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CHARACTERISTICS OF AIR PROPERTIES AND VARIABLE PARAMETERS FOR TURBULENT FLOW IN AN ASYMMETRICALLY HEATED SMOOTH SQUARE DUCT

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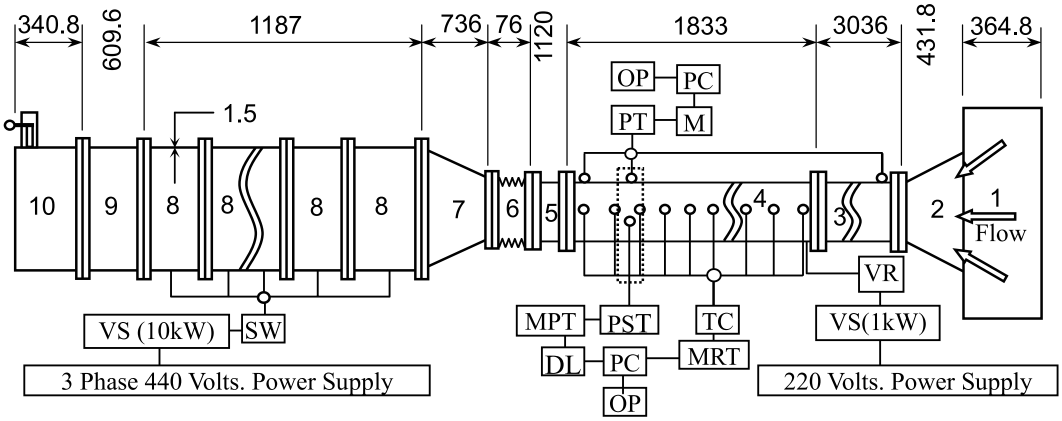
ABSTRACT
This paper presents distributions of air properties and related variable parameters for turbulent flow through an asymmetrically heated smooth square duct at constant heat flux boundary condition. The experimental setup, instruments, and boundary conditions are described first. The distributions different variable parameters studied typified by a symmetric and an asymmetric flow pattern based on the time mean velocity temperature, duct length and wall coordinates axes. The effects of Reynolds number, secondary flow, probe positions in space at different locations across the duct are studied. Finally, equations and correlations are developed to describe how these important parameters vary with each other. The assumption of constant Pr everywhere in the cross section is not valid and is completely inapplicable for turbulent flow through noncircular ducts. Thus the improved models of turbulent heat transfer that do not include the turbulent prandl number must incorporate it for the correct prediction of heat transfer for turbulent flow. Compact correlations are developed. These relations can be used to improve numerical analysis and better design of heat transfer equipments for engineering applications.

Keywords: Variable Parameters, Secondary Flow, Heat Flux.

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. No doubt this technique considerably reduces the number of variables involved in the experimental investigations but does not clearly help how the characteristics of each individual variable varies with respect to other parameter in complex situations. A brief literature survey indicates the lack of reliable experimental data on the characteristics of individual variable in a square duct heated asymmetrically with constant heat flux. A very few related experimental studies have also been conducted that provide information on velocity and temperature distributions in the cross section of a square duct. Hence, air properties and some of the most important variable parameters have been studied experimentally. The object is to provide a good understanding that how the individual variable varies with respect to other variable parameters for a turbulent flow of air through an asymmetrically heated smooth square duct with constant heat flux boundary condition.

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, [1]; [5]; [6] and [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 can not be ignored, [3]; [5]; [6] and [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, [3]; [4]; [5] and [7]. The literatures on air properties and different variable parameters for a turbulent flow even in a simple fundamental situation are very rare. Hence it is worth while to investigate experimentally how the secondary flow affects air properties as well as the characteristics of individual variable parameters varies with each other.



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

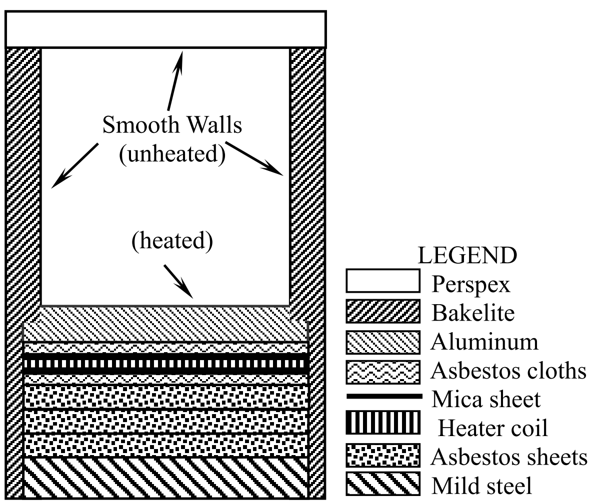


Fig : 2 Illustrating the Cross sectional View of the

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. 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 50mm×50mm consists of heated section of length 1833mm (=36.66D) and unheated section of 3036mm (=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 1220mm long having the same cross sectional dimensions as that of test duct is attached at the

