# STUDY OF PRESSURE DROP CHARACTERISTICS AND HEAT TRANSFER PERFORMANCE IN AN INTERNALLY FINNED TUBE

By

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A thesis submitted to the Department of Mechanical Engineering in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering



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# RECOMMENDATION OF THE BOARD OF EXAMINERS

The board of examiners hereby recommends to the Department of Mechanical Engineering, Bangladesh University of Engineering and Technology (BUET), Dhaka, acceptance of the thesis, **"STUDY OF PRESSURE DROP CHARACTERISTICS AND HEAT TRANSFER PERFORMANCE IN AN INTERNALLY FINNED TUBE"**, submitted by **Md. Jalal Uddin**, in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

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# ABSTRACT

Heat exchangers have numerous applications in power plants, industries, automobiles and electrical and electronic equipments. In the present work steady state turbulent flow heat transfer performance of circular tube having integral internal longitudinal fins was experimentally studied.

This report describes new experimental data for turbulent fluid flow and heat transfer in a tube having internal fins. An experimental set up was designed and installed to study the heat transfer performance and pressure drop characteristics in the entrance region as well as in the fully developed region of the finned tube. The tube and fin assembly was cast from brass to avoid contact resistance. Air was used as the working fluid. The Reynolds number based on inside diameter ranged from $1.86 \times 10^4$ to $3.97 \times 10^4$. The heat transfer test section was heated electrically by wrapping nichrome wire providing an uniform heat flux around the tube periphery over the entire length of the test section.

Results exhibited high pressure gradient and high heat transfer coefficients in the entrance region, approaching the fully developed values away from the entrance section. Nusselt numbers of the finned tube were compared with those for an unfinned tube for both constant Reynolds number and constant pumping power. The enhancement of heat transfer rate due to integral fins was found to be very significant over the entire range of the flow rates studied in this experiment. Heat transfer coefficient based on inside diameter and nominal area of finned tube exceeded unfinned tube values by as much as 76.69%.

The results of this study indicate that significant enhancement of heat transfer is possible by using internal fins with sacrificing less additional pumping power. These experimental results are expected to be very useful for the design of pipe-lines and heat exchanger tubes.

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# DECLARATION

No portion of the work contained in this thesis has been submitted in support of an application for another degree or qualification of this or any other University or Institution of learning.

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# CONTENTS

# NOMENCLATURE

| | |
|---|---|
| $A_x$ | Cross sectional area of tube having no fins ($m^2$). |
| $A_{xf}$ | Flow cross sectional area of finned tube ($m^2$). |
| $A_h$ | Effective heat transfer area (m). |
| $A$ | Nominal (heat transfer) area (m). |
| $D_i$ | Inside diameter (m). |
| $D_h$ | Hydraulic diameter (m). |
| $N$ | Number of fins. |
| $H$ | Height of the fin (mm). |
| $W$ | Width of the fin (mm). |
| $R_3$ | Constant pumping power criteria |
| $R_e$ | Reynolds number. |
| $N_u$ | Nusselt number. |
| $h_o$ | Heat transfer co-efficient of smooth tube ($w/m^2\,^oC$). |
| $h$ | Heat transfer co-efficient of finned tube ($w/m^2\,^oC$). |
| $T$ | Temperature ($^oC$). |
| $M$ | Mass flow rate (kg/s). |
| $Q$ | Total heat input to the air (w). |
| $Q^/$ | Heat input to the air per unit axial length (w/m). |
| $x$ | Axial distance (m). |
| $C_p$ | Specific heat of air ($j/kg^oC$). |

| | |
|---|---|
| $P$ | Pressure ($N/m^2$). |
| $V_i$ | Mean velocity in inlet section (m/s). |
| $V$ | Mean velocity in finned tube (m/s). |
| $V_c$ | Velocity at the axial of the tube (m/s). |
| $\rho$ | Density of air ($kg/m^3$). |
| $\mu$ | co-efficient of viscosity of air (Pa-s). |
| $\nu$ | Kinematic viscosity ($m^2/s$) |
| $k$ | Thermal conductivity of air ($w/m^oC$). |
| $F$ | Friction factor. |
| $L_t$ | Thermal entrance length (m). |

**Subscripts**

| | |
|---|---|
| i | Based on inside diameter. |
| h | Based on hydraulic diameter. |
| o | Smooth tube i.e. unfinned tube. |
| f | Finned tube. |
| w | Wall temperature. |
| b | Bulk temperature. |
| x | Axial distance. |
| p | Constant pumping power. |

# LIST OF FIGURES

# LIST OF GRAPHS

# LIST OF FIGURES

# CHAPTER - I

# INTRODUCTION

Fins have extensive applications power plants, chemical process industries, and electrical and electronic equipment for augmentation of heat transfer. Enhancement techniques that improve the overall heat transfer characteristics of turbulent flow in tubes are important to heat exchanger designers. Efficient design of heat exchanger equipment with fins can improve system performance considerably. Among several available techniques for augmentation of heat transfer in heat exchanger tubes, the use of internal fin appears to be very promising method as evident from the result of the past investigations.

Numerous studies have been made both experimentally and analytically with tubes having internal fins in laminar and turbulent flow. In case of analytical investigations, the analysis for laminar and turbulent flow is based on the differential equations for momentum and energy conservation in the flowing fluid. The solutions were obtained numerically integrating the partial differential equations with required boundary conditions. From the experimental and numerical study, it is evident that for both laminar and turbulent flow regimes, the finned tube exhibited substantially higher heat transfer coefficient when compared with smooth (unfinnd) tubes.

**OBJECTIVES**

Since finned surfaces have extensive applications in power plants, chemical process industries etc., a research work has been undertaken to study the heat transfer performance in the developing region of turbulent flow of air in a circular tube having internal longitudinal fins.

**The study has the following objectives:**

i. To study the effect of longitudinal internal fins in a circular tube on friction factor, pressure drop and temperature variation in the entrance region as well as fully developed region.

ii. To study the effect of internal longitudinal fins in a circular tube on heat transfer characteristics in the entrance region as well as in the fully developed region.

iii. To investigate the abrupt pressure fluctuation as obtained by Mafiz [12-13] near the entrance region of the tube using inclined ($30^o$) U-tube manometers.

# CHAPTER - II

## LITERATURE SURVEY

Enhancement techniques that improve the overall heat transfer characteristics of laminar and turbulent flow in tubes are important to heat exchanger designers. It is evident that for both laminar and turbulent flow regimes, the finned tubes exhibit substantially higher heat transfer coefficients when compared with corresponding smooth (unfinned) tubes, and significant savings can be made from employing finned tubes.

Hu and Chang [10] analyzed fully developed laminar flow in internally finned tubes by assuming constant and uniform heat flux in tube and surfaces. By using 22 fins extended to about 80% of the tube radius, they showed an enhancement of heat transfer as high as 20 times that of unfinned tube could be realized. Soliman et al. [18] also analyzed fully developed laminar flow, but accounting for conduction in the tube wall and fins keeping the outer surface of the tube at a constant temperature. In a later study, Prakash and Patankar [17] investigated the influence of buoyancy on heat transfer in vertical internally-finned tube under fully developed laminar flow condition. Analyses of thermally-developed laminar flow in internally finned tubes were carried out by Prakash and Liu [16] and Rustum and Soliman [18] by numerically solving the governing equations. They concluded that internal fin influences the thermal development in a complicated way, which makes it inappropriate to extend the smooth tube results to internally finned tubes on hydraulic diameter basis. Patankar et al. [15] analyzed the fully developed turbulent flow and heat transfer characteristics for tubes and annuli with longitudinal internal fins using a mixing length model. The local heat transfer coefficients exhibited a substantial variation along the fin heights with smallest value at the base and largest value at the tip.

Carnavos [2] experimentally determined the heat transfer performance for cooling air in turbulent flow in tubes having integral internal spiral and straight longitudinal fins. He conducted experiments with 21 tubes and found that these tubes were potentially capable of increasing the capacity of an existing heat transfer at constant pumping power by 12% to 66% by direct substitution of an intrnal fin tube for a smooth tube. In a later study, Carnavos [3] investigated experimentally the heat transfer performance for heating water in turbulent flow for 11 tubes having integral internal spiral and longitudinal fins. He also performed test for a longitudinal and spiral fins tube within the 11 tubes and a smooth tube with a 50% W/w ethylene glycol-water solution. From these strength he obtained correlation equation for heat transfer and Fanning friction factor that describe the air, water and ethylene glycol-water data within ±10%. Edwards et al. [6] performed an experimental investigation of fully developed, steady and turbulent flow in a longitudinally finned tube. They used a two-channel, four beam, laser-doppler velocimeter to measure velocity profiles and turbulent statistics of air flow seeded with titanium dioxide particles. They compared friction factor with different Reynolds numbers to literature results and showed good agreement for both smooth and finned tubes. Masliyah and Nanda Kumar [14] analyzed heat transfer characteristics for a laminar forced convection fully developed flow in a circular tube having internal fins of circular shape with axially uniform heat flux with peripherally uniform temperature using a finite element method. They obtained for a given fin geometry that the Nusselt number based on inside diameter was higher than that of a smooth tube. They also found that for maximum heat transfer there exists an optimum fin number for a given fin configuration. Zhang and Ebadian [22] investigated numerically the fluid flow and heat transfer of steady laminar forced convection in the entrance region of semicircular ducts with longitudinal internal fin subjected to a constant wall temperature. They concluded that increasing the number of fins and relative fin height, Nusselt number increases according, and thus delays the development of the local Nusselt number in the entrance region of the duct. Mafiz et al. [12-13] studied experimentally steady state turbulent flow heat transfer performance of circular tubes having six integral internal longitudinal fins and in his work Mafiz found an abrupt pressure fluctuation near the entrance region of the tube. Yet the results of this study indicates that significant enhancement of heat transfer is possible by using internal fins without scarifying additional pumping power. Experimental data for pressure drop and heat transfer coefficient in internally finned tubes has been reported, among others, by Hiding and Coogan [8], Bergles et al. [1] Watkinson [21], and Rustum and Soliman (19). Bulk of these data are characterized the laminar flow regimes using air, water and oil as working fluids.

# CHAPTER - III

# HEAT TRANSFER PARAMETERS

In this chapter the basic definition of heat transfer parameters and thermal boundary conditions in connection with the present have been discussed.

## 3.1 THERMAL BOUNDARY CONDITIONS

In the study of heat transfer performance in tubes two types of thermal boundary condition may be considered:

i. Uniform heat input per unit axial length.

ii. Uniform temperature both axially and peripherally. In the present study, this case is considered.

## 3.2 DEFINITION OF PARAMETERS

**Hydraulic Diameter:**

For internal flows the hydraulic diameter $D_h$ is often used as characteristic length. It is defined as

$$D_h = \frac{4x\,(Cross\,sectional\,area\,of\,the\,flow)}{Wetted\;\;perimeter}$$

$$= \frac{4\pi D^2{}_i / 4}{\pi D_i + 2NH} \tag{3.1}$$

(Assuming thin fin)

In the present study, fins can not be assumed as thin and hydraulic diameter $D_h$ was obtained as follows:

$$D_h = \frac{4\,A_{xf}}{\pi D_i + 2NH} \qquad (3.2)$$

where $A_{xf} = \left(\frac{\pi D_i^2}{4} - NWH\right)$

**Reynolds Number:**

Reynolds number based on inside diameter is defined as

$$Re_i = \frac{\rho V D_i}{\mu} \qquad (3.3)$$

Reynolds number based on hydraulic diameter is defined as

$$Re_h = \frac{\rho V D_h}{\mu} \qquad (3.4)$$

**Pressure Drop and Fanning Friction Factor:**

The dimensionless pressure drop at any axial location x, is given by the following equation,

$$P^*(x) = -\,\Delta P(x)/1/2\rho V^2 \qquad (3.5)$$

Where $\Delta P = P_i - P(x)$

Pi = Pressure at inlet

P(x) = Pressure at any axial location, x

The local friction factor based on hydraulic diameter is given by

$$F_h = \frac{(-\Delta P / x) D_h}{2\rho V^2} \qquad (3.6)$$

The local friction factor based on inside diameter is given by

$$F_i = \frac{(-\Delta P / x) Di}{2\rho V^2} \qquad (3.7)$$

**Heat Transfer: Uniform Heat Input per Unit Axial Length:**

The local bulk temperature of the fluid $T_b(x)$ can be defined as by the following heat balance equation

$$T_b(x) = T_i + \frac{Q^{/}x}{MC_p} \tag{3.8}$$

The local heat transfer coefficient at any axial location (for both tube and fin) can be defined as

$$h_x = \frac{Q^{/}}{(T_w - T_b)_x A_h} \tag{3.9}$$

where, $A_h = (\pi D_i + 2NH)$ (3.10)

The local Nusselt number can be defined as

$$Nu_x = \frac{h_c\, D_h}{k}$$

$$= \frac{Q'/(\pi D_i + 2NH)}{(T_w - T_b)_x} \frac{D_h}{k} \tag{3.11}$$

## 3.3 THERMAL ENTRANCE LENGTH

The thermal entrance length, $L_t$ is defined as the length required for the local Nusselt Number to equal 1.05 times its fully developed value. The results at the entrance region can be conveyed via the temperature ratio,

$$\frac{(T_w - T_b)_{fd}}{(T_w - T_b)_x}$$

The denominator is the wall to bulk temperature difference at any axial distance x, while numerator is the axially unchanging wall to bulk temperature difference in the fully developed regime. Representative axial distributions of the temperature ratio may be plotted as a function of the dimensionless axial coordinates (X/L). The entrance length $L_t$ may be defined as the length required for

$$\frac{(T_w - T_b)_{fd}}{(T_w - T_b)_x} = 1.05$$

## 3.4 HEAT TRANSFER IN FULLY DEVELOPED REGION

In uniform heat input per unit axial length condition the wall temperature and the fluid bulk temperature increases with axial distance of the test section up to certain length. There will be a portion of the test section where the wall and fluid bulk temperature are parallel, yielding a uniform value of ($T_w$ - $T_b$) in the fully developed regime.

The fully developed heat transfer coefficient h can be defined as

$$h = \frac{Q'}{A_h(T_w - T_b)_{fd}} \tag{3.12}$$

where $A_h = (\pi D_i + 2NH)$ (3.13)

$$Q' = MC_p\,(T_i - T_o)/L \tag{3.14}$$

Nusselt number for thermally developed regime is given

$$Nu = \frac{h \;\; D_h}{k} \tag{3.15}$$

## 3.5 HEAT TRANSFER COEFFICIENT BASED ON INSIDE DIAMETER AND NOMINAL AREA

The heat transfer data are presented both on the basis of the nominal ($A = \pi Di$) area, A of unfinned tube and of effective (heat transfer) area $A_h$ of finned tube. the heat transfer coefficient, $h_i$ based on inside diameter and nominal area is then related to the coefficient, h based on hydraulic diameter and effective area by the following equation given below:

$$Q' = A\, h_i\, \Delta T = A_h\, h\, \Delta T$$

Therefore, $h_i = \frac{h \;\; A_h}{A}$

$$= \frac{h(\pi D_i + 2NH)}{\pi D_i} \tag{3.16}$$

Thus, when heat transfer coefficient for the tubes are reported on a nominal area basis, the coefficient includes the effect of both the finned and unfinned tube. This is a useful means of expressing heat transfer performance, as it allows a direct measure of the results if a smooth tube is replaced by a finned tube.

## 3.6 CRITERIA OF EVALUATING HEAT TRANSFER PERFORMANCE

There are minor geometric differences in tube configuration among various experiments that affect specific performance. However, to minimize the influence of this and other variables on a direct comparison, the basis chosen is the constant pumping power criterion, $R_3$ which is really a most important parameter.

