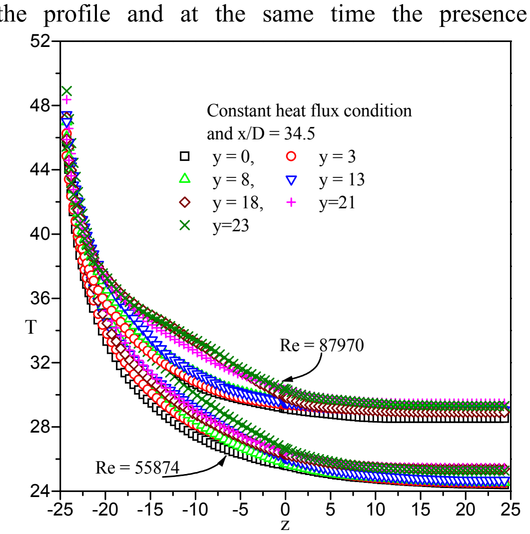


Fig 6(a): Distributions of local temperature in asymmetrically heated smooth square duct.

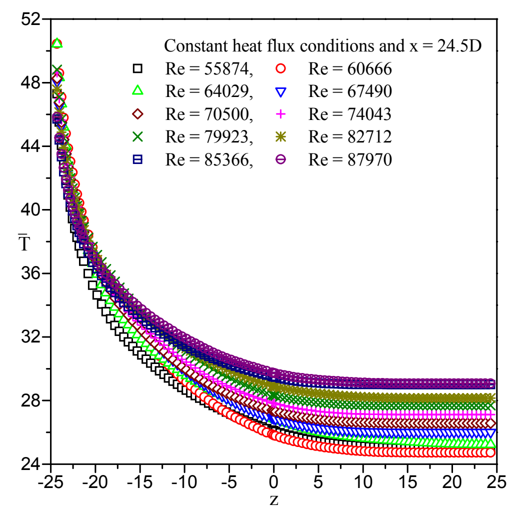


Fig 6(b): Comparison of mean temperature profiles in asymmetrically heated smooth square duct.

secondary flow however small it is in that region, greatly influence the mean velocity profile to its final shape as found in the experiment. The asymmetry of velocity profile is prominent within the boundary (z = ± 15 to 20 mm). Near the side wall (y > 13mm) the local velocity within the range -20<z<15 decreases due to the production of secondary velocity. Fig: 5(a) also shows that for the corresponding location the velocity gradient becomes gradually steeper at higher Reynolds number than that of the lower Reynolds number, higher heat transfer coefficient at the expanse of higher friction factor. Fig: 5(b) shows the Comparison of average of time mean velocity profiles at constant Reynolds number. The profiles shift upward with increase of Reynolds number. It is clearly seen that the velocity gradient is steeper near the bottom heated wall than that near the top unheated wall as well as near the bottom heated wall the velocity gradients are more steeper at higher Reynolds number than that at the top unheated wall.

7.2 Mean Temperature Profiles

As it can be seen from Fig: 6(a) that the air entering the heated test section the temperature remains almost constant in the top half cross section of the duct up to the profiles in asymmetrically heated smooth duct measuring section at x = 34.5D, which means that the duct behave like a flat plate. From z<5 mm the temperature of air starts increasing at the increasing rate until the bottom heated wall where the maximum temperature is reached. The Fig: 6(a) also shows that temperature profile in the region -17>y><0 is distorted indicating the strong influence secondary flow enhancing turbulence intensity and hence more heat transfer in this region. This effect is greater at higher Reynolds number. Fig: 6(b) shows the comparison of average temperate profiles at 10 different Reynolds number. It can also be seen from these plots that although the variation of uncontrolled (room temperature) 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 difference may be due to higher viscosity of air near the heated wall.