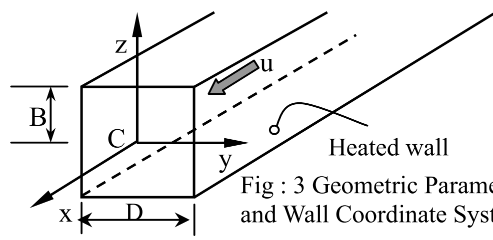


Fig : 3 Geometric Parameters and Wall Coordinate System

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 12mm thick to provide both high strength and to ensure no leakage of current. The top wall is made from transparent plexiglass plate of thickness 12mm in order to provide optical access for observation and necessary adjustment of probes. The entire test duct except the 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 in order to achieve uniform wall heat 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 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.

4. MEASUREMENT SYSTEM

The configuration, the dimensions of the test model, the flow direction, and the coordinate system are schematically shown in Fig: 3. 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 a 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.

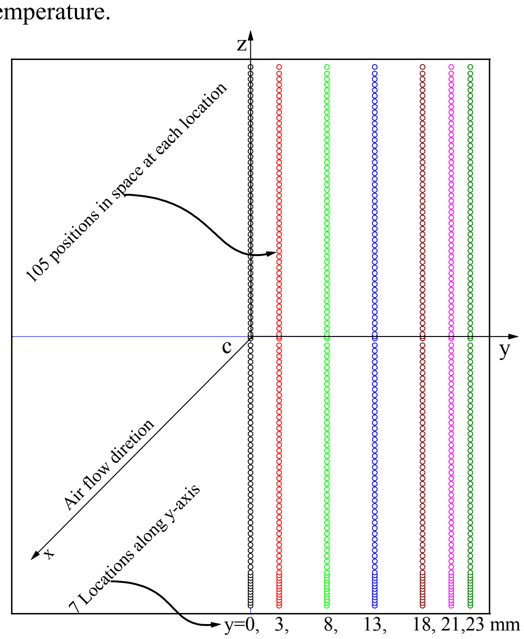


Fig 4: Dots represents probe positions in space for measurements of velocity and temperature.

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

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

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

The Log Mean outlet temperature difference of air 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:

$$T_{wc} = T_w - (Q\delta/k\,A_s) \quad (2)$$

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

$$h = q/(Twc\text{-}Tb) \quad (3)$$

The density, viscosity, and specific heat are calculated by interpolation for the corresponding pressure drop and temperature measurements data from experiment.

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. So the measurements are taken only in the 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,[5]; [10] and [11]. 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 [1].

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 velocity, $u_b = ½(u_o - u_i)$ [10]. 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 DISCUSSION

7.1 Wall and Air Temperature Distributions Along Length of Duct

Fig: 5 illustrates the distributions of heat ,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 as stated by many investigators, [5] and [9]. The figure shows the linear increase of bulk mean air temperature measured from the inlet to the outlet. At constant Re, the wall temperature increases at a decreasing rate approaching the asymptotic condition near x = 15D and after this the. The longitudinal distribution of bulk mean air temperature increases linearly up to the last station of measurement at x = 34.5D, [10]. The temperature increases linearly can be expressed by the equation:

$$\overline{T}_b = 25.715 + 0.121(x/D)^0C. \quad (4)$$

The heat supplied into the duct can be expressed by a polynomial equation of the form:

$$\overline{T}_{wc} = 53.5 + 2.42(x/D) - 0.09(x/D)^2 + 0.0011(x/D)^3 \quad (5)$$

for $5\times10^4 < Re < .1\times10^5$ and x = 34.5D.