**Criterion 3:**

Criterion $R_3$ aims at the increase of heat transfer the case of constant pumping power and constant geometric the tube (exchanger). $R_3$ can be evaluated as

$$R_3 = \frac{h_{if} \ at \ Re_{if}}{h_{op} \ at \ Re_o} \tag{3.17}$$

Pumping power can be defined as

$$P_m = (-\Delta P/\rho)\, M$$

$$= \frac{4\, F_i L}{D_i} \frac{V^2}{2} A_x\, V\rho \tag{3.18}$$

For equal (or constant) pumping power in smooth and finned tubes,

$$P_{mo} = P_{mf}$$

Therefore, $$A_{xo} \ F_{io} \ V_o^3 = A_{xf} \ F_{if} \ V_f^3 \tag{3.19}$$

$$\left(\frac{V_o}{V_f}\right)^3 = \frac{A_{xf} \ F_{if}}{A_{xo} \ F_{io}} \tag{3.20}$$

Reynolds number based on inside diameter for smooth tube is

$$Re_o = \frac{\rho V_o \quad D_i}{\mu} \tag{3.21}$$

Reynolds number based on inside diameter for smooth tube is

$$Re_{if} = \frac{\rho V_f \quad D_i}{\mu}$$

Therefore, $$\frac{Re_o}{Re_{if}} = \frac{V_o}{V_f} \tag{3.22}$$

From equation (3.21) and (3.22), we can write

$$A_{xo} \quad F_{io} \quad Re_o^3 = A_{xf} \quad F_{if} \quad Re_{if}^3 \tag{3.23}$$

The relation of Blasius, valid for 3000 < Ra < $10^5$, is used to eliminate the friction factor $F_{io}$, in the expression (3.23).

$$F_{io} = 0.079 \qquad (Re_o)^{-0.25} \tag{3.24}$$

The expression (3.23) becomes,

$$A_{xo}(0.079\ Re_o^{-0.25})Re_o^3 = A_{xf} \quad F_{if} \quad Re_{if}^3$$

Therefore,

$$Re_o = 2.517\ (Re_{if})^{12/11}(A_{xf}\, F_{if}/A_{xo})^{4/11} \tag{3.25}$$

The equivalent smooth tube Reynolds number can be evaluated from the equation (3.25). For constant pumping power comparison experiments will be carried out with fin tube at Reynolds number $Re_{if}$ and with smooth tube at Reynolds number $Re_o$, obtained from the equation (3.25). It is then possible to calculate constant pumping power criteria, $R_3$ from the equation,

$$R_3 = \frac{h_{if} \quad at \quad Re_{if}}{h_{op} \quad at \quad Re_o}$$

# CHAPTER - IV

# EXPERIMENTAL SET UP AND PROCEDURE

An experimental set up was designed, fabricated and installed to study the friction factor and heat transfer performance of circular tube having internal integral longitudinal fins. The schematic diagram is shown in the figure (A1 and A2). Air was used as the working fluid. Air in the test section was supplied by a centrifugal fan fitted at the end of the set up. It was driven by a motor [3 phase, 3hp, 420V, 4.1A]. The maximum flow rate under the free operation was approximately 30 $m^3$/min. A butterfly type valve was used to control the flow rate of air during the experiment in the test section and it was installed after the test section. The set up consisted of three sections:

I. inlet and flow measuring section

II. heat transfer test section and

III. fan assembly.

## 4.1 INLET SECTION

The unheated inlet section (shaped inlet) cast from aluminum and heat transfer test section cast from brass are of same diameter. The open end of the pipe would probably act to some extent as a sharp edged orifice, and the air flow would contract and not fill the pipe completely for a short distance from the end. This effect was avoided here in the experiment by fitting a specially made shaped inlet. The pipe shaped inlet, 530 mm long, (Fig. A3B) was made integral with the test section so as to avoid any flow disturbances at upstream of the test section for flow measurement. The coordinates of the curvature of the shaped inlet was suggested by Owner and Pankhurst (7) which is represented in table (C2), Appendix-C.

## 4.2 TEST SECTION

The test section contains eight internal integral longitudinal fins (1520 mm long and 70 mm inside diameter) with circular tube to avoid contact resistance. The fin and tube assembly was cast from brass, because of its high thermal conductivity and easy machinability. Two halves of circular tube with integral fins were cast separately and joined together to give the shape of a circular tube. Fig. (A3A) depicts half the tube configuration and dimension. As two halves were joined at the line of symmetry, it did not effect thermal and hydrodynamic results of the experiment. The test section was wrapped with mica sheet, glass fiber taped and insulation tape. Over mica sheet nichrome wire (of resistance 0.610 ohm/m) was spirally wound uniformly with spacing of 8 mm between the fins. The nichrome wire was covered with mica sheet, glass fiber tape and insulation tape to make it electrically insulated by covering with asbestos. The test section was placed in the test rig with the help of the bolted flanges, between which asbestos sheets were installed which act as heat guards in the longitudinal direction.

The test section electric heater was supplied power by 5 KVA variable voltage transformer connected to 220 VAC power through a magnetic contactor and temperature controller. The temperature controller was fitted to sense the air outlet temperature and give signal to heater for switching it off or on automatically. It protects the experimental set up from being excessive heated which may happen at the time of experiment when the heating system is in operation continuous for hours to bring the system in steady state condition. It also controls the air outlet temperature.

Fig. (A9) shows the electrical circuit diagram of the heating system of the test section. The particulars of the electric heater, temperature controller, fan and data acquisition system are given in table (C1) in Appendix-C.

## 4.3 FAN ASSEMBLY

A diffuser of cone angle 12° was made of 1.5875 mm M.S plate and fitted to the uction side of the fan. The diffuser was used for minimizing head loss at the suction side. In order to prevent the transmission of any vibration from the fan to the test section a flexible duct was installed between the inlet section of the fan and 76.20 mm diameter pipe of the set up as shown in the Fig. (A2). A butterfly type valve was used to control air flow rate during experiment at the suction side before the flexible duct.

## 4.4 FLOW MEASUREMENT BY TRAVERSING PITOT

Flow of air through the experimental set up was measured at inlet section with the help of a traversing pitot. A specially shaped inlet made of aluminum was installed at the inlet to the test section to have an easy entry and symmetrical flow. According to Owner and Pankhurst (7), from the inlet a traversing pitot was installed at a distance of 4 pipe diameters from the inlet. A drawing of the Micrometer traversing pitot is shown in Fig. (A6).

To find out the location of pitot tube for measurement of velocity head, the tube diameter was divided into five equal concentric areas and then their center were found out by using the following calculation. And the locations for the traversing pitot is shown in the Fig. A10.

**Arithmetic Mean Method:**

$$\frac{1}{5}\pi R^2 = \pi r_1^2$$

$$r_1 = 0.447R$$

$$\frac{1}{5}\pi R^2 = \pi r_2^2 - \pi r_1^2$$

$$\Rightarrow \frac{1}{5}R^2 + (0.447R)^2 = r_2^2$$

$$\therefore r_2 = 0.632R$$

Similarly,

$$r_3 = 0.774R$$

$$r_4 = 0.894R$$

$$r_5 = 1R$$

Now

$$\pi r_{5-4}'^2 = \frac{\pi R^2 + \pi(0.894R)^2}{2}$$

$$\therefore r_{5-4}' = 0.948R$$

Again $\pi r'^2 = \dfrac{\pi(0.894R)^2 + \pi(0.774R)^2}{2}$

$\therefore r'_{4-3} = 0.836R$

Similarly

$r'_{3-4} = 0.7065R$

$r'_{2-1} = 0.547R$

$r'_{1-0} = 0.316R$

Mean velocity was measured by traversing pitot tube along the diameter of the pipe at ten measuring points.

## 4.5 MEASUREMENT OF STATIC PRESSURE

The static pressure tappings were made at the inlet and outlet of the test section as well as equally spaced 7 axial location of the test section as shown in Fig. A5. Pipe wall pressure tappings (Fig. A8) for measurement of static pressure were made of brass and installed carefully such that they just flush inside the surface of test section. The outside parts of these were made tappered to ensure an air tight fitting into the plastic tubes which were connected to the manometer. Epoxy glue (Araldite) was used for proper fixing of the static pressure tappings. U-tube manometers at an inclination 30° were attached with the pressure tapping. The manometric fluid used in the U-tube manometer was water.

## 4.6 MEASUREMENT OF TEMPERATURE

The temperatures at the different axial locations of the test section were measured with the help of K-type thermocouples along with data acquisition system and computer at five minutes interval. The temperature measuring locations are-

i. Fluid bulk temperature at the outlet of the test section.

ii. Wall temperature at 8 axial locations of the test section.

iii. Fin-tip temperature at 8 axial locations of the test section.

The bulk temperature of the air at the outlet of the test section was measured using a thermocouple at the outlet of the test section. 16 thermocouples were installed in 8 cross sections (Fig. A4) with two in each cross section to measure the wall as well as the fin-tip temperatures of the test section. All the thermocouples and the data acquisition systems were calibrated before their use. Thermal contact between the brass tube and the thermocouple junction was assured by peening thermocouples junction into grooves in the wall. 1.5875 mm holes were drilled across the height of the fin in 8 cross sections to measure the temperature of the fin-tip. Thermocouples were inserted into the holes and peened into the grooves of the fin-tip.

## 4.7 PROCEDURE OF EXPERIMENT

The fan was first switched and allowed it to run for a few minutes so that the transient characteristic died out. The flow of air through the test section was varied and kept constant with the help of a flow control valve. Then electrical heating circuit was switched on.

All the pressure tapping for measuring static pressure were connected with inclined ($30^{O}$) U-tube manometers. The manometric fluid was water. The readings for the velocity head were taken by traversing the pitot along the diameter of the pipe. Inclined tube manometer was used for measuring velocity head and the fluid in the manometer was high speed diesel with sp. gr. 0.855.

The electrical current was adjusted with the help of a Regulating Transformer (or Variac) to attain steady state condition for a particular Reynolds number. Steady state condition for temperature at the different locations of the test section was defined according to D.L. Gee and R.L. Webb (5) by two measurements. First the variation in wall thermocouples was observed until constant values were attained; then the outlet air temperature was monitored. Steady state condition was attained when the outlet air temperature did not deviate over a period of 10-15 minutes. All the thermocouples readings were taken at steady state condition.

After one run of the experiment at a particular Reynolds number, the Reynolds number was changed with the help of the flow control valve keeping electrical power input constant. All the thermocouples readings and static pressure readings were taken at every tapping along the axial direction for each run of the experiment.

# CHAPTER - V

# RESULTS AND DISCUSSION

In this experimental work fluid flow and heat transfer performance were studied. Pressure drop characteristics and heat transfer performance were calculated in the entrance region as well as in the fully developed region of the finned tube.

The dimensionless pressure drop P* along the length of the tube is shown in the figure B1. The pressure gradient is high in the entrance region, then pressure recovers and finally approaches the fully developed values away from the entrance section. Friction factor was calculated from the pressure measured from 60 mm down stream of the inlet section. Figure B4 shows the friction factor, Fi based on inside diameter of the tube. It can be noted that friction factor for a finned tube is much higher than that of a unfinned tube, but the slope of the friction factor curve of the finned tube is nearly equal to the unfinned tube. Friction factor based on inside diameter of a finned tube increases by 489.825% to 622% over that of a smooth tube for Reynolds number range from $1.86 \times 10^4$ to $3.97 \times 10^4$. Friction factor based on hydraulic diameter was also calculated. It was found that friction factor based on hydraulic diameter of a finned tube increase over that of a smooth tube by 107.50% to 153.83% for Reynolds number based on hydraulic diameter range from $8.1 \times 10^3$ to $1.72 \times 10^4$. The friction factor based on hydraulic diameter is shown in the figure B2. Friction factor along the length of the finned tube is shown in the fig. B3 at two different Reynolds number based on inside diameter. It can be noticed that the friction factor is high near the entrance region due to large pressure drop then falls gradually to fully developed value.

Figure B5 shows the wall and bulk temperature distribution along the length of the finned tube for different two Reynolds number based on inside diameter. The wall temperature was found by direct measurement as described earlier and the bulk temperature was determined by energy balance equation (equation 3.8). At higher Reynolds number the wall temperature is low because more heat is taken away by the air. The figure shows that there is a portion of the test section where the wall and the bulk temperature distribution are parallel , yielding a uniform value of ($T_w$ - $T_b$) as expected from constant heat rate. Just up-stream of the exit, the slope of the wall temperature gradually falls due to end effect.

Figure B6 shows the variation of local Nusselt number along the length of the finned tube. Nusselt number is large in the entrance region due to the development of thermal boundary layer with the entrance section as the leading edge. It decreases with increasing the axial distance approaching the fully developed values and then it increases with the axial distance which is very unusual occurence and needs furthur investigation. The figure shows that at higher Reynolds number, the curve of the local Nusselt number along the length of the finned tube is higher. This is expected because a higher flow rate results in increment of heat transfer coefficient at both tube surface and the surface of the fins.

Figure B7 shows the ratio of the local Nusselt number and the Nusselt number at the fully developed condition for the same reynolds number. It can be noticed that a larger enhancement of heat transfer near the entrance region is realized when the Reynolds number is larger.

Figure B12 shows the heat transfer results of finned tube based on inside diameter and nominal area as a function of Reynolds number based on inside diameter and compares with that of the smooth tube. The heat transfer curve for finned tube is much above the smooth tube curve showing much enhancement of heat transfer due to fins. It also can be noticed that the slope of the heat transfer curve for finned tube and that of the smooth are almost same. Heat transfer for the finned tube exceeded smooth tube values by 51.17% to 76.69% for reynolds number range from $1.86 \times 10^4$ to $3.97 \times 10^4$ based on inside diameter.

Figure B13 plots the variation of Nusselt number at fully developed flow with different Reynolds number based on inside diameter. The fully developed Nusselt number increases with the increase of the Reynolds number.

Figure B11 plots the constant pumping power ratio, R3. It can be noticed that R3 varies over a range of 0.875 to 1.0453 for Reynolds number, $Re_o$ from $3.60 \times 10^4$ to $7.86 \times 10^4$. This clearly demonstrates the advantage of a finned tube over a unfinned tube for heat exchanger design.

Figure B10 shows the ratio of $(T_w - T_b)_{FD} / (T_w - T_b)_x$ along the axial length of the finned tube for different Reynolds number based on inside diameter. The $(T_w - T_b)_{FD} / (T_w - T_b)_x$ ratio is large in the entrance region, then it falls gradually and reaches to the fully developed value. Just up-stream of the exit, the curves again rise due to end effect.

## CHAPTER - VI

# CONCLUSIONS AND RECOMMENDATIONS

# 6.1 CONCLUSIONS

Steady state fluid flow and heat transfer performance of a circular tube having internal integral longitudinal fins were studied experimentally. The experimental study has revealed that heat transfer coefficient of finned tube is large in the entrance region and the enhancement of heat transfer in the fully developed region is remarkable due to fin effects. The findings of the present study are summarized below:

1. Friction factor of finned tube based on inside diameter is in the range of 5.89 to 7.21 times the smooth tube values. But the slope of the friction factor of finned tube is very nearly equal to that of the smooth tube.

2. The local friction factor is high near the inlet, and drops gradually to the fully developed value.

3. Nusselt number is high in the entrance region and it decreases with increasing the axial distance approaching asymptotically the fully developed value.

4. In the fully developed region, the slope of $N_{ui}/Pr^{0.4}$ vs $Re_i$ curve of finned tube and smooth is almost similar.

5. Based on inside diameter and nominal area, heat transfer for the finned tube exceeded smooth tube values by 51.17% to 76.69% for Reynolds number range from $1.86 \times 10^4$ to $3.97 \times 10^4$.

6. The heat transfer coefficient based on inside diameter and nominal area were in the range of 1.51 to 1.76 times the smooth tube values.

7. When compared with a smooth tube at constant pumping power and constant basic geometry of the tube an improvement as high as only 4% was obtained.

## 6.2 RECOMMENDATIONS

i. With some modifications of the experimental set up, thetest section can be replaced by anther one of varying tube diameter, number of fins, fin height,length of the test section etc.

ii. All the experiments can be done in Reynolds number keeping pace with previous work so that the exact comparison can be performed with the work available in literature. This may leads to the use of blowers of different specifications.

iii. Pressure transducers can replace the inclined U-tube manometers for measurement of pressure drop and friction factor along the length of the tube.