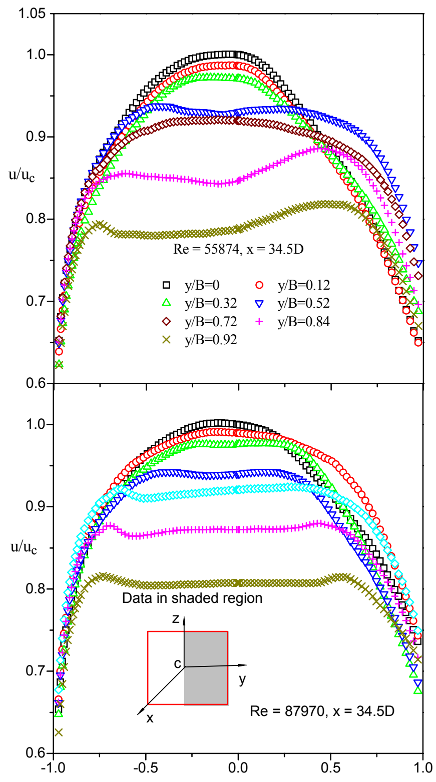


Fig 7(a): Normalized mean velocity profiles.

7.3 Normalized Mean Velocity Profiles

The normalized mean velocity profiles characteristics the behaviour of the velocity field that can be compared with the other researchers more easily because these give the variation in relative terms irrespective of both magnitudes of velocity and size of the duct. Along with this comparison the analysis of these curves will be universal. A shown in Fig: 7(a) corresponding profiles of all the figures exhibit the similar characteristics (Fig: 5(a)) with different magnitudes of variations. Similar to Fig: 5(b), the Fig: 7(b) shows the comparison of average of mean velocity profiles at constant pressure drop. But in Fig: 7(b) the normalized velocity profiles for 10 different Reynolds number are shown in the water fall plot for clear view since they overlap when plotted in a normal graph of linear axes.

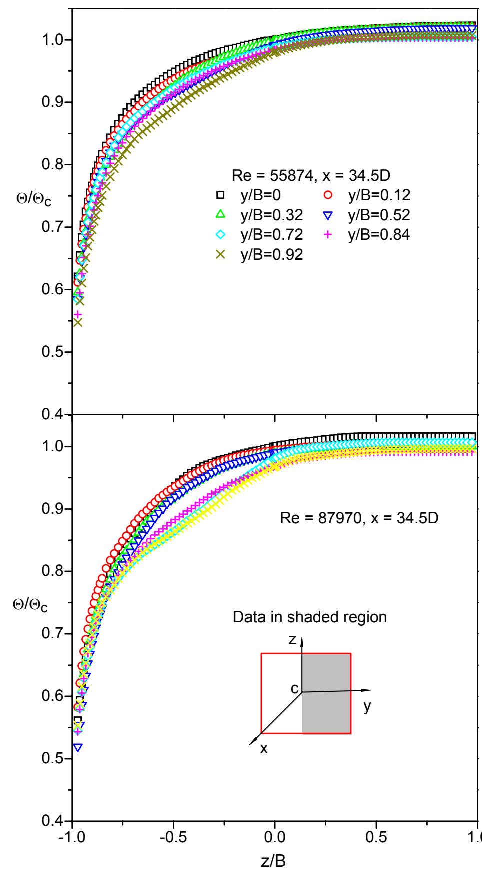


Fig 8(a): Normalized mean temperature profiles.

7.4 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: 7(a) for the lowest and the highest Reynolds number. Normalized mean temperature profiles of smooth duct show that Θ/Θ_c increases with decreasing rate up to the center of the duct and beyond that the Θ/Θ_c of air remains nearly the same as that of inlet air showing negligible effect of heating. Normalized mean temperature profiles shown in Fig: 8(a) shows that θ/θ_c increasing with decreasing rate and beyond the centre in the top half of the duct they remain almost constant meaning the air temperature in the top half at the section x = 34.5D is the same as that of the entry temperature. Thus it can be said that the duct behaves like a flat plate showing the little effect of the