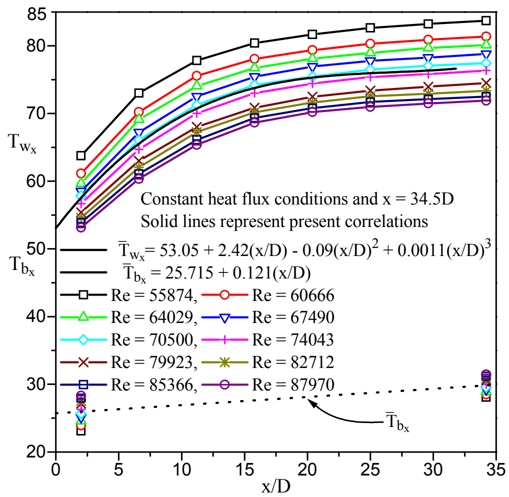


Fig :5 Wall and air temperature distributions along length of the duct.

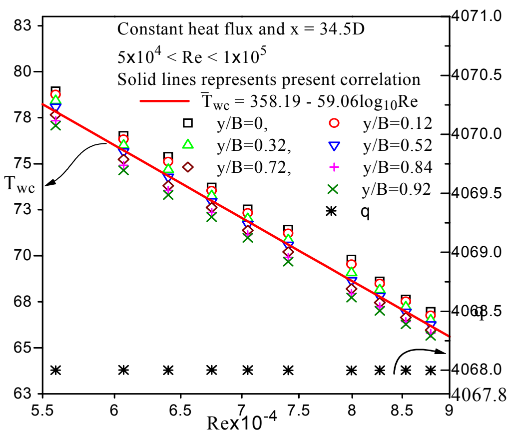


Fig :6 Effects of Reynolds number on local wall temperature and on local heat flux.

The temperature profiles in the figure show continuous shifting downward with the increase of Reynolds number confirming increased heat extraction from the wall with the increase of flow velocity. Typically, at downstream distances from x = 15D from the start of heating, as shown in Fig: 5, the wall temperature become nearly parallel to aforementioned bulk mean air temperature straight line. According to [5] and [9], for 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. Fig: 6 illustrates the effect of Reynolds number on local corrected wall temperature, T_{wc} and on heat flux, q. At constant heat flux, the wall temperature drops linearly with increase of Reynolds number according to semi-empirical correlation obtained as follows:

$$\overline{T}_{wc} = 358.19 - 59.06\log_{10}Re \quad (6)$$

and also the curve shifts downward with increase of y/B which means that the heat extraction by flowing air increases with increase of Reynolds number as well as with the increase of wall distance from centre towards the side walls at constant heat flux boundary condition. This effect is clearly shown in Fig: 7(a) and (b), where the wall temperature is compared with bulk mean velocity and temperature of air.

7.2 Local Wall Heat Flux Distribution

The heat supplied into the bottom wall of the test duct are measured thoroughly by 8 thermocouples evenly distributed. The local wall heat flux, q is calculated by applying Eqn. (1). As shown in Fig: 6 and Fig: 12 the local heat flux remains constant with the increase of Re. This is the boundary condition for the present experimental study of air properties and different variable parameters [1].

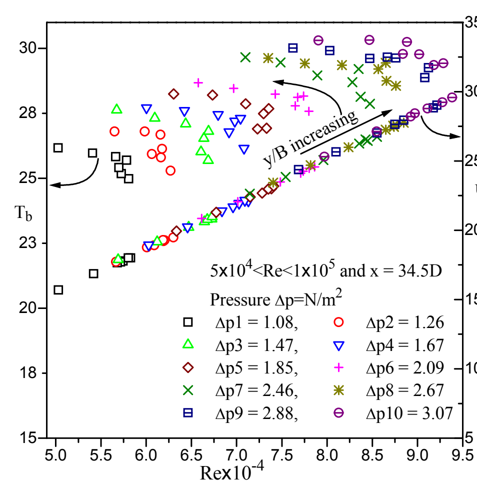


Fig: 7(a) Distributions and comparisons of local bulk mean velocity and temperature of air.