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# APPENDIX - A

# FIGURES

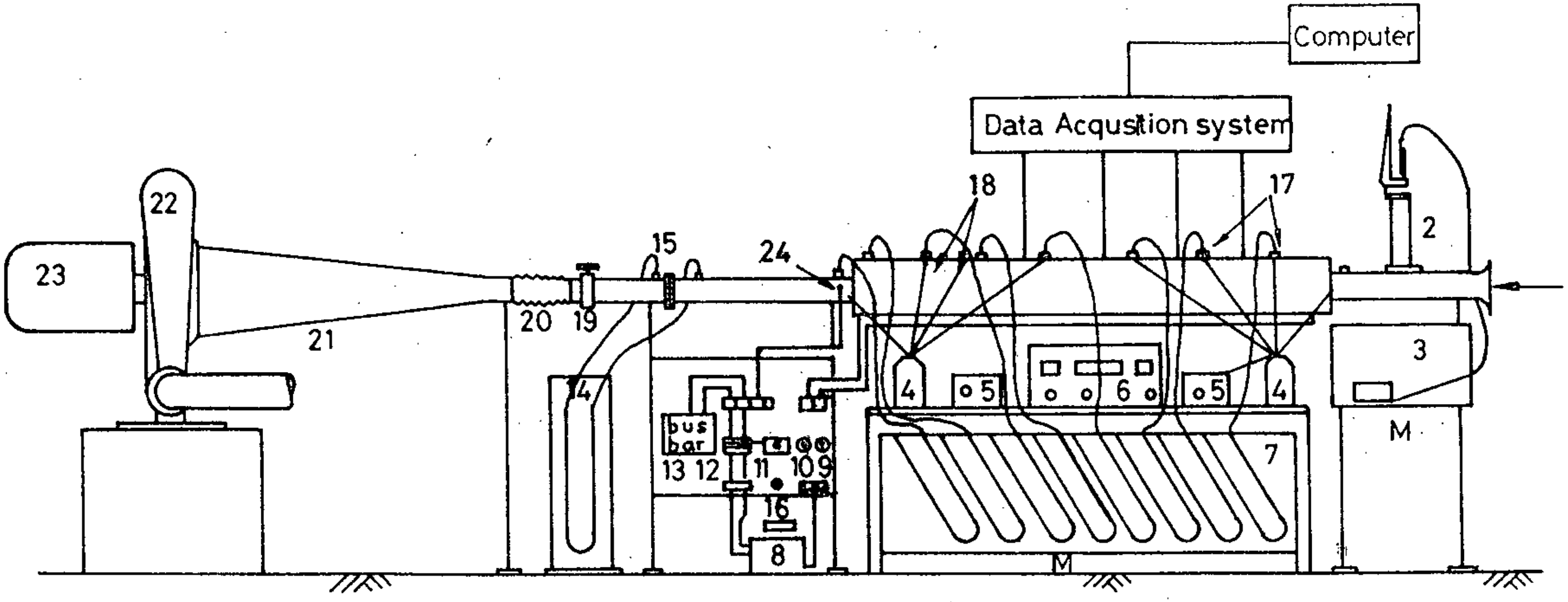
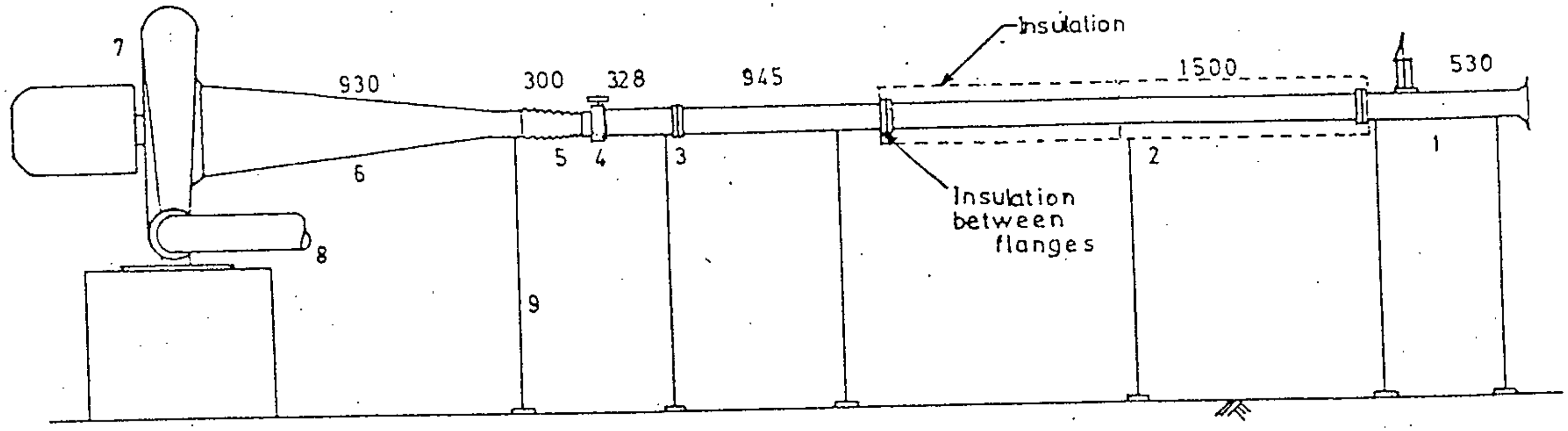



FIG. A 1 SCHEMATIC DIAGRAM OF EXPERIMENTAL SET UP




| | |
|---|---|
| 1 | Shaped inlet |
| 2 | Test section |
| 3 | Orifice meter |
| 4 | Butterfly valve |
| 5 | Flexible duct |
| 6 | Diffuser |
| 7 | Pan |
| 8 | Outlet |
| 9 | Supports |

All dimensions in mm.

FIG. A 2 SCHEMATIC DIAGRAM OF EXPERIMENTAL SET UP

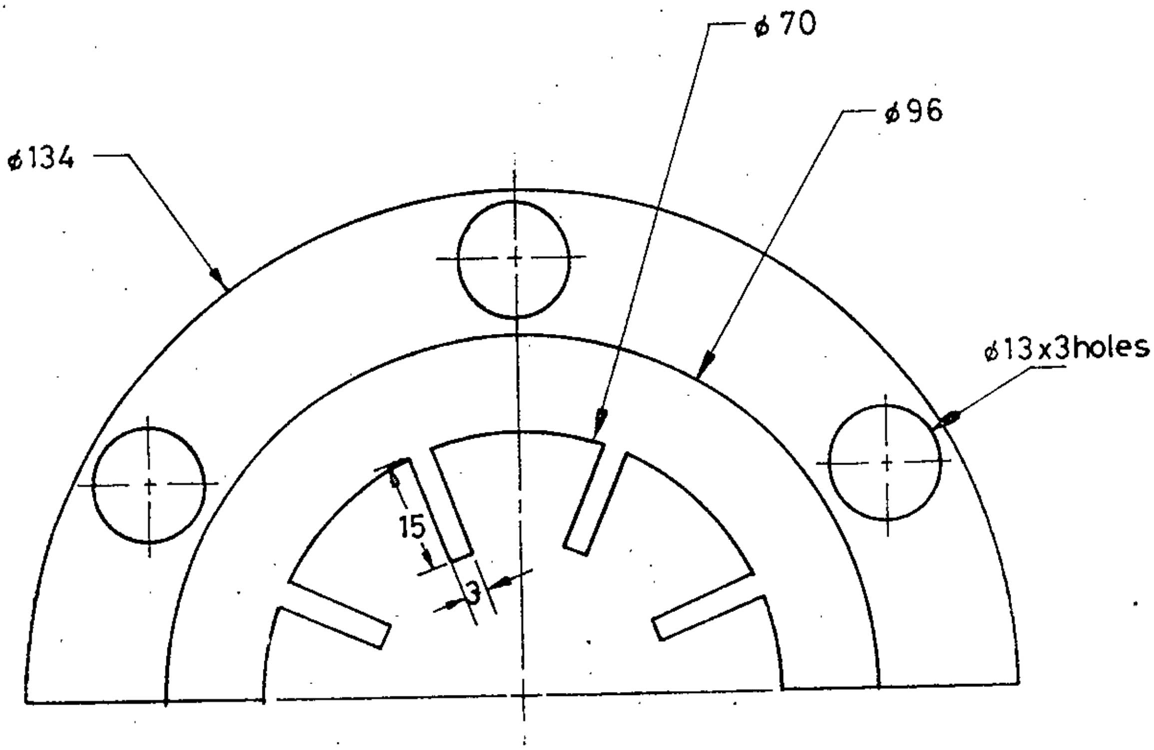



FIG. 13A HALF OF THE CIRCULAR TUBE WITH FINS

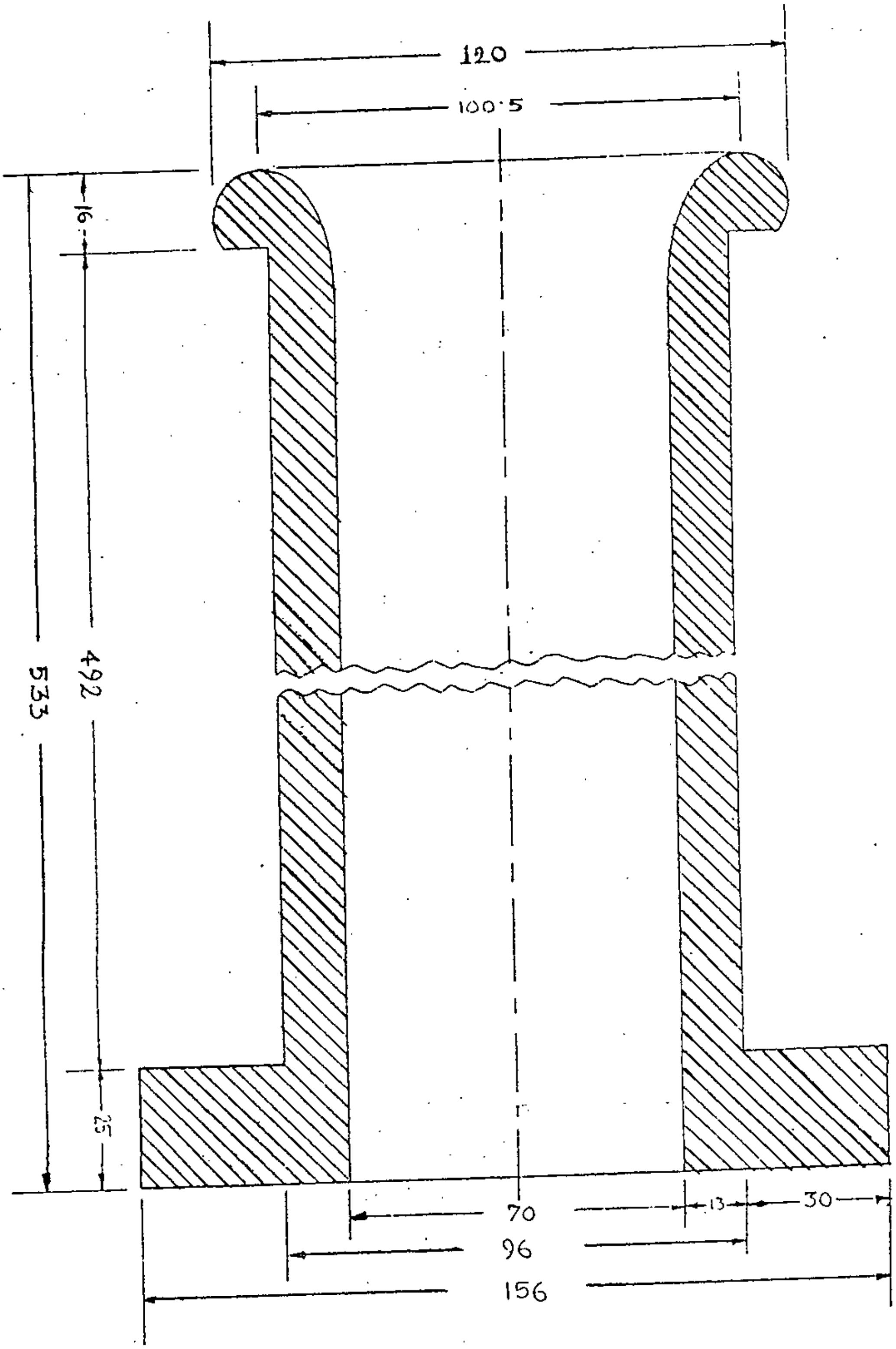
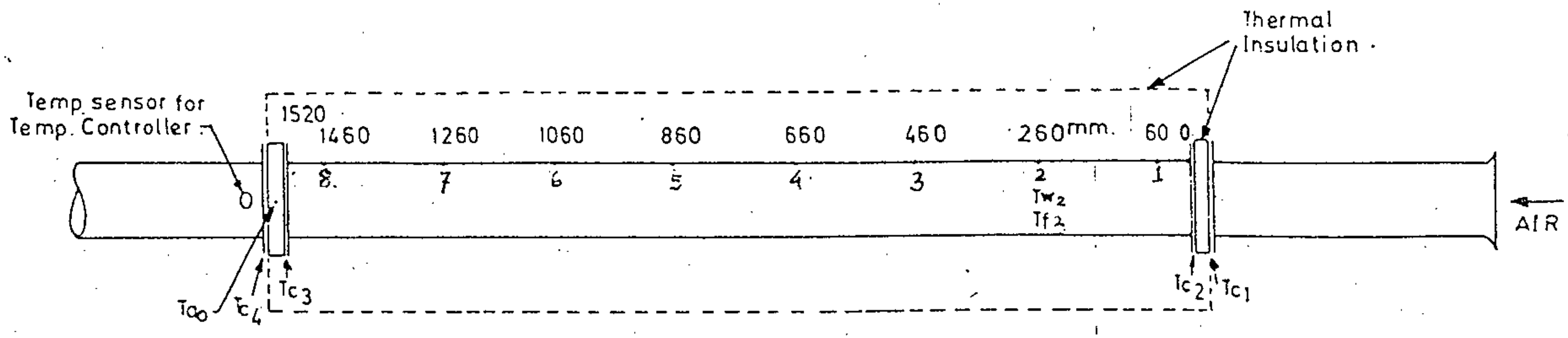



FIG. A 3B SHAPED INLET




W ~ Wall
f ~ Fin
i ~ Inlet
o ~ Outlet
a ~ Air
c ~ Coupling/Flange

Tao ~ Air outlet temp
Tc ~ Temp at the flange
$T_{w2}$ ~ Wall temp at 2
$T_{f2}$ ~ Fin-tip temp at 2.