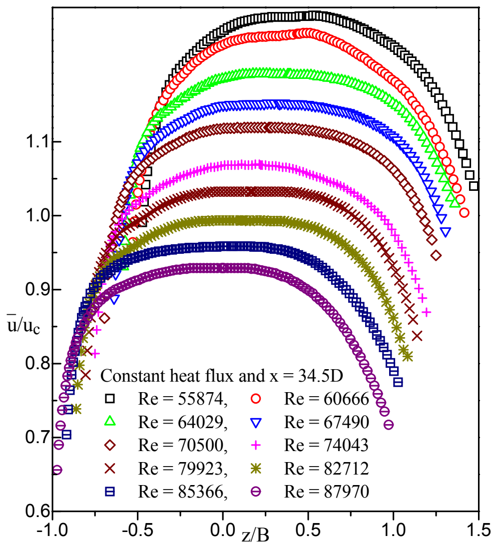


Fig 7(b): Comparison of normalized mean velocity profiles in asymmetrically heated duct.

side and top walls on their shapes. The shape of the profiles in the central part of the duct is similar to that over the flat plate showing the little effect of side wall and the top wall on their shape. The Fig. 8(a) also shows that temperature profile in the region -0.68<z/B<0 is distorted due to secondary flow indicating major heat transfer takes place in this region. This effect is greater at higher Reynolds number. Fig: 8(b) shows the comparison of average of the mean temperate profiles in a water fall plot for clear view since these curves overlap. It can be seen from these plots that although the variation of uncontrolled (room temperature) 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. It is also noticed that for the same heated wall distance the temperature is maximum near the sidewall and minimum at around y/B=0.52. This is due to the presence of secondary flow in the flow field. Comparing the mean velocity curves of Fig: 7(a) and the temperature curves of Figs: 8(a) it is seen that at y/B=± 0.52 velocities near both the heated wall and unheated top wall is comparatively higher than those of the profiles of other y/B locations indicating strong influence of secondary flow in the region of y/B=± 0.52. The influence of this velocity profiles positively influences the temperature profiles carrying high temperature air from the heated wall to this zone producing higher values of Θ/Θ_c. The velocity profiles at higher Reynolds number in Fig: 7(a) shows the higher velocity in the wall region (top and bottom wall) at locations of y/B = 0 and 0.12, than at y/B = 0.52. This is due to pronounced effect of secondary velocity on mean velocity (primary velocity) profiles and thus its effect is reflected on the temperature profiles of Fig: 8(a) showing higher values of Θ/Θ_c in the locations of y/B = 0 and 0.12 instead of y/B = 0.52 for the reasons as discussed above.

7.5 Correlations Between Mean Temperature and Velocity Profiles

The similarity between the nondimensional temperatures and the velocity fields,

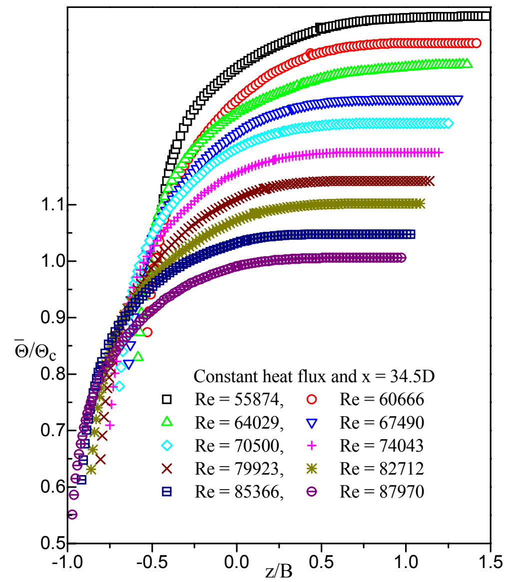


Fig 8(b): Comparison of mean normalized temperature profiles in asymmetrically heated duct.

