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STUDY ON FRICTION IN DEVELOPING FLOW THROUGH A SMOOTH SQUARE DUCT.

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Abstract The purpose of this investigation is to study the friction characteristics in the developing flow through a smooth square duct, with asymmetric heating. The duct is made of insulating materials. The bottom wall is made of 12.5mm thick aluminum sheet, and heated supplying an a. c current. Pressure drop between the inlet and the outlet of the contraction varies between 81 to 91 N/m^2, which makes the flow more uniform and flatten and reduces the turbulence level and adverse pressure gradient before entering into the duct. The mean friction factors along the length of the duct for different Reynolds number are investigated. The universal velocity distributions in the flow are examined at three different cross-sections. The result shows that the static pressure drops along the length of the test duct increases with Reynolds number. The friction factor increases with an increase of Reynolds number, but decreases along the length of the duct. From the result, it is seen that, the correlation of friction law varies with a linear expression. On the basis of the experimental data, three empirical equations have been developed which correlates all data generated in the experiments and finally the results have been compared to other relations available in the literature.

1. INTRODUCTION:

Turbulent flow is important in engineering applications because, it is involved in the vast majority of fluid flow and heat transfer problems commonly encountered in engineering practice occur in a straight non-circular ducts; examples include heat exchangers, nuclear reactors, tubomachinery, air conditioning systems, etc.

In narrow channels the boundary layer formation on the side walls interferes with the boundary layer growth at the bed of the channel. Consequently, in the corners the loss of energy is more than that in the rest of the cross-section. A mass-transport mechanism operates to maintain flow in the corners. The cross-flow pattern thus set up to supply more fluid at the corners from the rest of the cross-section is called the secondary flow pattern. These types of secondary flows are always set up in the straight channels. In general, the secondary flow is set up by virtue of the transverse pressure gradients, set up in the main flow and thus retards the flow velocity.

The fluid flow through channel is characterized by the occurrence of secondary flow in the cross-sectional plane, which results from the anisotropy of Reynolds normal stresses in the cross-sectional plane, induced by the three-dimensionality of the flow. Although the magnitude of turbulence-driven secondary motion is only of the order of 2–3 percent of the streamwise mean velocity. This motion causes the streamwise mean velocity and temperature fields to be distorted considerably towards the corners and thus can have an important consequence. For example, both the local wall shear stress and the local wall heat flux along the duct periphery are dominated by the presence of the secondary flow, which causes these variables to increase initially towards the corner. Therefore, it is important, from a design standpoint, to be able to predict local fluid flow and heat transfer behavior accurately in straight non-circular ducts.

Study on velocity distributions in turbulent flow was very complicated. No fundamental theory is yet available to determine the velocity distribution rigorously by purely theoretical approaches. Therefore, empirical and semi-empirical relations are used to correlate the velocity field in turbulent flow.

Nikuradse J has presented careful measurement of velocity distribution in turbulent flow through a smooth pipe. Laufer.J has performed an experimental study with the structure of turbulence in fully developed pipe flow and tried to develop empirical relations that would fit for the velocity distribution in turbulent flow.

2. EXPERIMENTAL SETUP:

An experimental facility has been designed, developed and manufactured to study of frictional characteristics in the hydrodynamic entry region as shown in Fig.1. Air at room temperature and atmospheric pressure is introduced into the square duct after flowing through a contraction. The cross-sectional dimension of the duct is 50mm×50mm and its hydraulic diameter (D) is 50mm. It is ascertained that the fully developed velocity profiles are attained about 50D (2500mm) downstream from the entrance of the developing duct. The definition of co-ordinate systems and geometric parameters are shown in Fig.2

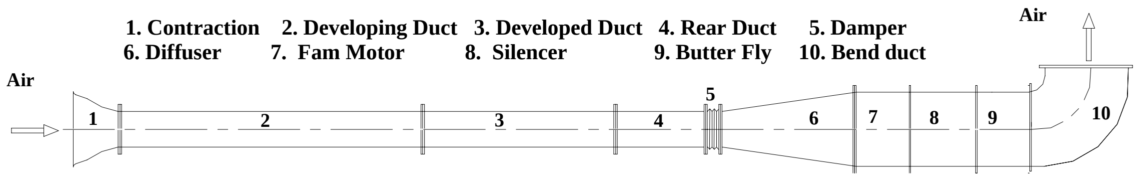

Fig.1 The schematic diagram of the experimental apparatus

The length of the developing duct is 3000mm. The test duct is made of insulating materials. Bakelite sheets of 16mm thick are used to make its side walls and supporting wall. The top wall is made of 12.5mm thick clear perspex sheet and is fastened with the side walls by means of Allen bolts. The smooth wall is made of 12.5mm aluminium sheet as shown in Fig.3. The lengths of the developed duct and the rear duct are