FIG: A4 LOCATION OF THERMOCOUPLES

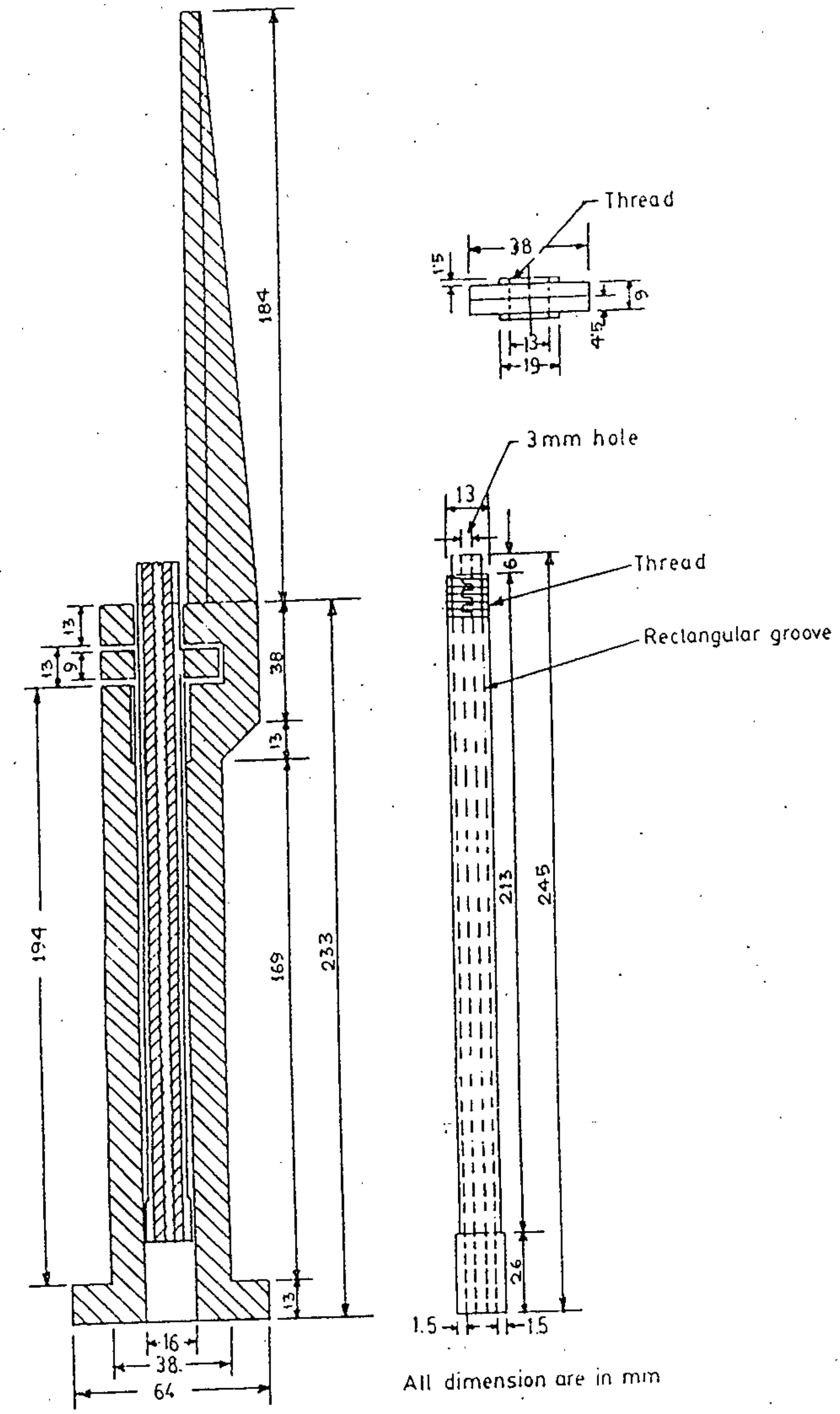



FIG. A6 TRAVERSING PITOT HOLDING DEVICE

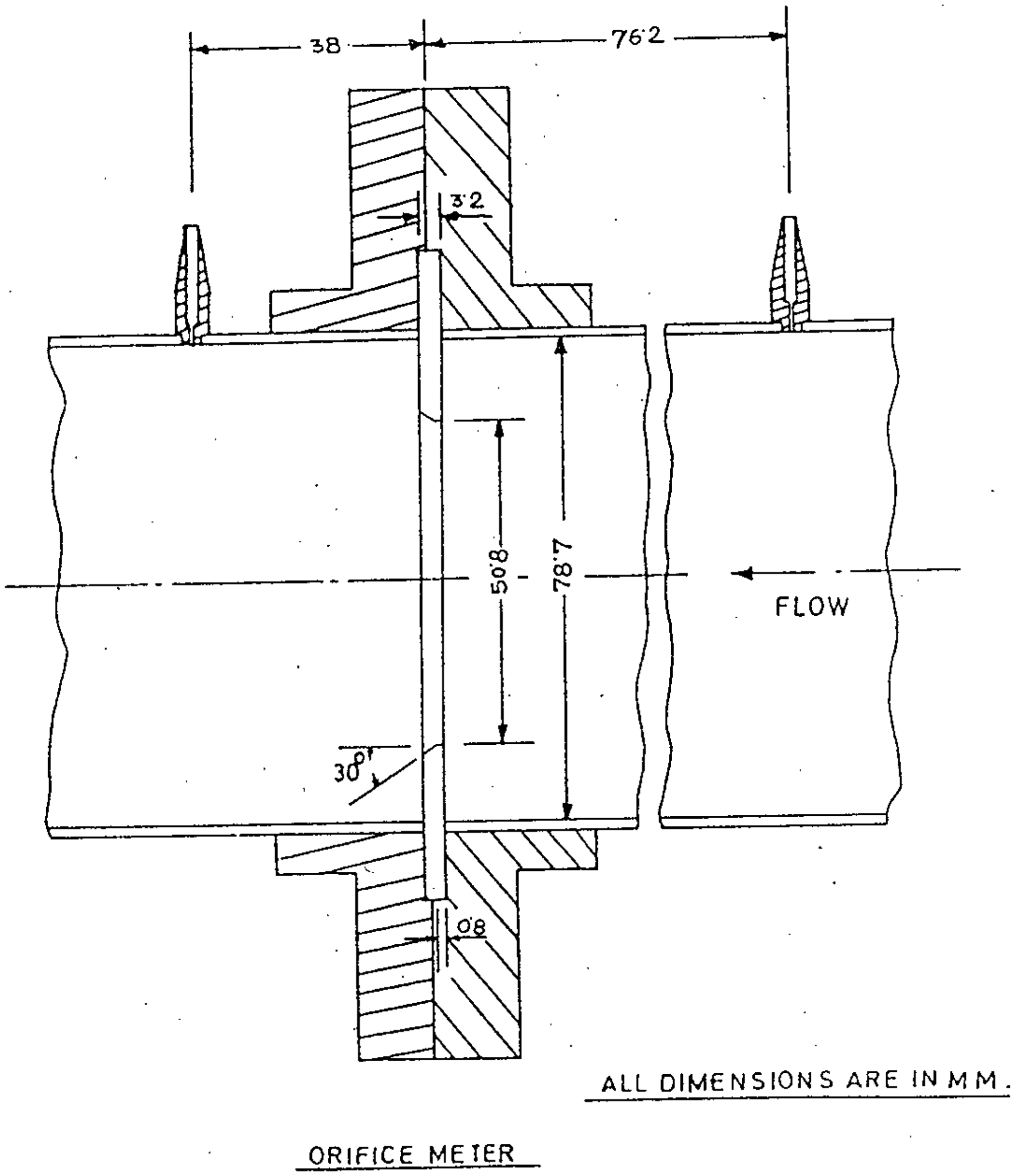
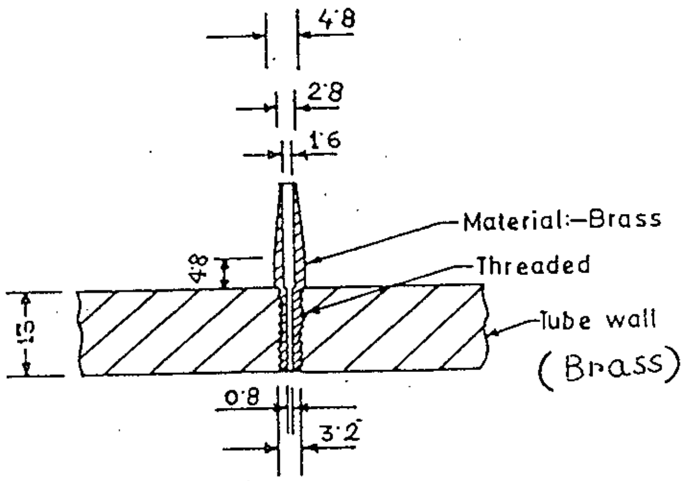
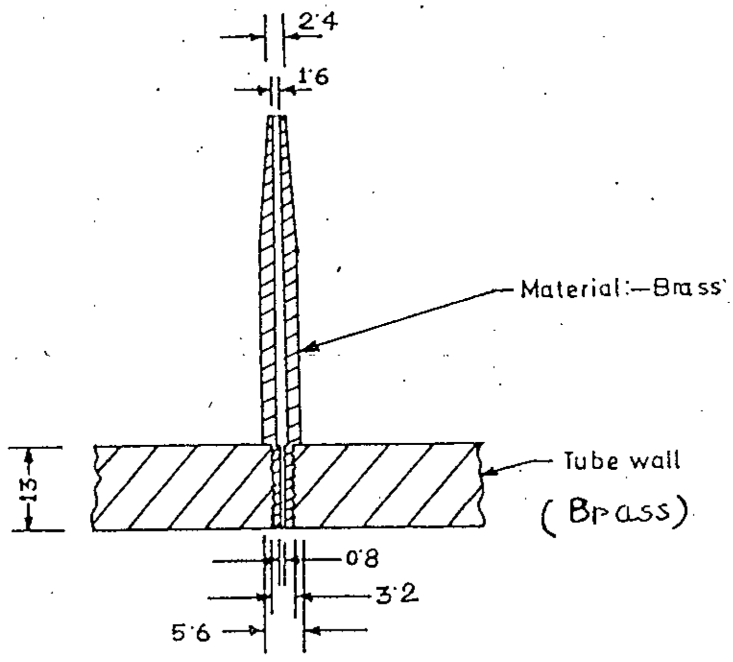



FIG: A7 ORIFICE METER



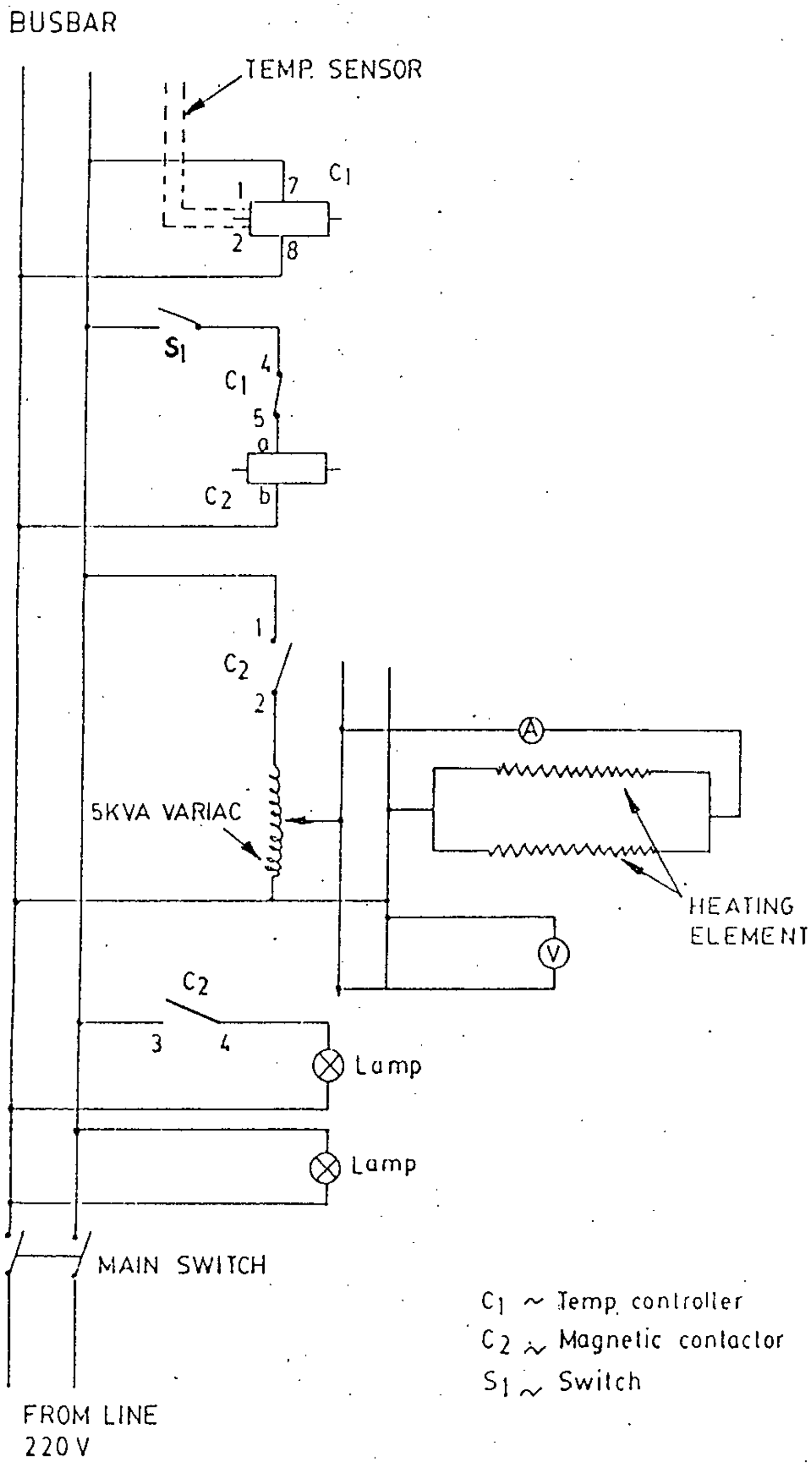

PRESSURE TAPPINGS AT SHAPED INLET




PRESSURE TAPPINGS AT THE TEST SECTION

ALL DIMENSIONS ARE IN MM.