the values obtained in the region –1 (z/B) ≤ 0 and 0 ≤ (y/B) ≤ 0.92 at the cross section x =
34.5D downstream from the inlet of the heated section for the lowest and the highest Reynolds number are shown in Fig: 9(a). The temperature profile is in perfect agreement with that of velocity then, Θ/Θ_c is equal to u/u_c everywhere and the correlation follows the straight line, [20]. 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, [14]. Fig: 9(a) shows that the results fall on a straight line. This demonstrates that Θ/Θ_c, correlates highly with u/u_c, and suggests the similarity between the temperature and the velocity fields. The corresponding ratios of Θ/Θ_c are smaller than those of u/u_c for $\Theta/\Theta_c \le 0.85$. Fig: 10(a) also shows that the degree of similarity decreases from the centre towards side walls. Fig: 9(b) shows the comparison of correlations between the temperature and velocity fields. Except the lowest Reynolds number 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. In the present investigation the slopes of the curves ranges lie between 1.418 and 1.096. In a similar experiment, [15] found the slope of 1.34 of similar plot for Reynolds number $6.5\text{x}10^4$. The present values are very close to that found by, [14] and [22] for smooth duct was 1.22. These slopes indicate the degree of similarity between the temperature and velocity fields. The minimum slope of 1.096 is found at the highest Reynolds number of the present

measurement set indicating maximum similarity between these two main variables of the flow fields. Though in the experiment there is no systematic decrease of the slope with the increase of Reynolds number but it is expected from the trend of variation of slopes that with further increase of Reynolds number the temperature and velocity fields will approach to perfect similarity

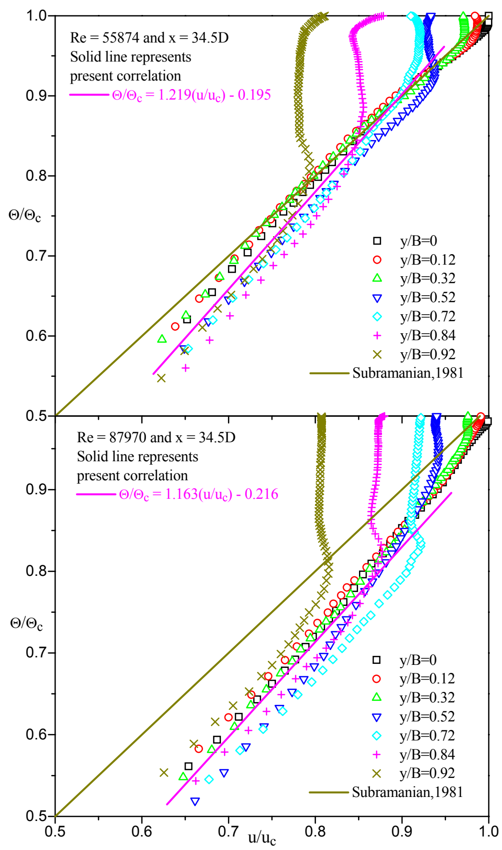


Fig 9(a): Correlations between nondimensional mean velocity and temperature profiles.

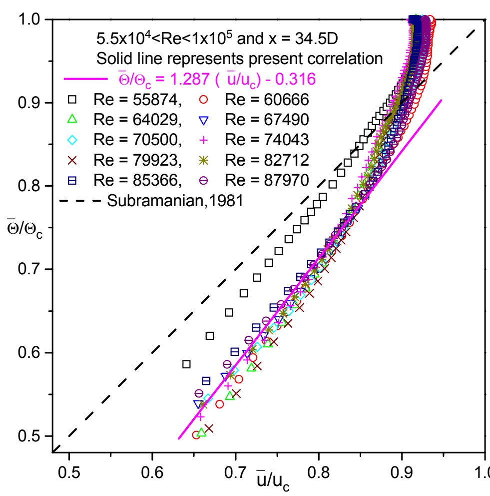


Fig 9(b): Comparison of average correlations nondimensional velocity and temperature profiles.

condition. In the present investigation the average of the mean correlations obtained is as follows:

$$\Theta/\Theta_c = 1.287(u/uc) - 0.316 \qquad (10)$$

For $5\times10^4<Re<1\times10^5$, x=34.5D

and fully developed flow with constant heat flux.