7.3 Local Air Velocity and Temperature Distributions

At constant pressure drop the local air temperature drops with increases air velocity as expected and also both curves shift upward with increase of Reynolds number, Fig: 7(a), indicating strong influence of secondary flow around the bisector of corner of the duct. The effects of Reynolds on velocity and temperature and their comparison can be seen in Fig: 7(c), where both velocity and temperature increase with increase of Reynolds number but the velocity and temperature curves shift downwards and upwards respectively as expected. This is because at higher velocity both the turbulent intensity and secondary flow increase helping more mixing up of air there and thus increasing more extraction of heat from the wall by the flowing air. Compact correlations obtained are expressed as follows:

$$\overline{T}_{wc} = 72.186 - 1.821(y/B) \quad (7)$$

$$\overline{T}_b = 21.12 + 3.08(y/B) - 4.12(y/B)^2 + 2.59(y/B)^3 \quad (8)$$

$$\overline{T}_b = 1.34\times10^{-4}Re - 18.12 \quad (9)$$

$$\overline{u}_b = 24.422 - 2.885(y/B) \quad (10)$$

$$\overline{u}_b = 3.34\times10^{-4}Re - 1.32 \quad (11)$$

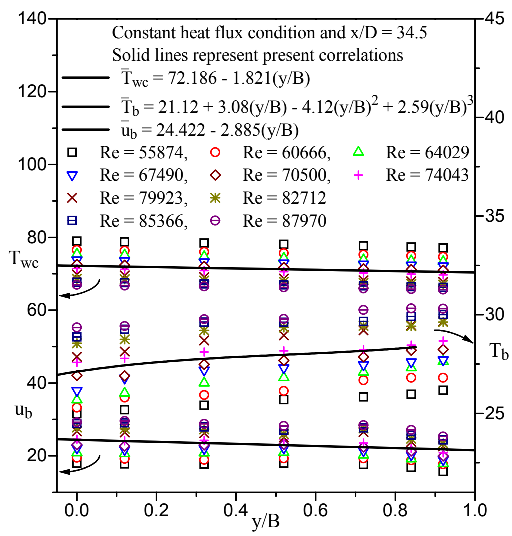


Fig :7(b) Comparisons of local wall temperature, bulk mean velocity and temperature of air.

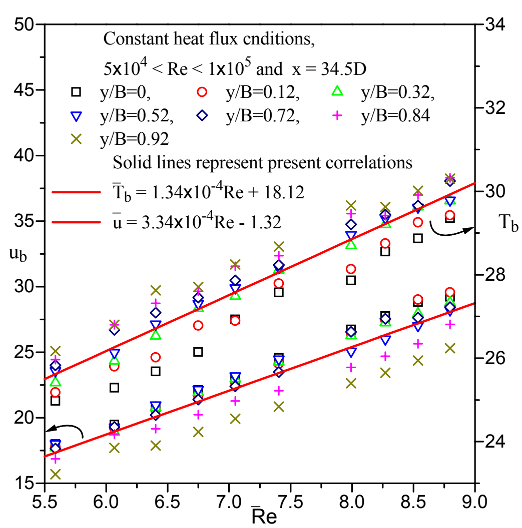


Fig :7(c) Effects of Reynolds number on local bulk mean velocity and temperature of air.

7.4 Local Air density, viscosity, and specific heat Distributions

Fig: 8, 9, and 10 illustrate the characteristics behavior of density, viscosity, and specific heat of air in an asymmetrically heated smooth square duct at constant heat flux. These variable parameters vary according to the equations in the respective plots. Fig: 8 shows as expected the local density decreases with the increase of both the Reynolds number and y/B because the heat extraction by the flowing increases for the reasons explained above. Fig: 9 and 10 show that with increase of Reynolds number both viscosity and specific heat

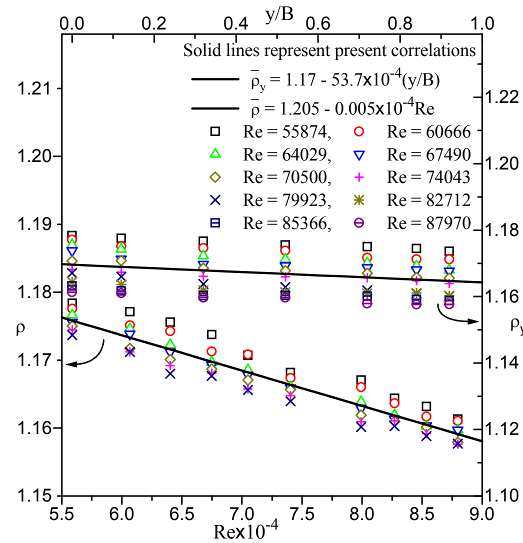


Fig :8 Variations of density of air with Reynolds number across duct heated asymmetrically.