FIG. A 8 PRESSUQE TAPPINGS




$C_1$ ~ Temp. controller
$C_2$ ~ Magnetic contactor
$S_1$ ~ Switch

FIG. 1A9 ELECTRIC CIRCUIT DIAGRAM FOR HEATING SYSTEM

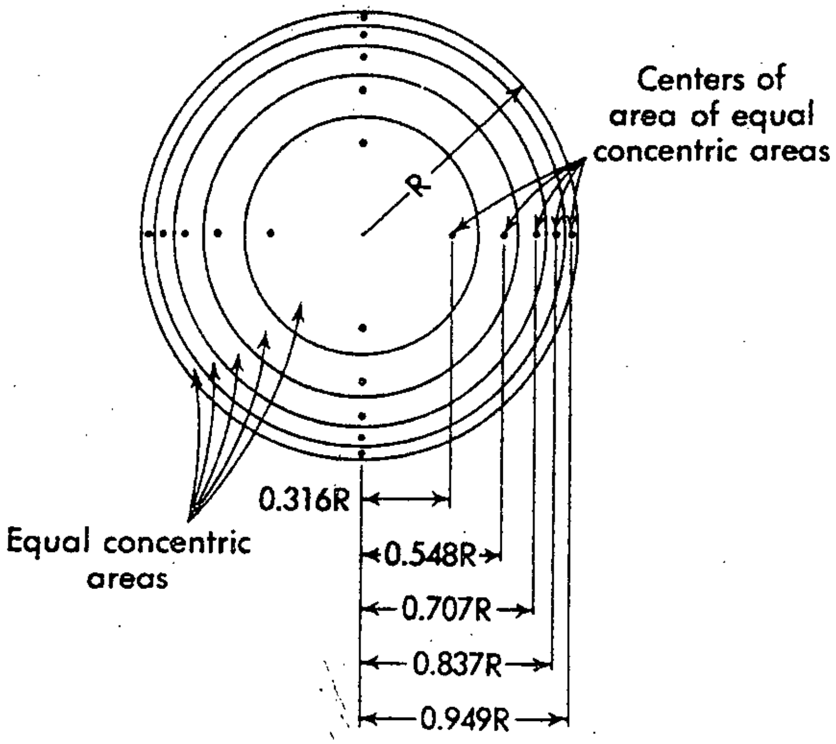



Fig. A10 Location of Pitot Tube for Measurement of Velocity Head

# APPENDIX - B

# GRAPHS

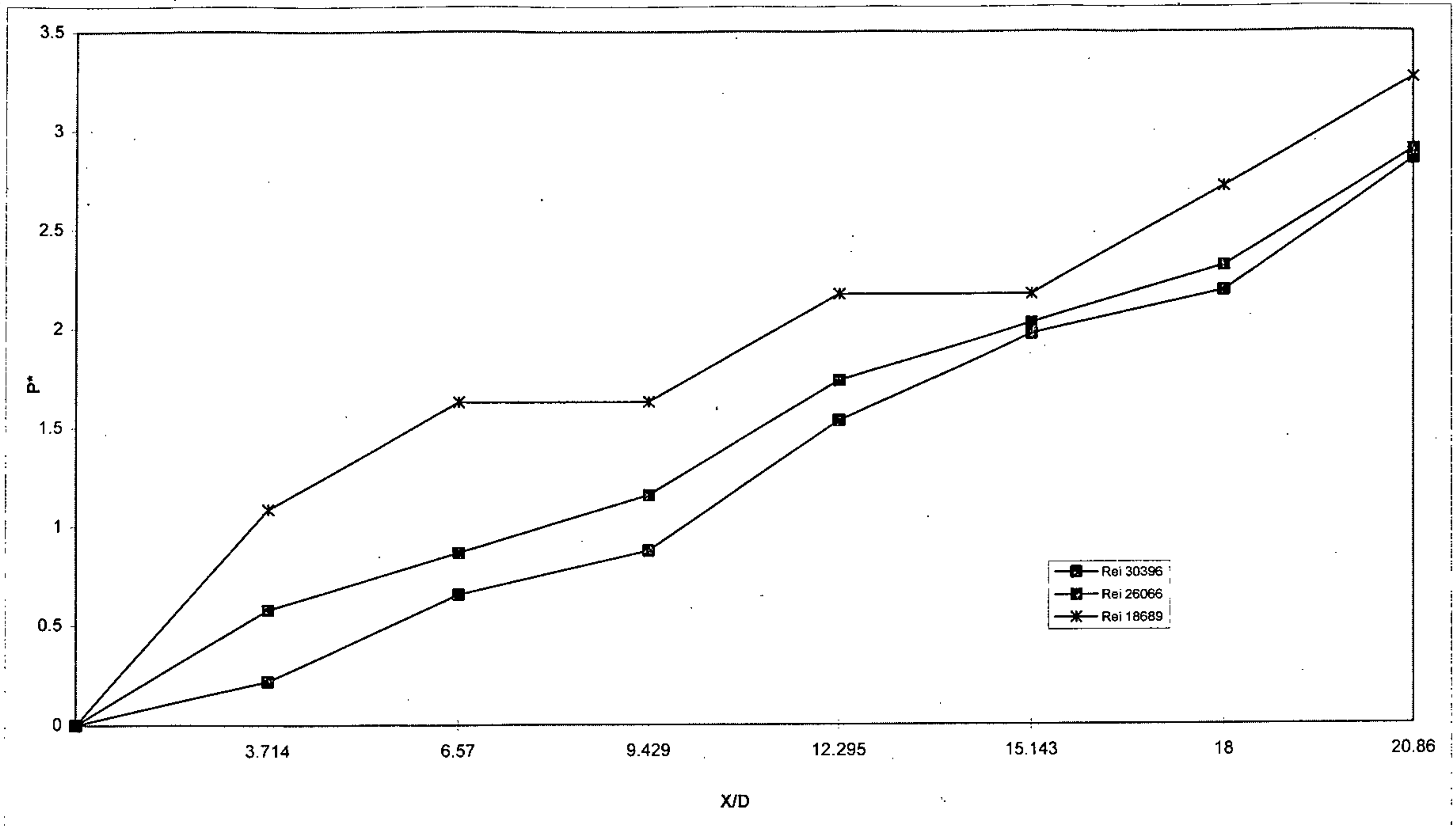



**Fig.B1 Pressure Drop Along the Length of the Tube.**

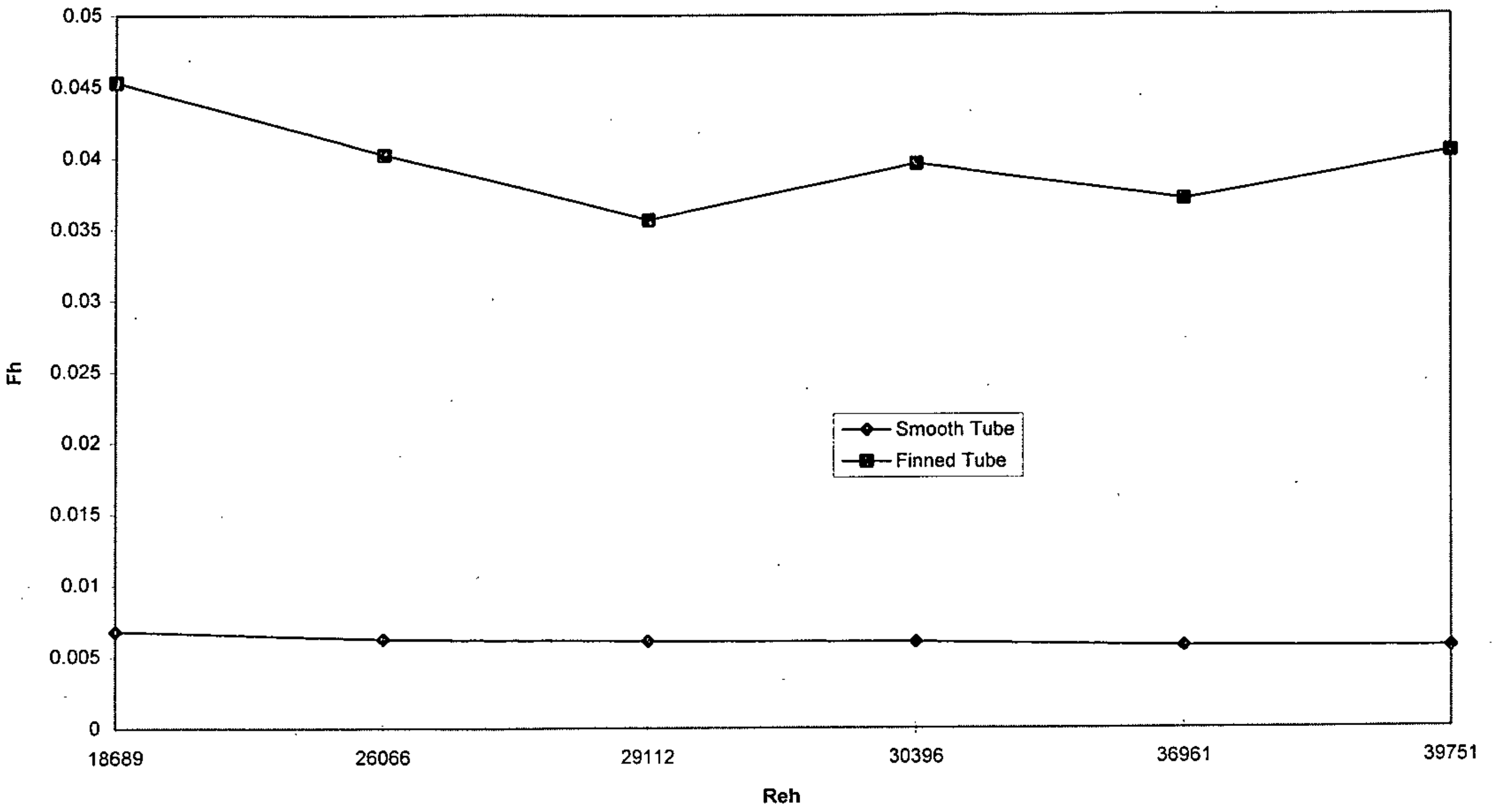
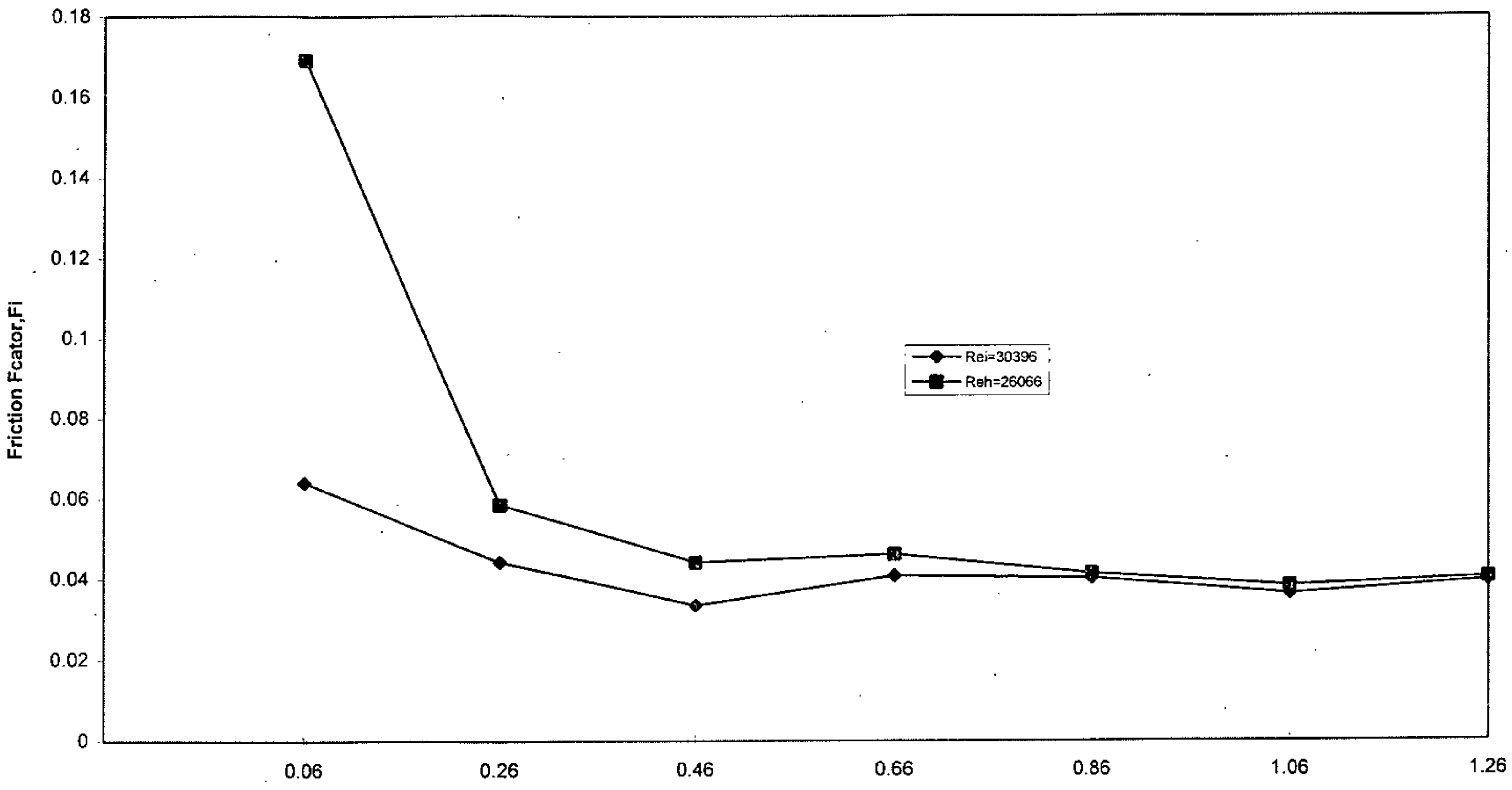