7.6 Universal Velocity and Temperature Distributions

As both the time mean velocity and temperature of a heated flow field are of some polynomial function of

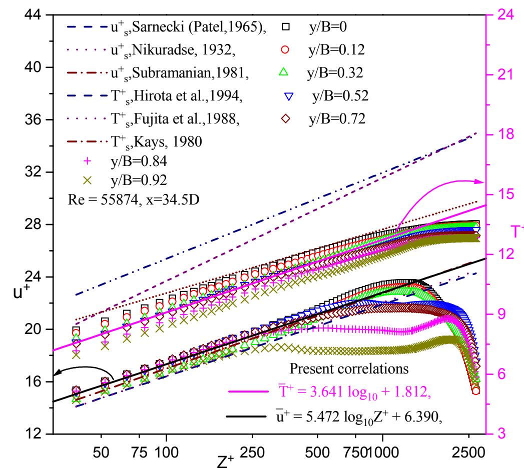


Fig 10(a): Distributions of universal velocity and temperature profiles in asymmetrically heated duct.

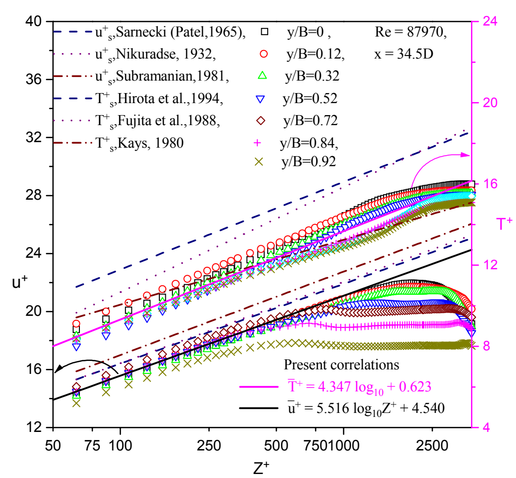


Fig 10(b): Distributions of universal velocity and temperature profiles in asymmetrically heated duct

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 smooth wall flow field is greatly influenced by the wall shear stress, τ_w and air density, ρ and with the increase of air of temperature the density of air decreases. 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 Figs: 10(a) and 10(b) for the lowest and highest Reynolds number of the present investigation. In these figures, the experimental values for different y/B locations are plotted and their mean values are shown by the solid lines while broken lines represent the published data by other well known researchers as indicated in the figure. As shown in these figures the time mean profiles in the turbulent part of the wall region, $30< Z^+< 230$, the time mean velocity profiles are in good agreement with the logarithmic law velocity profiles proposed by the published data. The data obtained can be expressed by the following equation as follows:

$$u^+ = 5.733 \log_{10} Z^+ + 5.002 \qquad (11)$$

Fully developed region, Constant heat flux and 5×104<Re<1×105.

The results of T^+ for each location of y/B fall on a line in the turbulent part of the wall region $30< Z^+<230$, where the inner law is generally valid, [15], and [16]. This indicates that the inner law expressed by the logarithmic formulation is valid for the mean temperature distribution for the smooth square duct heated asymmetrically with constant heat flux. The following logarithmic law is obtained and is represented by the solid line in Fig: 10(b):

$$T_s^+ = 4.638 \log_{10}Z^+ - 0.464 \qquad (12)$$

Fully developed region, Constant heat flux and $5\times10^4<Re<1\times10^5$.