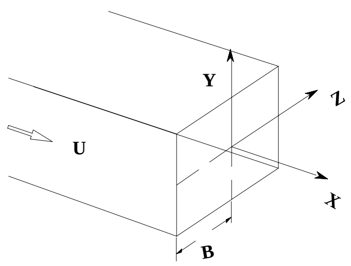

Fig.2 Definition of coordinate systems.

1850mm and 925mm respectively.

A flexible duct (damper) is made of canvas, connected between the outlet of the tailing action minimising duct and inlet of the diffuser. The length of the diffuser is 920mm made of 16-gauge mild steel sheet, and introduced between the outlet of the flexible duct and the inlet of the fan motor. One end of the diffuser is square in size (50mm×50mm) and the other end is circular having 482mm as inside diameter. A flow splitter of length 470mm, made of 16-gauge mild steel sheet is set-up inside the diffuser at a distance of 200mm from its square end to suck air from the duct more conveniently. The flow splitter is actually a right circular truncated hollow cylinder, having 70mm diameter at the smaller end and 285 mm diameter at the larger end. A fan motor having 2.75 hp and 2900 rpm is used to suck air positively and uniformly. The bore diameter and the length of the fan motor are 482mm and 410mm respectively.

A silencer of length 650mm and inside diameter of 482mm is introduced at the outlet of the fan motor to minimise creating sound and vibration. A butter fly having 410mm in length and same bore diameter as the fan motor is set-up with the other end of the silencer to control flow rate of air for measuring the required Reynolds number. At the end of the butter fly, a 90^o-bent duct is used to flow exhaust air to vertically upward. All elements of the experimental set-up are mounted on stands with proper levelling at a convenient height of about 1050mm from the base and different sections are assembled to one another by means of flanges and nuts-bolts.

The bottom surface of the aluminium wall of the test duct is heating uniformly with electricity, supplying an a.c current. For this purpose, an electric heater is designed and constructed by wounding Nichrome flat wire around a mica sheet. The constant heat flux to the bottom surface of the aluminium wall is maintained. Ten pressure-taps are mounted on the side wall, at the same level of 25mm below from its top end to measure the static pressures at wall by means of an inclined manometer. The positions of the pressure taps are exactly at a distance of X/D=3, 9, 15, 21, 27, 33, 39, 45, 51 and 57. Eleven thermocouples are distributed at eleven different positions along the bisector under the bottom surface of aluminium wall to record its out side temperatures.

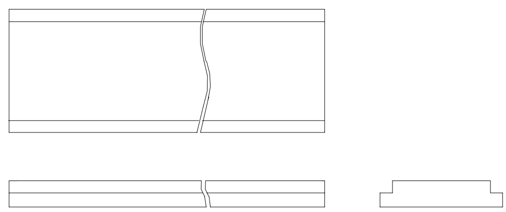

Fig.3 Orthographic views of the smooth wall.

3. RESULTS AND DISCUSSIONS:

The results and their discussions of the present study are given in the following headings.

(i). Pressure Drop Along the Contraction:

Both the inlet and outlet of the contractions are square in sizes, having dimensions of 50mm×50mm and 371mm×371mm respectively and its length is 380mm

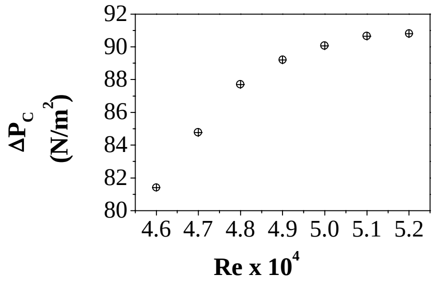

Fig.5 Pressure drop along contraction.

and inlet and outlet area ratio is 55. The contraction is placed upstream of the working section for reducing the flow area and accelerating the flow to the desired velocit by compensating reasonable pressure drop. In the tests, pressure drop occurs between the inlet and outlet of the contraction, varies from 81 N/m^2 to $91N/m^2$ for the corresponding Reynolds number between 4.6×10^4 to 5.2×10^4. Figure 5 shows that the pressure drop is increasing gradually with Re, having a smooth curve path, and entering into the test duct. Thus, contraction is providing some useful functions like the mean velocity profile becomes more uniform and flattened, its boundary layer is subjected to a favourable pressure gradient, relative turbulence level is reduced and avoid reasons of adverse pressure gradient before entering into the duct.