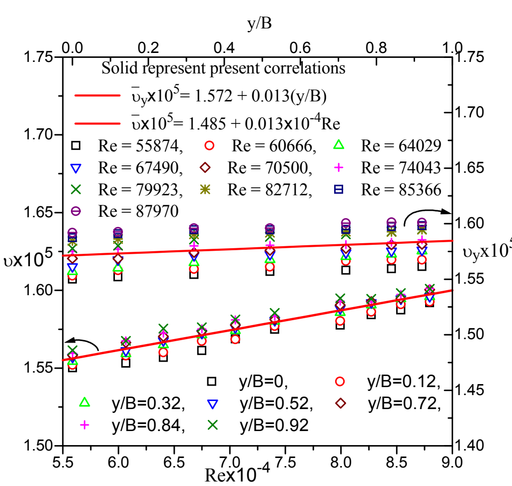


Fig :9 Variations of viscosity of air with Reynolds number across duct heated asymmetrically.

increase and with the increase of y/B both curves shift upwards. Fig: 10 shows that the viscosity increases with increase of Reynolds number and y/B that is the increase of viscosity means the increase of friction at the expanse of energy loss. These variable parameters vary according to the following expressed as:

$$\overline{\rho}_y = 1.17 - 53.7\times10\text{-}4(y/B) \quad (12)$$

$$\overline{\rho} = 1.205 - 0.005\times10^{-4}Re \quad (13)$$

$$\overline{\upsilon}_y\times10^5 = 1.572 + 0.013(y/B) \quad (14)$$

$$\overline{\upsilon}\times10^5 = 1.485 + 0.013\times10^{-4}Re \quad (15)$$

$$\overline{C}_{py}=1004.91+0.17(y/B)-0.228(y/B)^2+0.141(y/B)^3 \quad (16)$$

$$\overline{C}_p = 1004.41 + 0.075\times10^{-4}Re \quad (17)$$

7.5 Local heat transfer coefficient Distribution

Fig: 11 and 12 show the distributions of local heat transfer coefficient with Reynolds number across the duct at constant pressure drop. The heat transfer coefficient varies similar to that of bulk mean temperature of air Fig: 7(a). The Fig: 11 and 12 also show that both curves shift upwards with the increase of y/B and the Reynolds number. This reflects that at higher velocity both the turbulent intensity and the production of secondary flow increase and hence increasing heat transfer according to the following Eqns.:

$$\bar{h}_y = 1004.91 + 0.17(y/B) - 0.228(y/B)^2 + 0.141(y/B)^3 \quad (18)$$

$$\bar{h} = 17.902 - 0.0011\ \bar{R}e \quad (19)$$

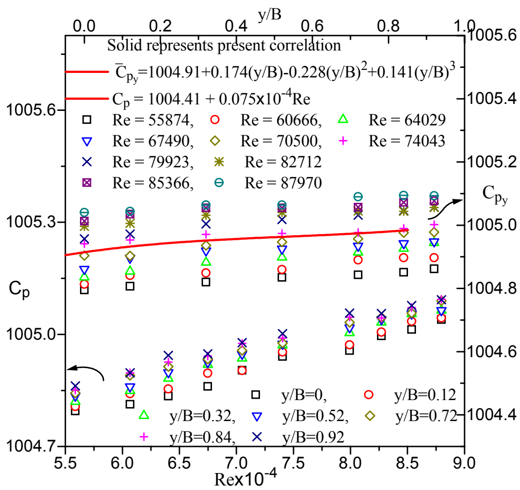


Fig :10 Variations of specific heat of air with Reynolds number across duct heated asymmetrically.

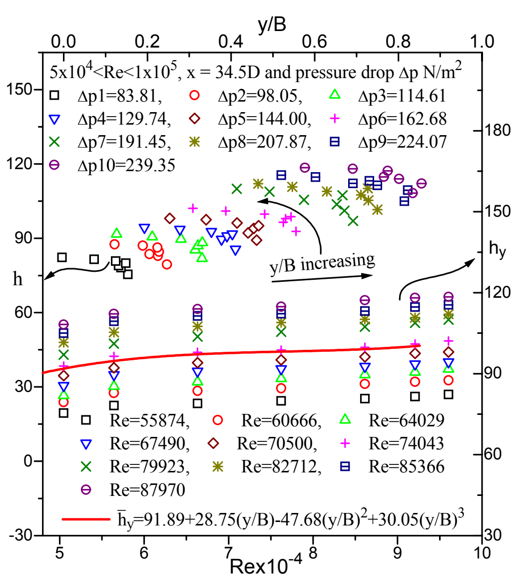


Fig :11 Distributions and effects of Reynolds number on local heat transfer coefficient across duct.