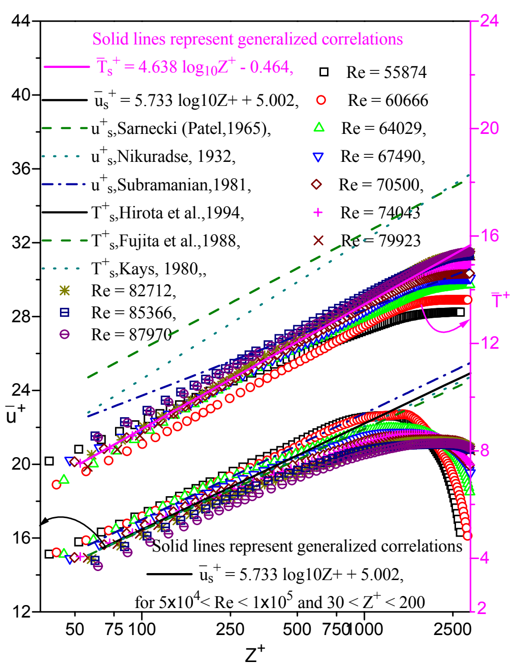


Fig 10(c): Comparison of universal velocity and temperature profiles in asymmetrically heated duct

The distribution of Eqn. (10) is similar to that of obtained by others, e.g., [8], in a smooth square duct. The present results lie on a straight line and agree with those of other researchers with certain variations. This variation is due to the different experimental setup and methodology different boundary conditions, [10] and [15]. Their curves are found to be on higher level than the present work. This difference however decreases with the increase of Reynolds number for T^+. The curves also show nearly the same slope as found by the other researchers e.g., [10], [15], and [17]. The higher temperature of their experiment may be due to the fact that in their case all the four walls of their test ducts were heated while in the present experiment only the bottom wall is heated.

The semi logarithmic plot of mean velocity shows that they lie on a straight line indicating that the present experimental values obey the universal velocity distribution law. Fig: 10(c) shows the comparison of the universal distributions of mean velocity and temperature for the range of Reynolds numbers studied. The figure shows the little effect of their distribution due to the variation of Reynolds number.

8. ACKNOWLEDGMENT

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9. REFERENCES

1. Abdul Hamid, A.K.M.; 2004, “Experimental Study on Convective Heat Transfer with Turbulence Promoters”, Ph.D. thesis, Bangladesh University of Engineering and Technology, Dhaka, Bangladesh.
2. Abdul Hamid, A.K.M.; Akhanda, M.A.R.; and Taher Ali, M.A., 2003, “An Experimental Study On Forced Convective Heat Transfer In Asymmetric Duct Flows With Periodic Turbulent Promoters.” 3rd International Conference of Mechanical Engineers and 8th Annual Paper Meet on E-Manufacturing, *Mech. Engg. Division, IEB,* 20-22. Paper No. 28, pages: 07-215.
3. Abdul Hamid, A.K.M.; and Taher Ali, M.A., 2004, "Characteristics of Mean Velocity and Mean Temperature Fields for a Turbulent Flow in an Asymmetrically Heated square Ribbed Duct." 4th International Conference of Mechanical Engineers and 9th Annual Paper Meet on E-Manufacturing, *Mech. Engg. Division, IEB,* 29-31, Paper No7, pp32.
4. Abdul Hamid, A.K.M.; and Taher Ali, M.A., 2007, "Characteristics Of Air Properties And Variable Parameters For Turbulent Flow In An Asymmetrically Heated Smooth Square Duct." th International Conference of Mechanical Engineers - 2007
5. Akhanda, M.A.R., 1985, “Enhanced heat transfer in forced convective boiling,” Ph.D Thesis, University of Manchester, Institute of Science and Technology.
6. Ali, M. T., 1978, “Flow through square duct with rough ribs,” Ph.D. Thesis, Imperia College, University of London, U.K.
7. Brundett, E., and Baines, W.D, 1964, "The Production and Diffusion of Vorticity in the Duct Flow," *J. Fluid* , Vol. 19, pp. 375-394.
8. Brundrett. E. and Burroughs, P.R. 1967, "The Temperature Inner-Law and Heat Transfer for Turbulent Air Flow in a Vertical Square duct," *Int. J. Heat Mass Transfer*., Vol. 10 1967, 1133.