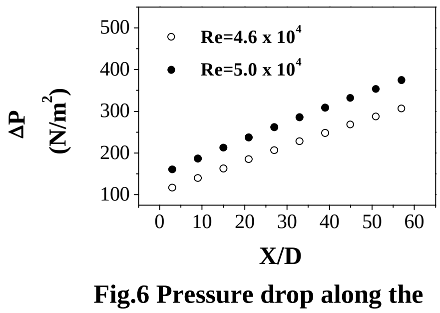

Fig.6 Pressure drop along the length of the test duct.

(ii). Pressure Drop Along the Length of Test Duct:

Figure 6 compares the static pressure drops along the length of the test duct for two different Reynolds numbers. The active length of the test duct is considered to be same of 2.7m (X/D=3 to X/D=57) for all tests. The results show that pressure drops at different Reynolds number increases gradually in the same manner along the length of the test duct, It is seen in the figure, the higher Reynolds number develops higher pressure drop at every corresponding points than the lower Reynolds number.

(iii). Friction Factors Through the Length of the Test Duct:

Figure 7 is an example of the distribution of friction factors against the length of the test duct. The friction factors decrease along a curve profile as increases the length of the duct. It is seen that, initially, the rate of decreasing of the friction factors is too higher up to X/D=15, due to completely unstable flow is prevailing in this range. It is then decreasing with a moderate rate up to X=45. In this range, the flow is being gradually

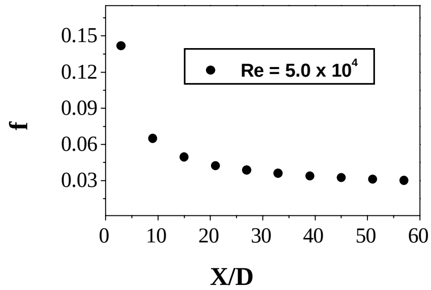

Fig.7 Friction factor along the length of the test duct.

stable. After X/D=45, the flow is very close to stable, and in this range the rate of decreasing of friction factor is very low and closed.

(iv). Effect of Reynolds Number on Friction Factor.

Figure 8 exhibits the effect of Reynolds number on friction factor occurs through the length of the test duct.

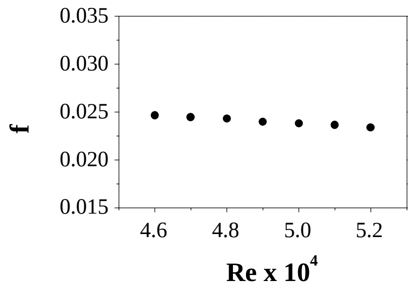

Fig.8 Friction factor on smooth duct for different Re.

It decreases with an increase of Reynolds number. Stanton number is an important parameter of heat transfer characteristic on which heat transfer coefficient is directly related. On the other hand Stanton number increases with an increase of friction factor. So lower the Reynolds numbers and higher the friction factor, up to a reasonable limit of pressure drop penalty, should be the main objective in forced convection heat transfer for obtaining higher heat transfer coefficient, which ultimately gives the maximum efficiency of the system.

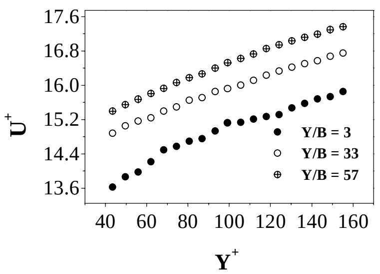

Fig.9 Universal velocity distibutions at three different planes.

(v). Universal Velocity Distributions at Three Different Sections of the Test Duct.