7.6 Local Prandtl number Distribution

In this experimental investigation Prandtl number for air is found to be less than 1 as expected but it does not remain constant as shown in the Fig: 13. Both with the increase of Re and y/B heat transfer from wall to air increasing as shown in Fig: 7(a) and (b) and hence the molecular diffusion by energy transfer is dominating in Pr. Thus with the increase of both Re and y/B decreases the Pr decreases, Fig: 13. The Pr obtained in this experimental investigation is expressed as function of Re

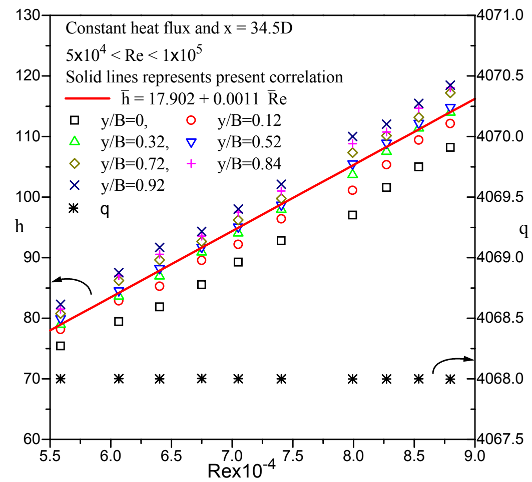


Fig :12 Effects of Reynolds number on local heat Transfer coefficient and heat flux across the duct.

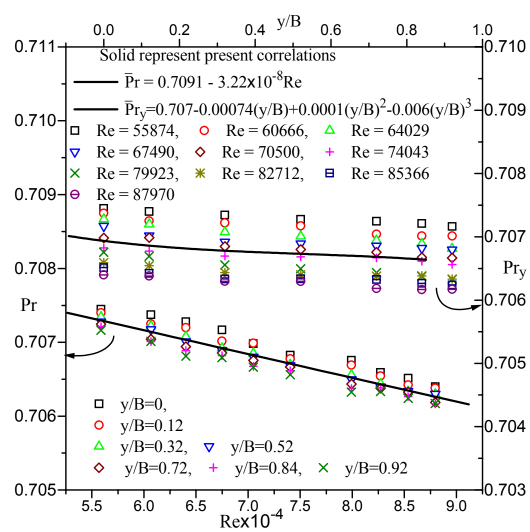


Fig :13 Variations of Prandtl number of air with Reynolds number across duct heated asymmetrically.

and y/B across the duct as follows:

$$\bar{P}r = 0.7091 - 3.22\times10^{-8}Re \quad (20)$$

$$Pr = 0.707 - 0.007(y/B) + 0.0001(y/B)^2 - 0.0006(y/B)^3 \quad (21)$$

These results leads to the conclusion that assumption of constant Pr everywhere in the cross section as generally used in numerical analysis of temperature fields, [10] is not valid and completely in applicable for turbulent flow through both circular and noncircular ducts. Thus improved models of turbulent heat transfer that do not include the turbulent prandtl number must incorporate it for the correct prediction of heat transfer for turbulent flow.

8. ACKNOWLEDGMENT

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10. NOMENCLATURE

Symbol	Meaning	Unit
A	Area	m^2
B	Half of width of duct (= D/2)	m
C	Specific heat, Centre	(W.s/kg^{O}C)
D	Hydraulic diameter of duct	m
G	Mass flux, (= ρu)	kg/m^{2}s
h	Heat transfer coefficient	W/m2,^{0}C
K	Thermal conductivity	W/m,^{0}C
L	Length	m
P	Pressure	N/m^2
Pr	Prandl number	dimensionless
Q	Net heat transfer	W
q	Heat transfer rate	W/m^2
Re	Reynolds' number, , (=uD/ν)	dimensionless
T	Mean temperature	^{0}C, K
u	Time mean velocity	m/s
x, y, z	Coordinate system defined in Figs: 3 and 4	
ρ	Density	kg/m^3
ν	Kinematics viscosity	m^2/s
Subscripts		
a	Ambient temperature	
b	Bulk mean	
C	Centre of duct cross section, Corrected	
s	Smooth duct, surface	
f	Fluid	
i	Inlet	
in	Input	
ln	Log Mean	
p	pressure	
L	Loss	
m	Mean	
o	Outlet	
w	Wall	