9. Emery, A.F., Neighbors, P.K., and Gessner, F.B., 1980, “The Numerical Prediction of Developing Turbulent Flow and Heat transfer in a Square Duct.” JSME International Journal, Vol.102, pp. 51-57, 1980.
10. Fujita, H., Yokosawa, H., Hirota, M. and Nagata, C., 1988, "Fully developed turbulent flow and heat transfer in a square duct with two rough ended facing walls", Chemical Engineering Communications, Vol. 74, pp. 95-110.
11. Gessner, F.B., 1964, "Turbulence and Mean-flow Characteristics of Fully Developed flow in Rectangular Channels," Ph.D. Thesis, Dept. Mech. Engg. Purdue University.
12. Gessner, F.B., and Emery, A.F., 1981, "A Length-Scale Model for Developing Turbulent Flow in a Rectangular Duct," ASME Journal of Fluids Engineering, 1981, Vol. 103, pp. 445-455.
13. Han, J.C. Ou, S., Park, J.S., and Lei, C.K., 1989, "Augmented Heat Transfer in Rectangular Channels of Narrow Aspect Ratios With Rib Turbulators, "International Journal of Heat and Mass Transfer, Vol. 32, No. 9, pp. 1619-1630.
14. Han, J.C., 1984, "Heat Transfer and Friction in Channels with Two Opposite Rib-Roughened Walls," ASME Journal of Heat Transfer, Vol. 106, No. 4, pp. 774-781.
15. Hirota, M., Fujita, H., and Yokosawa, 1994, “Experimental study on convective heat transfer for turbulent flow in a square duct with a ribbed rough wall (characteristics of mean temperature field),” *ASME Journal of Heat Transfer*, Vol. 116, pp.332-340.
16. Hishida, M., Nagano, Y. and Shiraki, A., 1978, "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 1978, 3145.
17. Kays, W.M., and Crawford, M.E., 1980, “Convective Heat and Mass Transfer”, McGraw-Hill, New York.
18. Kiline, S.J., and McClintock, F. A., 1953, "Describing Uncertainties in Single-Sample Experiments," Mechanical Engineering, Vol. 75, pp. 3-8.
19. Komori, K., Iguch, A., and Iguni, R., 1980, "Characteristics of fully developed Turbulent flow and Mass Transfer in a Square Duct.” Int. Chem, Eng, 20. (2), 219-225.
20. Melling, A., and Whitelaw, J.H., 1976, "Turbulent flow in a rectangular duct ,” J.Fluid Mech., Vol. 78, part 2, pp. 289-315.
21. Subramanian, C. S., and Antomia, R. A., 1981, “Effect of Reynolds number on a slightly Heated Turbulent Boundary Layer.” *Int. J. Heat Mass Transf.*, Vol. 27No. 11, pp 2133- 2144.
22. Yokosawa, H., Fujita, H., Hirota, M., and Iwata, S, 1989,"Measurement of turbulent flow in a square duct with roughened walls on two oposite sides." *Int. J. Heat Fluid Flow.*, 2, 125-130.

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
G	Mass flux	Kg/m^2s
L	Length	m
P	Pressure	N/m^2
Q	Heat transfer	W
q	Heat flux	W/m^2
u	Time mean velocity	m/s
h	Heat transfer coefficient	$W/(m^2, {}^0C)$
K	Thermal conductivity	$W/(m, {}^0C)$
Re	Reynolds' number	Dimension less
T	Mean temperature	^0C
T*	Friction temperature	^0C
T^+	Mean universal temperature of air	Dimension less
u*	Friction velocity	m/s
u^+	Time meanuniversal velocity	Dimension less
x, y, z	Coordinate system defined in Fig: 3 and 4	
τ	Shear stress	N/m^2
ν	Kinematics iscosity	m^2/s
ρ	Density	kg/m^3
Θ	Difference between wall temperature and air temperature	$(T_w - T)^0C$ or K
Θ_c	Difference between wall temperature and air temperature at centre of duct	$(T_w - T_C)^0C$ or K
a	Ambient temperature	
B	Bulk mean	
C	Centre of duct cross section	
s	Smooth duct, surface	
f	Fluid	
i	Inlet	
in	Input	
ln	Log mean	
L	Loss	
m	Mean	
O	Outlet	
w	Wall	