Figure 9 shows the universal velocity distributions at three different planes namely; X/D=3, 33 and 57. The measurement are made along the Y-axis, through the bisector of the bottom heated wall and beginning from the distance of Y/B =-0.972 to Y/B=-0.90 at an interval of 0. 1mm. Three different planes exhibit three profiles of different characteristics, though in all cases universal velocity (U^+) increases with an increase of the dimensionless distance (Y^+). The Lucas of the correlation at X/D=3 is travelling through a zigzag path, which is the indication of unstable flow. The figure

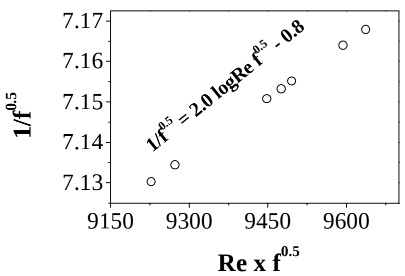

Fig.10 Friction law for turbulent flow through smooth duct (experiment data)

shows that the Lucas of the correlation at X/D=33 is moving also through a zigzag path, but its zigzagness is lower than the former, which is the sign of improvement of flow stability. From the figure it is clear that the Lucas at X/D=57 is moving through nearly a straight path, which means that the flow at that plane is more stable than the others.

(vi). Evaluation of Friction Law .

Figure 10 shows the friction law for the flow in the present investigations. The data for friction factors over the test duct are calculated from different pressure drops and corresponding mean velocities of flow, created for different Reynolds number. From the figure, it is clear that the correlation is varying with a linear expression, which agrees with theoretical approach developed by other researchers for turbulent flow inside ducts.

(vii). Comparisons of Friction Factors of Author's Developed Equation with Others.

Figure 11 shows the comparisons of friction factors among experimental data, approximate analytical equation and author's developed equation. The values of frictions gets from the approximate analytic expression of smooth pipes are much lower than the frictions obtained from experiments in a square duct for various Reynolds number. The regions is that, in a narrow square duct the boundary layer formation on the side walls interferes with the boundary layer growth at the

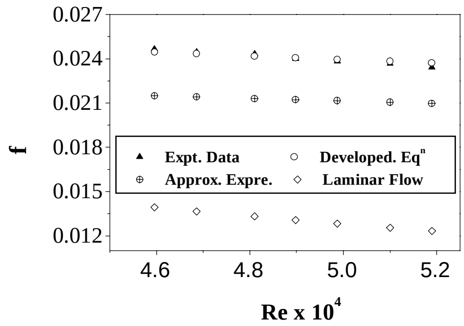

Fig.11 Comparisons on frictin factors among author's developed Eq & others

bed of the channel. Consequently, in the corners the loss of energy is more than that in the rest of the cross-section. A mass-transport mechanism operates to maintain flow in the corners. These types of secondary flows are always set up in the straight channels. In general, the secondary flow is set up by virtue of the transverse pressure gradients, set up in the main flow and thus retards the flow velocity. But in smooth circular pipe it does not occur. For this reason, pressure drop in a smooth square duct is more than a smooth circular pipe, which is the principal factor for increasing higher friction in non-circular ducts over circular ducts.

Author's developed equation to determine the mean friction factor is given below;

$$f = 0.358\, Re^{-0.25} \qquad (1)$$

(viii). Comparisons on Velocity Inner-Law of Author's Developed Equation with Others.

Figure 12 shows the comparisons on velocity inner law among the experimental data in developing flow with Sernecki equation and the theoretically established equation of logarithmic law at X/D=57. For making this comparisons velocities are measured along the Y-axis, through the bisector of the bottom heated wall, beginning from Y/B =-0.972 to Y/B=-0.90 at an interval of 0. 1mm. The developed equations which compared with the other two are given below.

Author’s Developed Equation (in Turbulent layer of Developing Flow)

$$\mathbf{U^+ = 4.39 \log Y^+ + 7.82 \quad [40<Y^+<160]} \tag{2}$$

Sernecki’s Equation (in Turbulent layer of Developed flow)

$$\mathbf{U^+ = 5.50 \log Y^+ + 5.45 \quad [30<Y^+<200]} \tag{3}$$

Logarithmic-Law (in Turbulent layer of Developed flow).

$$\mathbf{U^+ = 2.5 \ln Y^+ + 5.50 \quad [Y^+>30]} \tag{4}$$

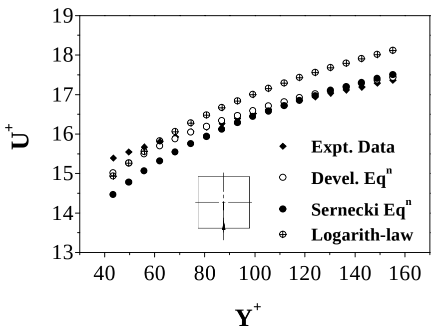

Fig.12 Comparisons on Velocity inner-law of Author's developed Eq. with other's

(ix). Comparisons on TemperatureInner-Law of Author’s Developed Equation with Others.