**Fig. B2 Friction Factor Based on Hydraulic Diameter**



Axial Distance in meter

**Fig.B3 Distribution of Friction Factor Along Axial Distance**

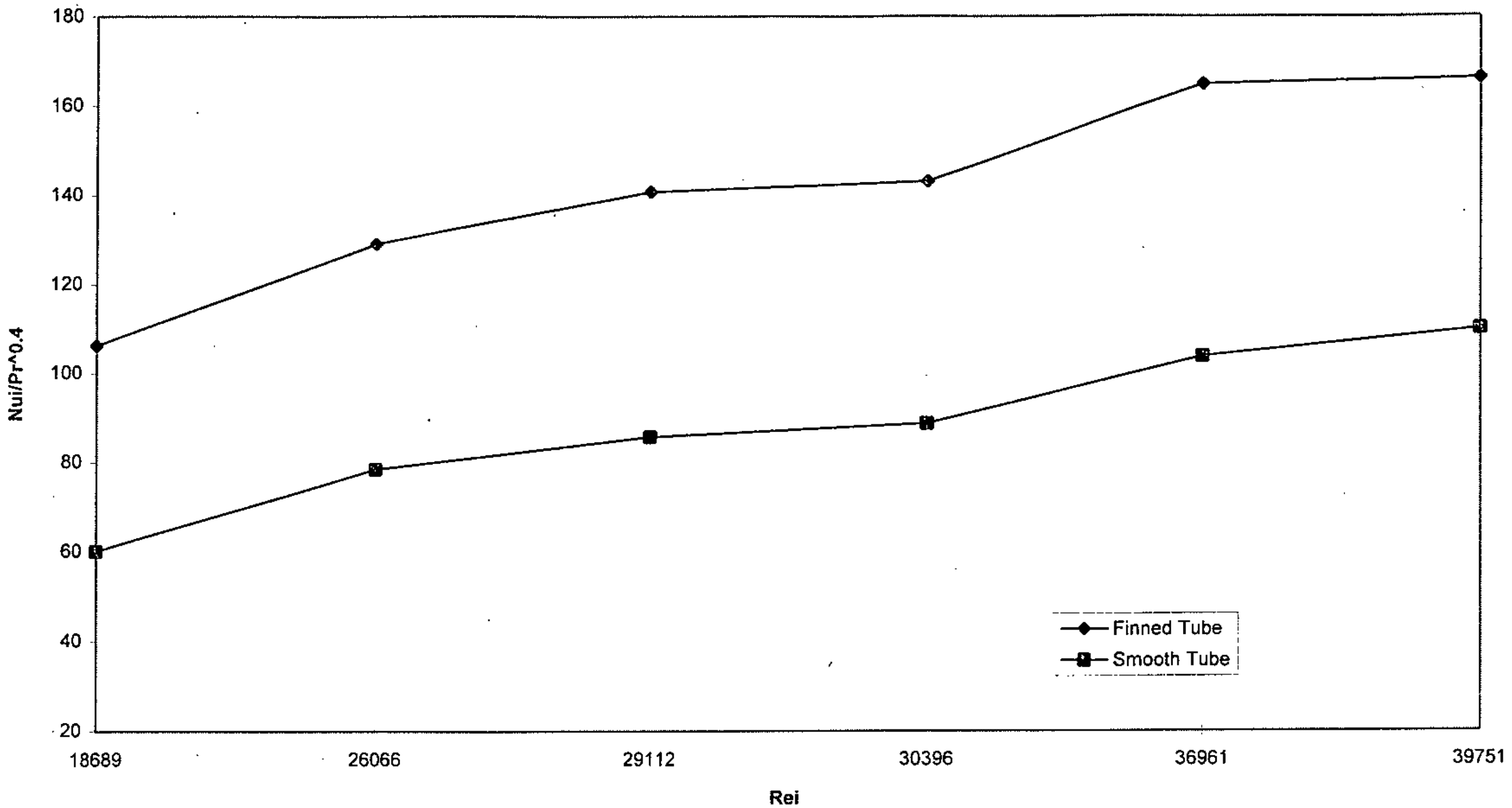



**Fig. B12 Heat Transfer Result Based on Inside Diameter and Nominal Area**

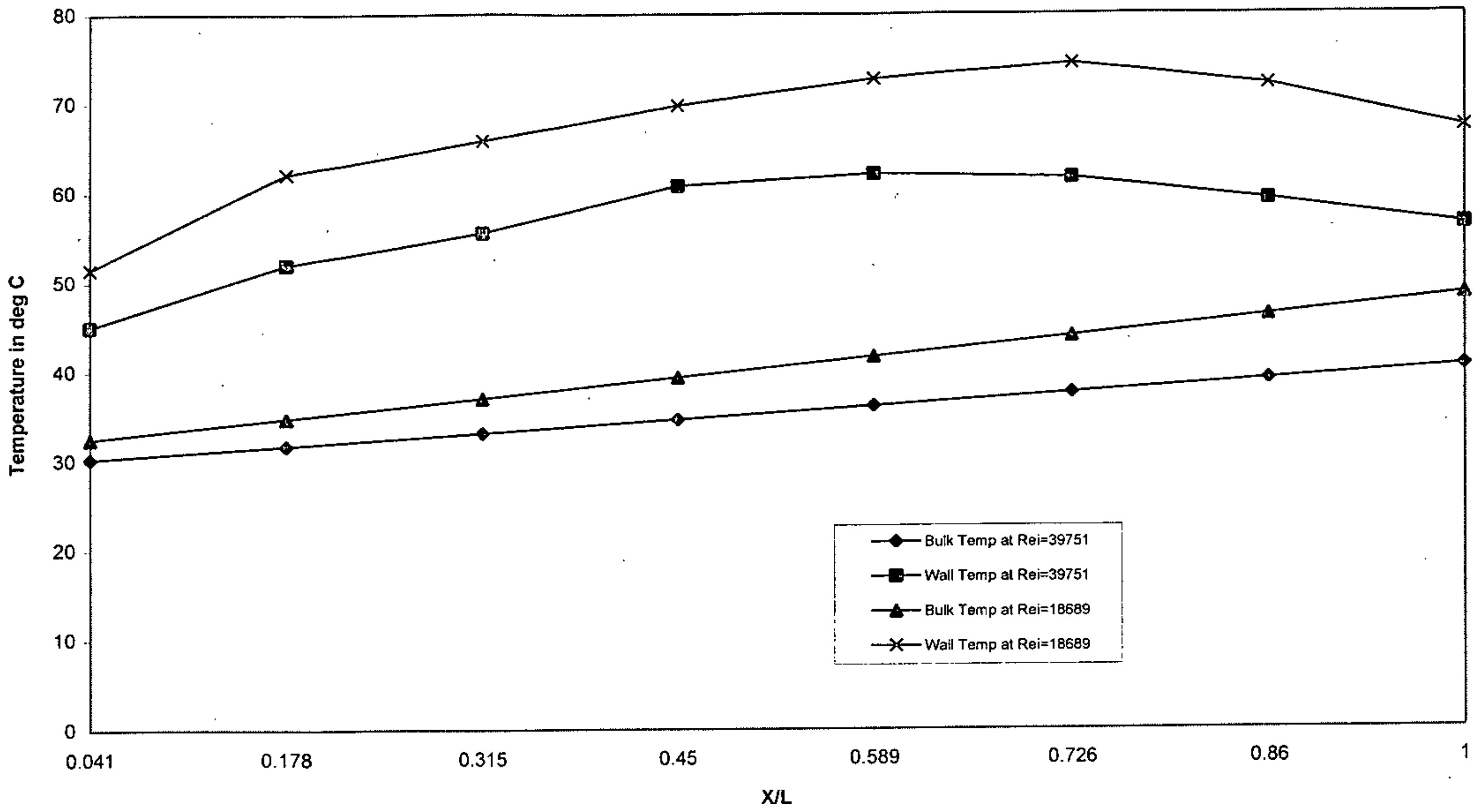



**Fig. B5 Wall and Bulk Temperature Distribution Along Axial Distance**

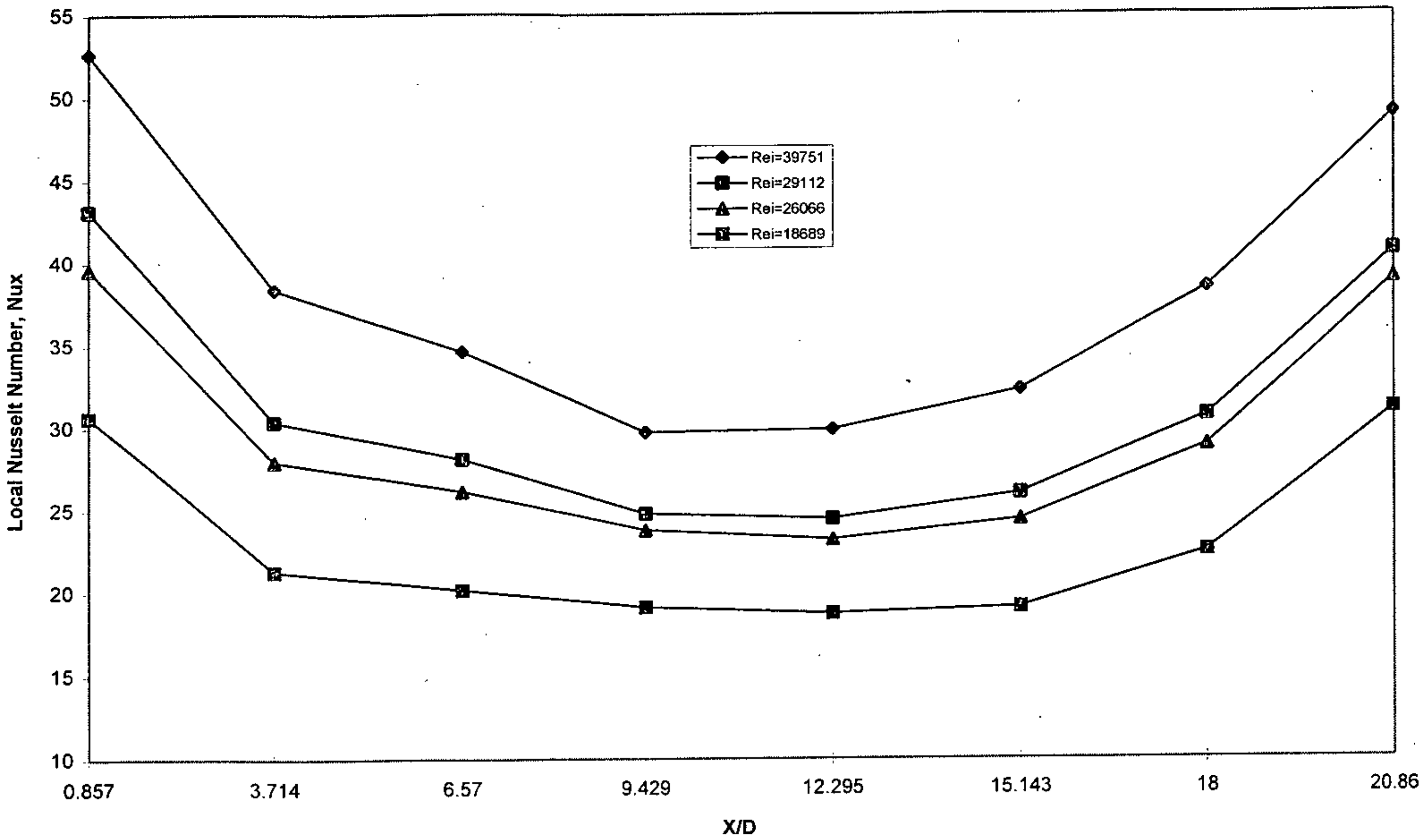



**Fig. B6 Local Nusselt Number Along the Length of Fin Tube at Different Reynolds Number**

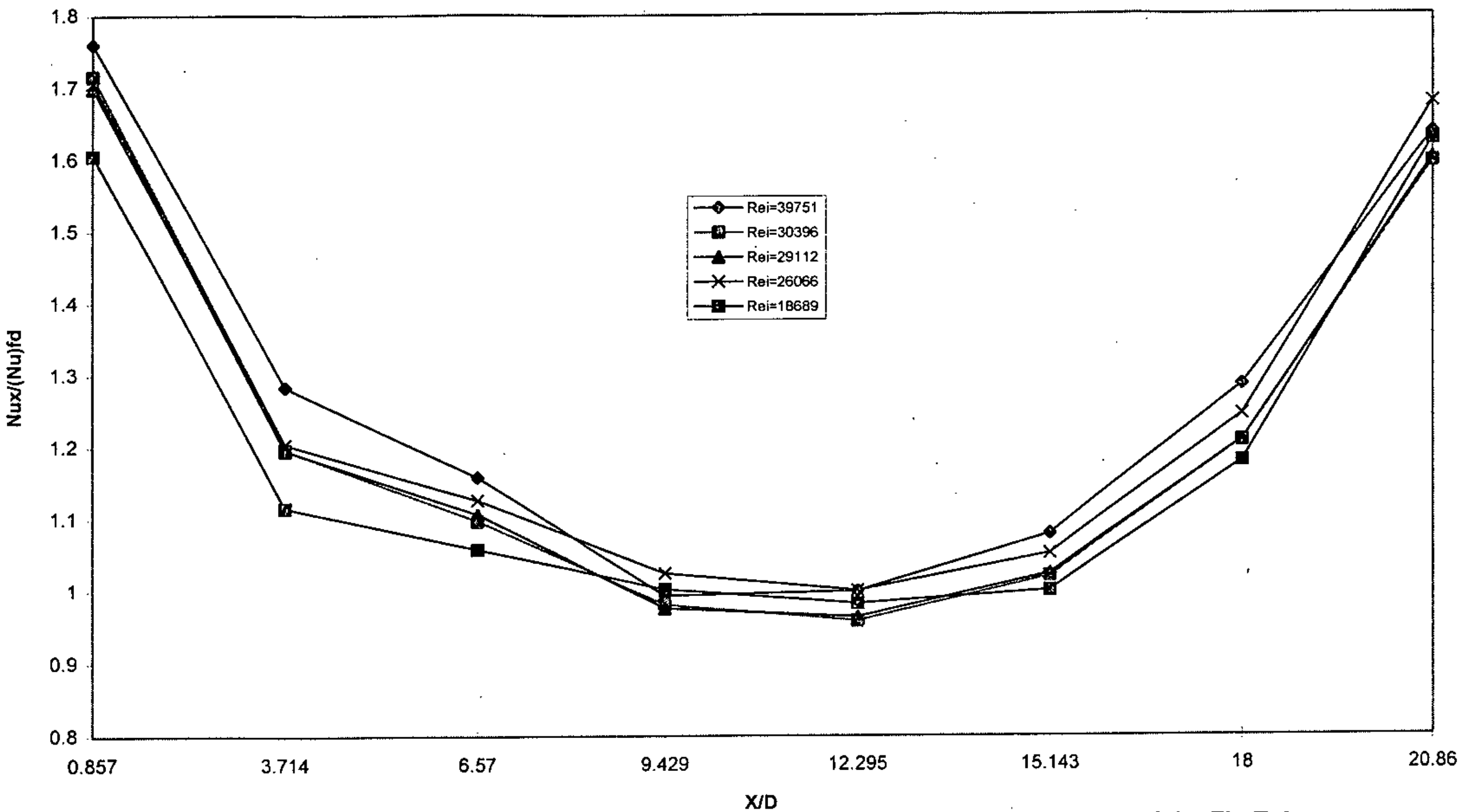



**Fig. B7 Distribution of Nusselt Number Ratio Nux/(Nu)FD Along the Length of the Fin Tube**

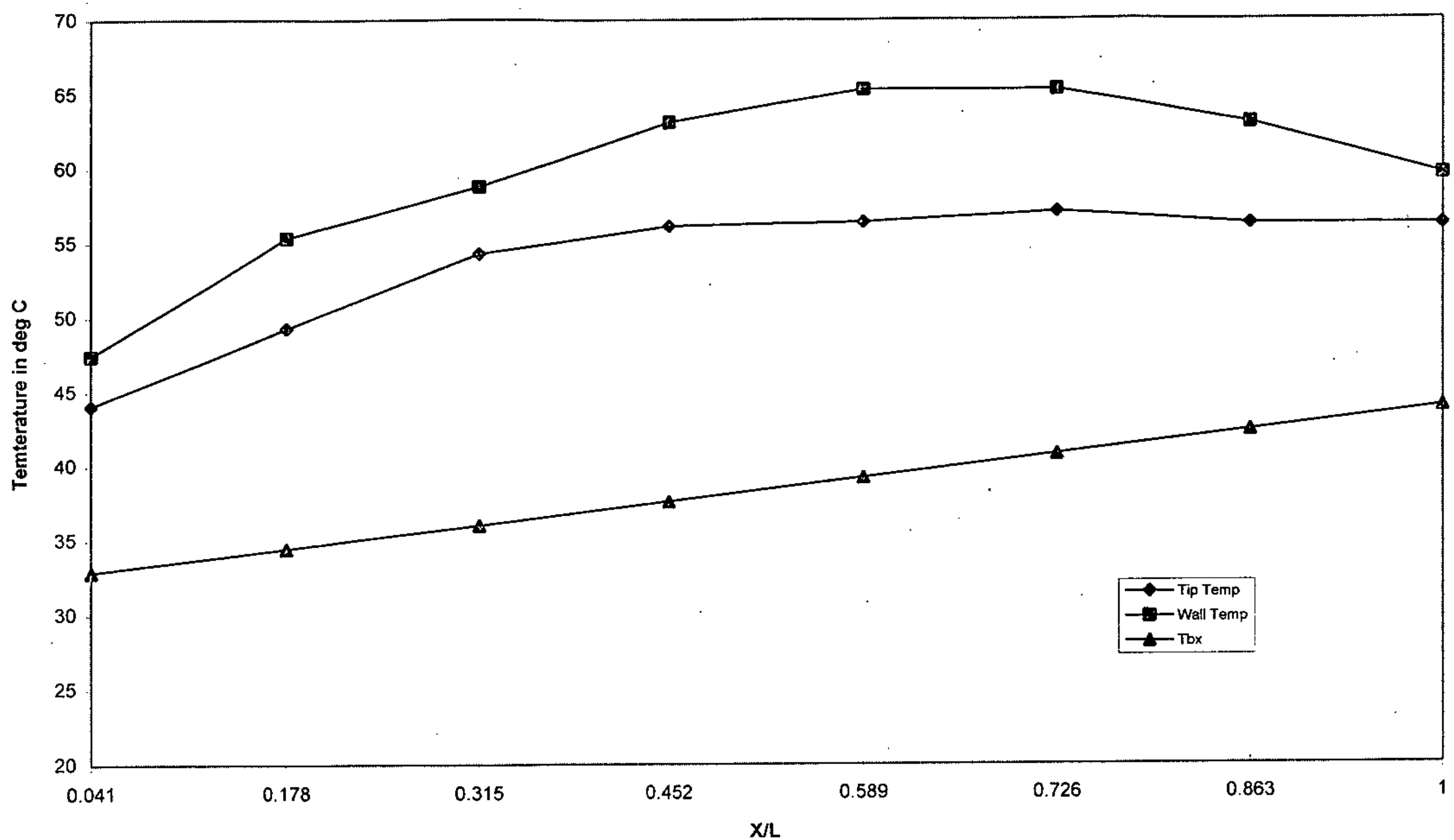



**Fig. B8 Wall, Fin-Tip and Bulk Temperature Distribution Along Axial Distance (Rei = 29122)**

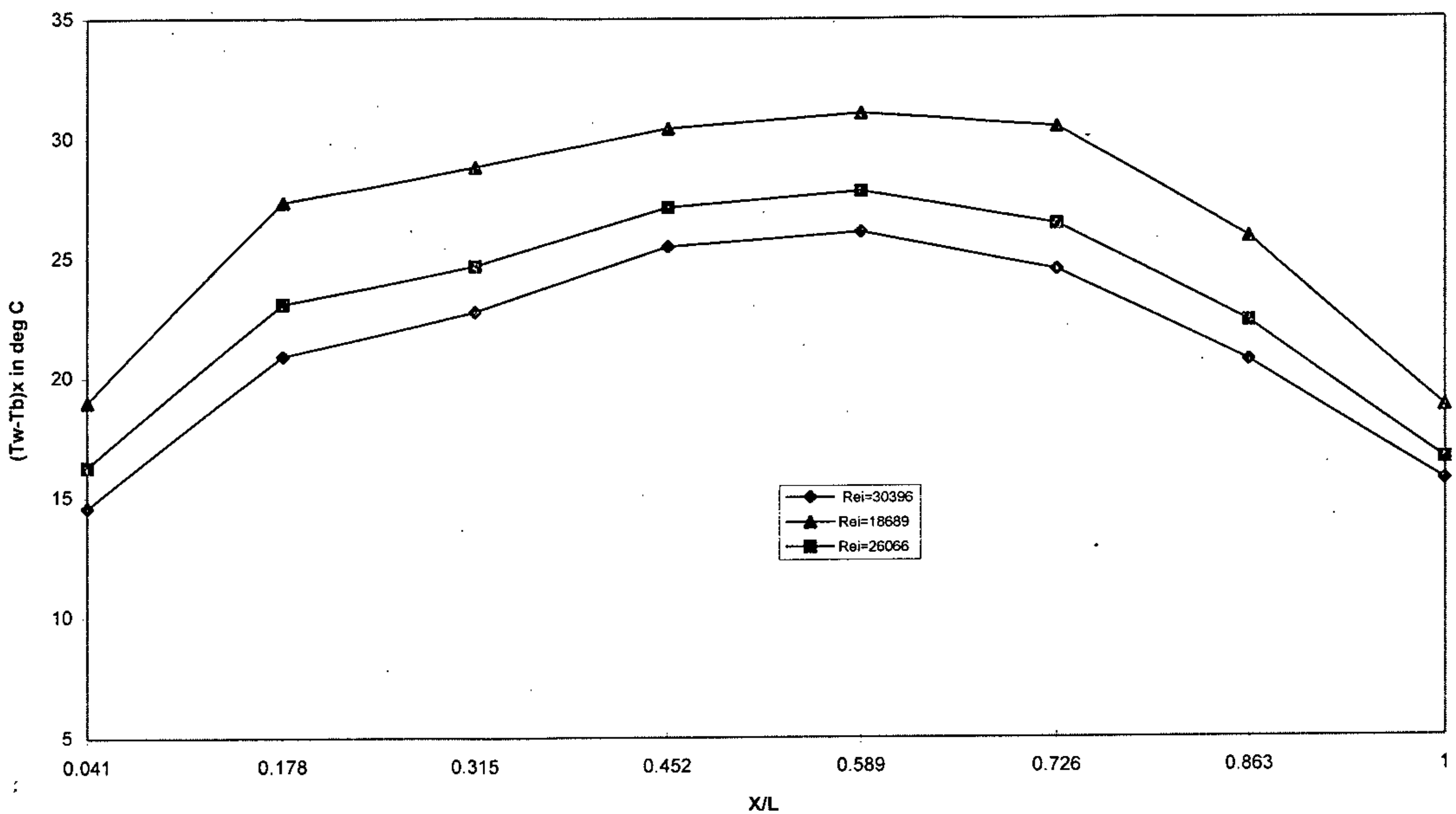