Figure 13 shows the comparisons on temperature inner law among the experimental data in developing flow with Hirota’s equation and the theoretically established equation of logarithmic law at X/D=57. For making this comparisons temperatues are measured in the same way as measured the velocities mensioned in article(viii). The author’s developed equation which compared with the others two are shown below;

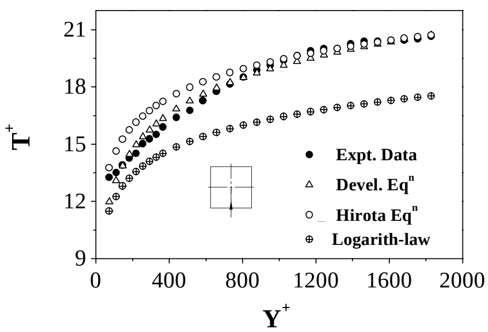

Fig.13 Comparisons on tmperature inner-law of Author's developed Eq. with other's

Author’s Developed Equation (in Turbulent layer of Developing Flow)

$$\mathbf{T^+ = 6.25 \log Y^+ + 0.33 \quad [70<Y^+<1830]} \tag{5}$$

Horota’s Equation (in Turbulent layer of Developed flow)

$$\mathbf{T^+ = 4.98 \log Y^+ + 4.48 \quad [30<Y^+<200]} \tag{6}$$

Logarithmic-Law (in Turbulent layer of Developed flow).

$$\mathbf{T^+ = 4.32 \log Y^+ = 3.44 \quad [30<Y^+ >200]} \tag{7}$$

5. CONCLUSIONS:

The experimental results as mentioned above lead to the following major conclusions;

(i) In the tests, pressure drop occurs between the inlet and outlet of the contraction, varies from 81 N/m^2 to 91N/m^2 for the corresponding Re between 4.6×10^4 to 5.2×10^4. It provides some useful functions, namely; the mean velocity profile is more uniform and flattened, its boundary layer is subjected to a favourable pressure gradient, the relative turbulence level of flow is reduced before entering into the duct and it avoids reasons of adverse pressure gradient.

(ii) The static pressure drop along the length of the test duct increases with an incease of Reynolds number.

(iii) The friction factor decreases with the increase of length of the duct. The rate of decrease is very higher up to X/D=15 and then decreases gradually up to X/D=45. After X/D=45 its rate of decrease is minimum and almost uniform, which is the sign that the flow is becoming developed.

(iv) The correlation of friction law is varying with a linear expression, which is agreed with theoretical approach developed by other researchers for turbulent flow inside ducts.

(v) The friction factor through the length of the test duct decreases with an increase of Reynolds number. Friction facto up to a reasonable limit of pressure drop penalty, should be the main goal in forced convection heat transfer for obtaining higher heat transfer coefficient.

(vi) On the basis of the experimental data, author’s developed equation for the friction law of turbulent flow inside smooth square duct, which will be helpful for the determination of the average friction factor in the developing region.

(vii) On the basis of the experimental data, author’s have developed another two equations for the velocity inner-law and temperature inner-law in turbulent layer of developing flow, which will be almost similar in form as the Hirota’s equation and Logarithmic law.

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NOMENCLATURE:

B	Half of the side of test duct, = D/2, m.
D	Equivalent diameter of the duct, 4×duct cross section area /wetted perimeter, m.

$f_{X/D}$	Local friction factor at each X/D length.
f	Mean friction factor over the length of the test duct.
L	Active length of the developing duct.
ΔP_c	Pressure drop from the inlet of the test duct to individual tap, N/m^2..
ΔP_L	Pressure drop over the length of the test duct, N/m^2.
ρ	Density of air, kg/m^3.
τ_o	Total shear stress at the wall, N/m^2.
U	Velocity of air parallel to the bottom wall surface ,m/s
U^*	Friction velocity, $\sqrt{\tau_o/\rho}$, m/s.
U^+	Universal velocity, U/U^*
Y^+	Dimensionless distance, YU^*/ν
T	Time-mean temperature of air ^oC
T^*	Friction temperature $\ddot{q}/\rho C_p U^*$
T^+	Universal temperature, $(T_{wi} - T)/T^*$
X/D	Length/hydraulic diameter ratio.
X,Y,Z	Three axes of the co-ordinate system.

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