Fig. B9 Distribution of Wall to Bulk Temperature Along the Length of the Tube

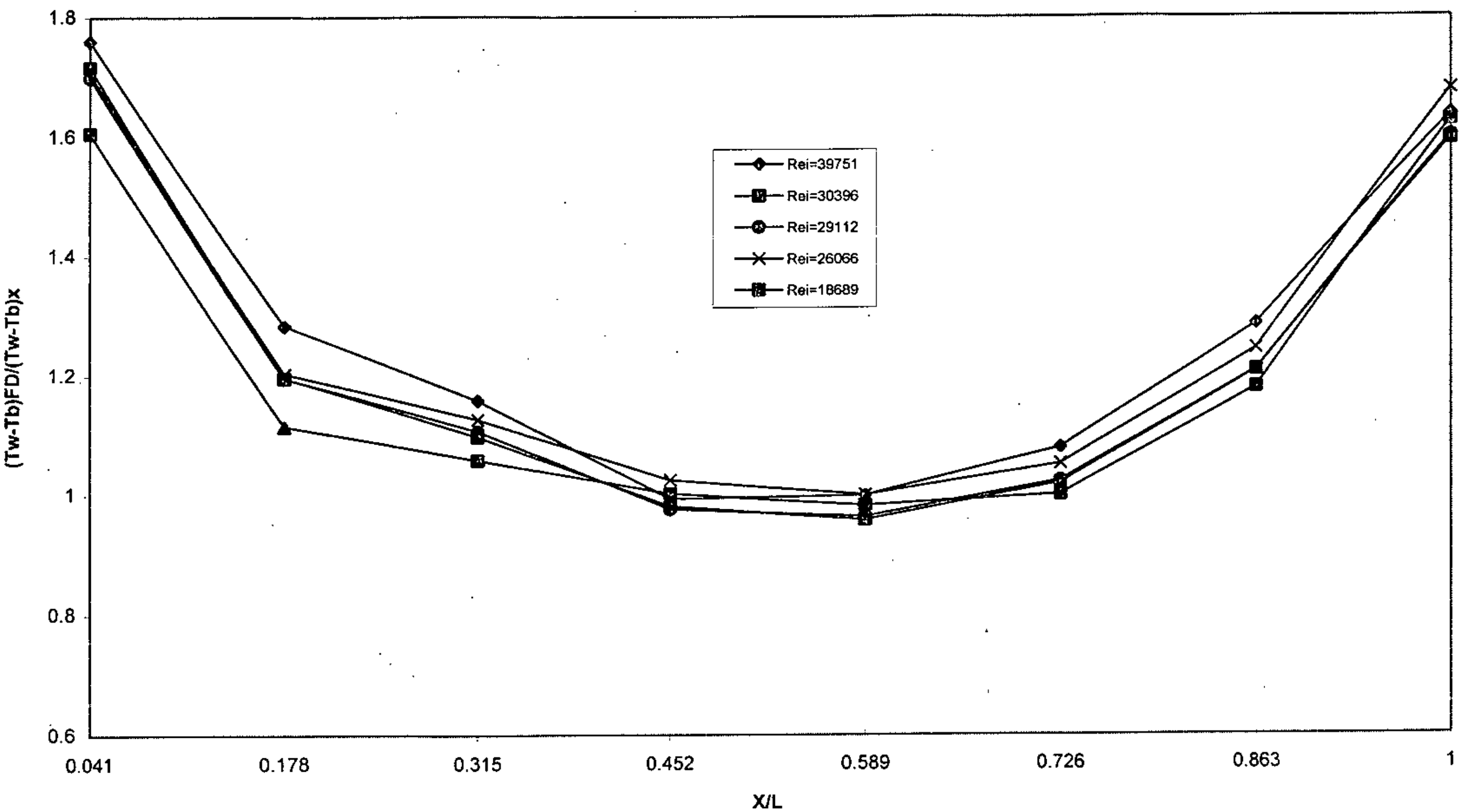



**Fig. B10 Distribution of Temperature Ratio (Tw-Tb)fd/(Tw-Tb)x Along the Length of the Fin Tube**

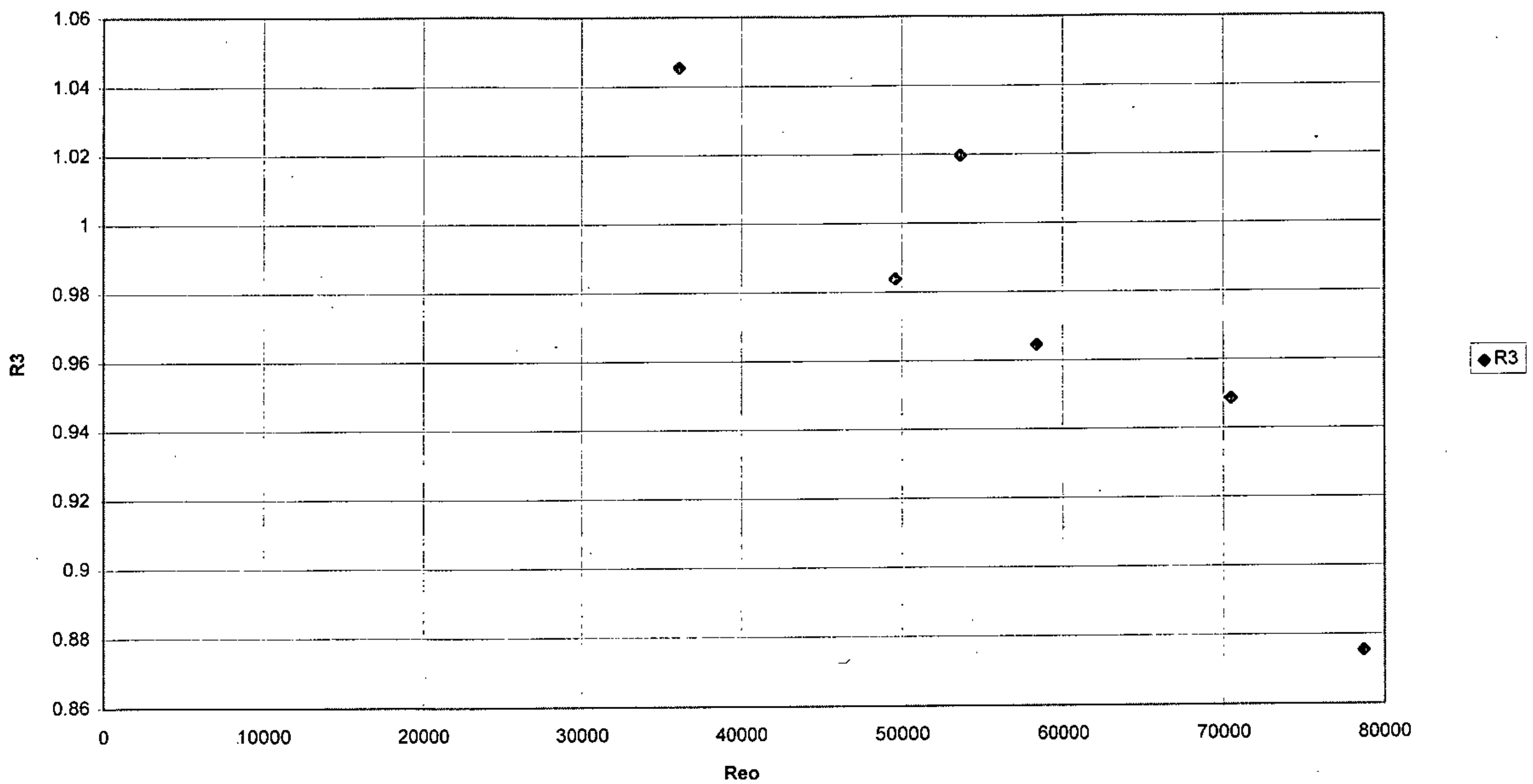



Fig. B11 Constant Pumping Power Comparison With Smooth Tube

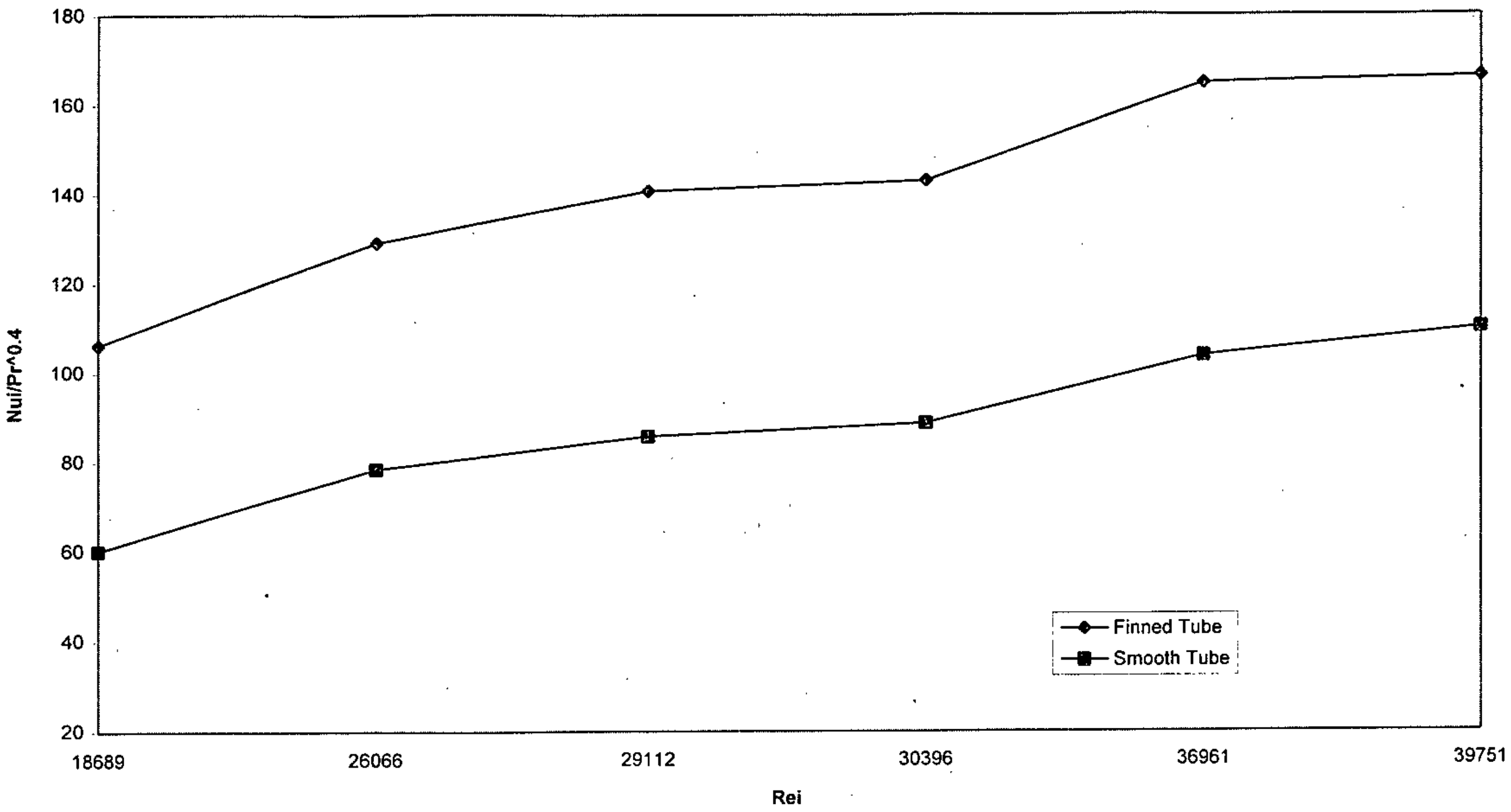


Fig. B12 Heat Transfer Result Based on Inside Diameter and Nominal Area

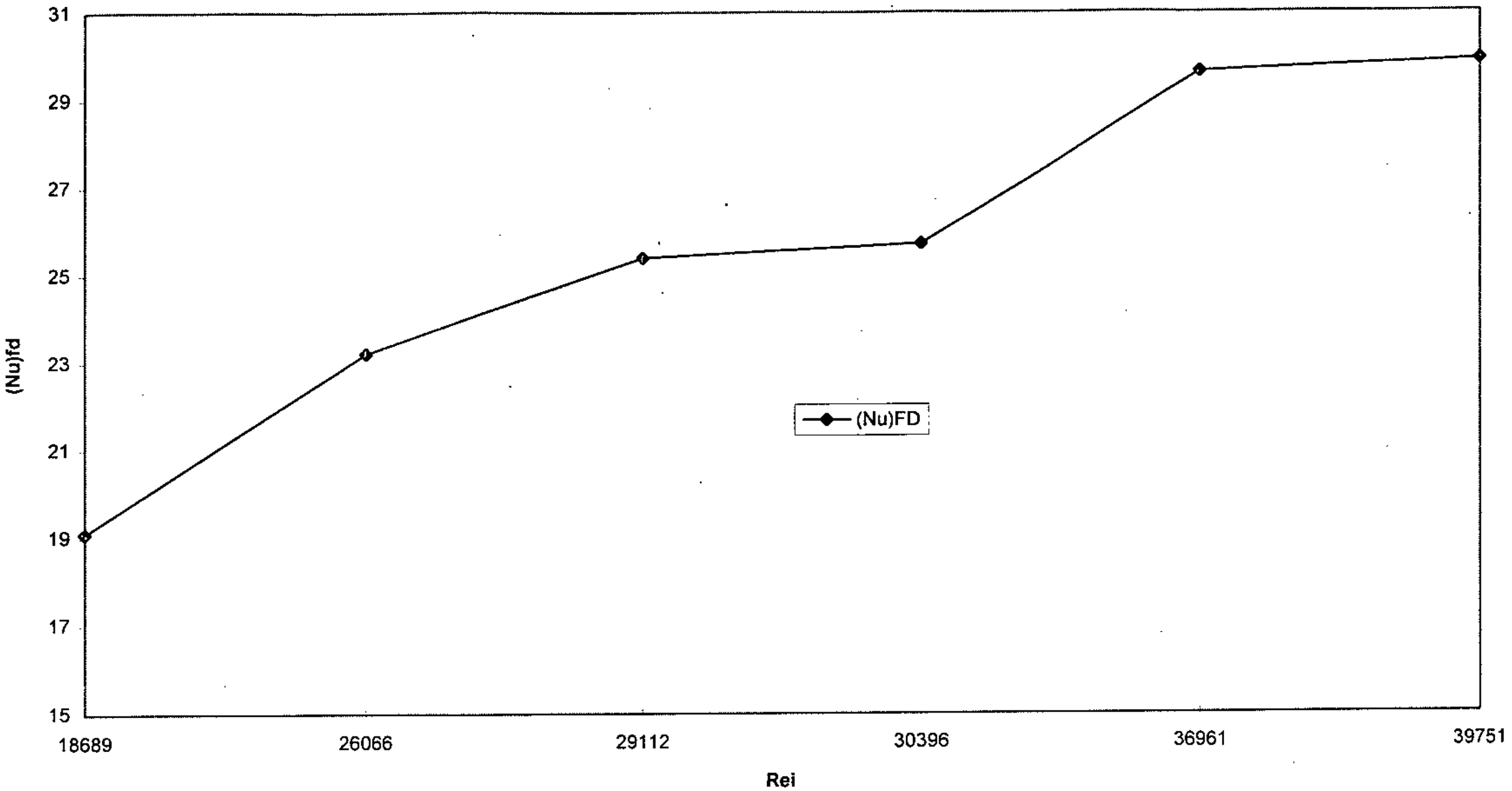


**Fig. B13 Fully Developed Nusselt Number at Different Reynolds Numbers**

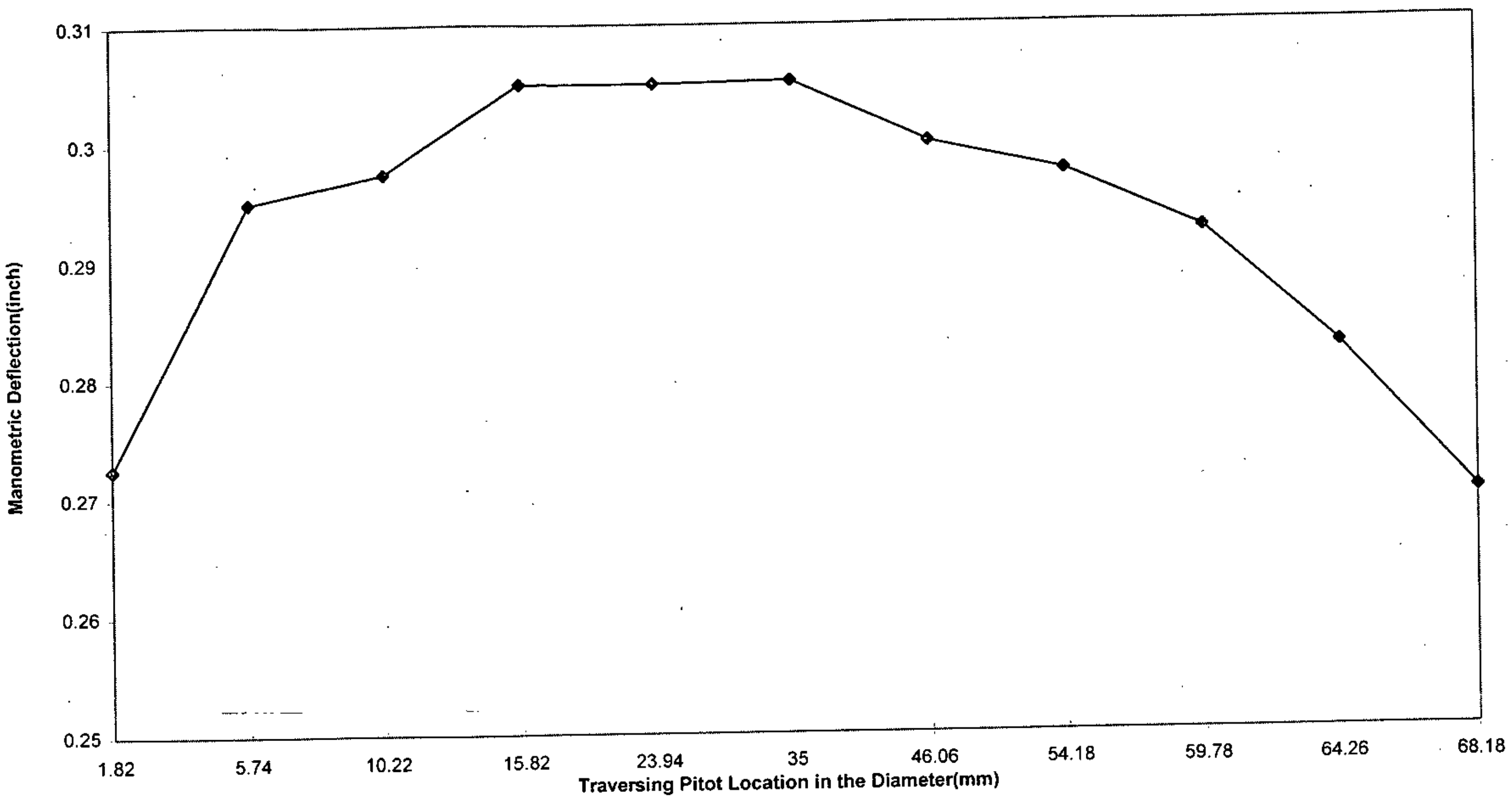



Fig. B14 Velocity Head Along the Diameter of The Tube

# APPENDIX - C
# SPECIFICATION

## Table-C1

**1. Fan:**

| | | |
|---|---|---|
| Capacity | : | 30 $m^3$/min. |
| Pressure | : | 125 mm of water |
| H.P | : | 3 |
| Phase | : | 3 |
| Current | : | 4.1 A |
| Voltage | : | 380 V |

**2. Temperature Controller:**

| | | |
|---|---|---|
| Range | : | 0 - 200°C |
| Input voltage | : | 220 V |

**3. Electric Heating System:**

| | | |
|---|---|---|
| Heat Resistance | : | 8.75 Ohm |
| Maximum Voltage | : | 220 Volts |
| Maximum Current | : | 25 A |
| Power | : | 5.5 kW |

4. **Data Acquisition System:**

**Cole - Parmer (USA origin)**

| | | |
|---|---|---|
| Number of Inputs | : | 14 Model PCA - 14 |
| Resolution | : | 16 Bits |
| Accuracy | : | ± 0.02% of range |
| Linearity | : | ± 0.015% |
| Input Impedance | : | 10000 Megohms/0.1 UF |
| Binary Inputs closure | : | 11 ON - OFF TTL or contact |
| | | or 10 - Bit pulse counter |
| Power | : | 120 V 50/60 Hz 0.5 A |

## Table-C2 Shaped Inlet.

**Co-ordinates for Shaped Inlet**

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| x/d | 0 | 0.094 | 0.109 | 0.1225 | 0.141 | 0.156 | 0.172 | 0.188 | 0.203 |
| y/d | 0 | 0 | 0.001 | 0.001 | 0.002 | 0.003 | 0.004 | 0.006 | 0.008 |

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| x/d | 0.218 | 0.234 | 0.250 | 0.266 | 0.281 | 0.297 | 0.312 | 0.328 | 0.344 |
| y/d | 0.057 | 0.067 | 0.078 | 0.091 | 0.107 | 0.127 | 0.154 | 0.219 | 0.284 |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| x/d | 0.438 | 0.422 | 0.406 | 0.391 | 0.375 | 0.359 | 0.344 | 0.328 |
| y/d | 0.308 | 0.325 | 0.338 | 0.347 | 0.353 | 0.358 | 0.361 | 0.362 |

# APPENDIX- D
# TABLES

**Table-D1:** **Comparison of Friction Factor of Finned Tube Based on Hydraulic Diameter.**

| Reynolds Number Reh | Friction Factor | | $F_h/F_o$ | Increase in Friction Factor Over Smooth Tube % |
|---|---|---|---|---|
| | Finned Tube $F_h$ | Smooth Tube $F_o$ | | |
| 17207 | 0.006898 | 0.017508 | 2.538278 | 153.8278 |
| 16020 | 0.007022 | 0.016069 | 2.288355 | 128.8355 |
| 13175 | 0.007374 | 0.017161 | 2.327344 | 132.7344 |
| 12636 | 0.007451 | 0.015461 | 2.075034 | 107.5034 |
| 11298 | 0.007663 | 0.017448 | 2.27703 | 127.703 |
| 8100 | 0.008327 | 0.01964 | 2.358448 | 135.8448 |

**Table-D2: Comparison of Friction Factor of Finned Tube Based on Inside Diameter with that of Smooth Tube.**

| Reynolds Number Rei | Friction Factor | | $F_i/F_o$ | Increase in Friction Factor Over Smooth Tube |
|---|---|---|---|---|
| | Finned Tube $F_i$ | Smooth Tube $F_o$ | | % |
| 18689 | 0.006757 | 0.045312 | 6.706317 | 570.6317 |
| 26066 | 0.006217 | 0.040256 | 6.474697 | 547.4697 |
| 29112 | 0.0060048 | 0.035672 | 5.89825 | 489.825 |
| 30398 | 0.005983 | 0.039594 | 6.617736 | 561.7736 |
| 36961 | 0.005698 | 0.037074 | 6.506929 | 550.6929 |
| 39751 | 0.005595 | 0.040395 | 7.21992 | 621.992 |

**Table-D3:** **Comparison of Heat Transfer Coefficient of Finned Tube Based on inside Diameter and Nominal area with that of Smooth Tube at Constant Reynolds Number.**

| Reynolds Number $Re_i$ | Heat Transfer Coefficient of Finned Tube $h_{if}$ | Heat Transfer Coefficient of smooth Tube $h_o$ | $h_{if}/h_o$ |
|---|---|---|---|
| 39751 | 56.92013 | 37.65299 | 1.511703 |
| 36961 | 56.82996 | 35.76057 | 1.58918 |
| 30396 | 49.30394 | 30.58177 | 1.612201 |
| 29112 | 48.6945 | 29.65012 | 1.642304 |
| 26066 | 44.67083 | 27.14116 | 1.645871 |
| 18689 | 36.99158 | 20.93593 | 1.766895 |

**Table-D4:** **Comparison of Nusselt Number Based on inside Diameter and Nominal area of Finned Tube with that of Smooth Tube at Constant Reynolds Number.**

| Reynolds Number $Re_i$ | Heat Transfer Coefficient of Finned Tube $Nu_{if}$ | Heat Transfer Coefficient of smooth Tube $Nu_o$ | Percentage of Increase of Nusselt Number over Smooth Tube. % |
|---|---|---|---|
| 39751 | 144.3626 | 95.49761 | 51.17028 |
| 36961 | 143.097 | 90.0446 | 58.91795 |
| 30396 | 124.1466 | 77.00445 | 61.22006 |
| 29112 | 122.1726 | 74.39099 | 64.23035 |
| 26066 | 112.0774 | 68.09609 | 64.58706 |
| 18689 | 92.14984 | 52.15355 | 76.68947 |

**Table-D5: Constant Pumping Power Comparison.**

| Fin Tube Reynolds Number $Re_i$ | Equivalent Smooth Tube Reynolds Number at Constant Pumping Power $Re_o$ | Heat transfer Coefficient for Finned Tube $h_{if}$ | Heat Transfer Coefficient for Smooth Tube for Constant Pumping Power $h_{op}$ | Criteria $R_3 = \frac{h_{if}}{h_{op}}$ |
|---|---|---|---|---|
| 39751 | 78695.10 | 56.92013 | 65.02482 | 0.87536 |
| 36961 | 70454.13 | 56.82996 | 59.91487 | 0.948512 |
| 30396 | 58297.31 | 49.30394 | 51.1202 | 0.964471 |
| 29112 | 53543.80 | 48.6945 | 47.75744 | 1.019621 |
| 26066 | 49598.63 | 44.67083 | 45.40781 | 0.98377 |
| 18689 | 36018.04 | 36.99158 | 35.38665 | 1.045354 |

**Table-D6: Constant Pumping Power Comparison.**

| Reynolds Number $Re_i$ | At Constant Reynolds Number $h_{if}/h_o$ | At Constant Pumping Power $h_{if}/h_{op}$ |
|---|---|---|
| 39751 | 1.511703 | 0.87536 |
| 36961 | 1.58918 | 0.948512 |
| 30396 | 1.612201 | 0.964471 |
| 29112 | 1.642304 | 1.019621 |
| 26066 | 1.645871 | 0.98377 |
| 18689 | 1.766895 | 1.045354 |

# APPENDIX-E
# SAMPLE CALCULATIONS

Internal diameter of the tube, $D_i = 70$ mm

Fin height, $H = 15$ mm

Fin width, $W = 3$ mm

Number of fin, $N = 8$

$$A_x = \frac{\pi D_i^2}{4} = 3.8485 \times 10^{-3} \text{sq.m}$$

$$A_{xf} = \left(\frac{\pi D_i^2}{4}\right) - WHN = 3.4885 \times 10^{-3} \text{sq.m}$$

$$A_h = \pi D_i + 2HN = 0.45991 \text{ m}$$

**Hydraulic Diameter:**

$$D_h = \frac{4\,A_{xf}}{\pi D_i + 2HN} = \frac{4\,A_{xf}}{A_h} = 0.030340 \text{ m}$$

**Determination of Mean Velocity:**

$$\Delta P' = \frac{1}{2}\rho V^2 \qquad \text{(E-1)}$$

If V is to be m/s, $\rho$ must be expressed in kg/m$^3$ and $\Delta P'$ in Pascal (N/m$^2$). If h is the velocity head expressed in cm of water

$$\Delta P = \gamma_{water} h = 9.81 \times 10^3 \times \frac{h}{10^2} = 98.1 \times h \quad N/m^2 \qquad \text{(E-2)}$$

Standard atmospheric properties at the sea level are

Pressure = 760 mm of Hg

Temperature = 15°C

Density = 1.225 kg/m$^3$

For any other temperature, t $^oC$ and pressure, b mm of Hg, the value of the density in $kg/m^3$ is

$$\rho_2 = \frac{P_2}{P_1} \cdot \frac{T_1}{T_2} \cdot \rho_1 = \frac{b}{760} \times \frac{288}{t+273} \times 1.225$$

$$=> \rho_2 = 0.4642\left(\frac{b}{273+t}\right) \qquad \text{(E-3)}$$

From equations (E-1), (E-2) and (E-3)

$$V = 20.56\left(\frac{273+t}{b}\right)^{1/2} (h)^{1/2}$$

$$=> V = C\,(h)^{1/2}$$

$$\text{Where, } C = 20.56\left(\frac{273+t}{b}\right)^{1/2} \qquad \text{(E-4)}$$

h = velocity head, in cm of water.

For, room temperature, $t = 29.8^oC$

atm. pressure, b = 760 mm Hg

$C = 12.978$

The experiment was done using a manometric fluid of sp. gr. = 0.855 but it was recommended to perform with a fluid of sp. gr. = 0.834. This is why a correction is needed, which was as follows:

$$\omega_1 H_{\omega 1} = \omega_2 H_{\omega 2}$$

$$=> 0.834 \times H_{\omega 1} = 0.855 \times 0.855\, H_D$$

$$\therefore H_{\omega 1} = h_1 = \frac{(0.855)^2 x\, H_D}{0.834}$$

$$= 0.877\ x\ 2.54\ H_D\ cm$$

[$H_D$ in inch of water]

Measurement of mean velocity by ten points Log Linear method is given by,

$$\text{Mean velocity, } V_i = \frac{C}{10}\left(h_1^{1/2} + h_2^{1/2} + h_3^{1/2} + \ldots\ldots, + h_{10}^{1/2}\right)$$

$$=> V_i = 10.094 \text{ m/s}$$

Mass Flow rate, $M = \rho A_X V_i$

$= 1.134438 \times 0.00385 \times 10.094$

$= 0.044086$ kg/s

[$\rho = 1.0824$ kg/m$^3$ of air at 317.53 K]

$$V = \frac{M}{\rho A_{xf}} = 11.674\ m/s$$

**Reynolds Number:**

$$R_{ei} = \frac{\rho V D i}{\mu} = 39751$$

**Friction Factor:**

Local friction factor based on inside diameter is given by

$$F_i = \frac{(-\Delta P/x) D_i}{2\rho V^2}$$

$$= \frac{(-\Delta P/x) \times 0.07 \times 9.81}{2 \times 1.0824 \times (11.674)^2}$$

$$= 2.3276 \times 10^{-3}(-\Delta P/x) \qquad \text{(E -5)}$$

Local fiction factor based on hydraulic diameter is given by

$$Fh = \frac{(-\Delta P/X) Dh}{2\rho V^2} \qquad \text{(E-6)}$$

| h1 (mm) | h2 (mm) | delh (mm) | delP (mm) | delP (kg/m**2) | Dimless P | del X | delP/delX | Fi |
|---|---|---|---|---|---|---|---|---|
| 29 | 34 | 5 | | | | | | |
| 82 | 90 | 8 | 3 | 2.983 | 0.396 | 0.06 | 49.728 | 0.115 |
| 72 | 85 | 13 | 8 | 7.956 | 1.0576 | 0.26 | 30.6018 | 0.071 |
| 56 | 71 | 15 | 10 | 9.945 | 1.322 | 0.46 | 21.620 | 0.050 |
| 61 | 79 | 18 | 13 | 12.929 | 1.7186 | 0.66 | 19.5898 | 0.045 |
| 86 | 107 | 21 | 16 | 15.912 | 2.1152 | 0.86 | 18.5034 | 0.043 |
| 69 | 92 | 23 | 18 | 17.902 | 2.3796 | 1.06 | 16.8887 | 0.039 |
| 64 | 91 | 27 | 22 | 21.880 | 2.9084 | 1.26 | 17.3653 | 0.040 |

**Heat Transfer Calculation:**

$T_i = 29.8°C$

$T_o = 40.95°C$

Properties of air are calculated at $T_f = 317.53$ K

$C_p = 1.0058$ KJ/kg-°C

$K = 0.0276$ w/m-°C

$\mu = 19.2643$ microkg/sec-m

$\rho = 1.0831$ kg/m3

Total heat input to the air

$Q = MC_p (T_o - T_i) = 494.44$ Watt.

Q' = heat input per unit length

$$= \frac{Q}{L} \qquad [L = 1.52 \text{ m}]$$

$$= 325.287 \text{ W / m}$$

The local bulk temperature of this fluid is,

$$T_{bx} = T_i + \frac{Q'_x}{MC_p}$$

$$= 29.8 + \frac{325.287 \text{ x}}{0.044086 \text{ x } 1.0058 \text{ x } 10^3}$$

$$\therefore T_{bx} = 29.8 + 7.336x \qquad \text{(E-7)}$$

| X | Tbx |
|---|---|
| 0.06 | 30.24 |
| 0.26 | 31.707 |
| 0.46 | 33.174 |
| 0.66 | 34.641 |
| 0.86 | 36.108 |
| 1.06 | 37.576 |
| 1.26 | 39.043 |
| 1.46 | 40.510 |

Local convective heat transfer coefficient is given by,

$$h_x = \frac{Q'}{(T_w - T_b)_x A_h}$$

$$= \frac{325.287}{(T_w - T_b)_x 0.4599}$$

$$= \frac{707.299}{(T_w - T_b)_x} \text{ W / m}^{2\circ}\text{C} \qquad \text{(E-8)}$$

Local Nusselt Number is

$$N_{ux} = \frac{h_x\, D_h}{K}$$

$$= 1.099\, h_x \qquad \text{(E-9)}$$

| hx | Nux |
|---|---|
| 47.87 | 52.625 |
| 34.913 | 38.379 |
| 31.516 | 34.645 |
| 27.026 | 29.709 |
| 27.183 | 29.881 |
| 29.347 | 32.260 |
| 34.973 | 38.445 |
| 44.507 | 48.925 |

From the graph $(T_W - T_b)_X$ Vs x/L, Fig. 2.9

$$(T_W - T_b)_{FD} = 26^oC$$

Fully developed heat transfer coefficient,

$$h = \frac{Q'}{A_h(T_w - T_b)_{FD}} = 27.20\ W/m^{2o}C$$

Fully developed Nusselt Number,

$$Nu = \frac{hD_h}{K} = 29.9045$$

Heat Transfer Coefficient based on inside diameter and nominal area

$$h_{if} = \frac{hA_h}{A} = 56.92\ W/m^{2o}C$$

Nusselt Number based on inside diameter and nominal area

$$Nu_{if} = \frac{h_{if}D_i}{K} = 144.36$$

**Constant Pumping Power Result:**

Constant Pumping Power smooth tube Reynolds Number is given by,

$$Re_o = 2.17(Re_{if})^{12/11}\left(A_{xf}F_{if}/A_{xo}\right)^{4/11} = 78695.10$$

For smooth tube,

$$Nu_{op} = 0.023(Re_o)^{0.8}(Pr)^{0.4}$$

$$= 164.92$$

Heat Transfer Coefficient for smooth tube at $Re_o = 78695.10$

$$h_{op} = \frac{Nu_{op}.K}{D_i} = 65.02\ W/m^{2o}C$$

$$\therefore R_3 = \frac{h_{if} \text{ at } Re_{if}}{h_{op} \text{ at } Re_o} = 0.875$$