STUDY OF HEAT TRANSFER PERFORMANCE OF A TUBE HAVING INTERNAL FINS

BY

MAFIZUL HUQ

B.Sc.Engg.(Mech.)

A THESIS

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EXAMINERS

Dr. A. M. Aziz-ul Huq — Chairman
Professor,
Dept. of Mechanical Engg.
B.U.E.T.,Dhaka

Professor and Head — Member
Dept. of Mechanical Engg.
B.U.E.T,Dhaka

Dr. Abdur Razzaq Akhanda — Member
Associate Professor,
Dept. of Mechanical Engg.
B.U.E.T,Dhaka

Dr. Abdur Rashid Sarker — Member
Assistant Professor,
Dept. of Mechanical Engg.
B.U.E.T.,Dhaka

Khaliqur Rahman

Dr. Khaliqur Rahman — Member(External)
Professor,
Dept. of Chemical Engg.
B.U.E.T.,Dhaka

THIS IS TO CERTIFY THAT THIS WORK HAS BEEN DONE BY ME AND HAS NOT BEEN SUBMITTED ELSEWHERE FOR THE AWARD OF ANY DEGREE OR DIPLOMA.

SIGNATURE OF THE SUPERVISOR

SIGNATURE OF THE CANDIDATE

A B S T R A C T

Steady state turbulent flow heat transfer performance of circular tube having integral internal fins was experimentally studied. An experimental set-up was designed to study the heat transfer performance in the entrance region as well as in fully developed region.

The test section is 1520 mm long, 70 mm inside diameter, fin height is 15 mm and the ratio of fin height to radius is 0.429. Six numbers of fins were made integral with circular tube to avoid the contact resistance. The fin and tube assembly was cast from aluminum, as aluminum has high thermal conductivity and easy machinability.

Air was used as the working fluid in heating mode and Reynolds number based on hydraulic diameter was in the range of 10^4 to 10^5. Heat was supplied from an electric heating system and the heating mode was kept constant per unit axial length of the tube.

Heat transfer data were presented both on the basis of nominal area of an unfinned tube and of the effective area of the finned tube. Nusselt number of finned tube was compared with that of theoretically obtained values for unfinned tube (smooth tube) for both constant Reynolds number and constant pumping power.

Results exhibit high pressure gradients and high heat transfer coefficients in the entrance region,approaching, the fully developed values away from the entrance section.

Heat transfer results, based on inside diameter and nominal area for finned tube exceeded unfinned tube values by 97% to 112% for Reynolds number range from 2.66×10^4 to 7.86×10^4. When compared with a unfinned tube (smooth tube) at constant pumping power, an improvement as high as 52% was obtained in heat capacity.

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TO MY PARENTS

C O N T E N T S

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N O M E N C L A T U R E

A_x	Cross sectional area of tube having no fins.
A_{xf}	Flow cross sectional area of finned tube.
A_{xc}	Open core free flow area at fin I. D.
A_h	Effective heat transfer area.
A	Nominal (heat transfer) area.
D_i, d	Inside diameter.
D_h	Hydraulic diameter.
N	Number of fins.
H	Height of the fin.
W	Width of the fin.
Re	Reynolds number.
Nu	Nusselt number.
h_o	Heat transfer co-efficient of smooth tube.
h	Heat transfer co-efficient of finned tube.
T	Temperature.
M	Mass flow rate.

Q Total heat input to the air.

Q' Heat input to the air per unit axial length.

x Axial distance.

C_p Specific heat of air.

P Pressure.

V_i Mean velocity in inlet section.

V Mean velocity in finned tube.

V_c Velocity at the axis of the tube.

ρ ρ Density of air.

μ μ Co-efficient of viscosity of air.

ν ν Kinematic viscosity, μ/ρ μ/ρ

k Thermal conductivity of air.

F Friction factor.

L_t Thermal entrance length.

<u>Subscripts.</u>

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.

C H A P T E R - I

INTRODUCTION

Heat Exchangers have numerous applications in power plants, industries,automobiles and electrical and electronic equipments etc. Efficient design of heat exchanger equipments can improve system performance considerably. For the purpose of heat transfer augmentation, the use of internal fins in tubes and annuli have become one of the important research areas in recent years in many countries like United States of America,Canada,United Kingdom,Australia and West Germany etc.

The augmentation of heat transfer performance of circular tubes and annuli by longitudinal internal fins in laminar and turbulent flow has been investigated in several recent experiments (1-9). When compared with smooth tube (unfinned tube), the finned tubes exhibited substantially higher heat transfer coefficients.

Several investigations (10,11) were under taken recently to analyze laminar and turbulent flow heat transfer in circular tubes with longitudinal internal fins. The analysis is based on the differential equations for momentum and energy conservation in the flowing fluid. The solutions were obtained by numerically integrating the partial differential equations. Experimental studies have also been carried out for augmentation of heat transfer performance of circular tubes having internal fins. Most of the research investigations of turbulent forced convection were made with fins in tubes and annulus for fully developed region.

OBJECTIVE

Considering the application of finned surfaces in a variety of practical heat exchanger devices, a research scheme 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 heat transfer performance in entrance region as well as in fully developed region have been studied and compared with that of a smooth tube.

The specific objectives of the present work are enumerated below :

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

(2) To study the effect of longitudinal internal fins in a circular tubes on friction factor and pressure drop in the entrance region as well as in fully developed region.

C H A P T E R - II

LITERATURE SURVEY

With the passage of time, fins are being utilized more and more for augmentation of heat transfer. Many research investigations have been carried out to improve heat transfer performance by employing internal fins in tubes. Researchers have been trying to enhance heat transfer in tubes and annuli by (a) making internal surface of the tube rough (b) inserting spiral wire (c) inserting twisted tape and (d) making internal fins in tubes and annuli. Use of internal longitudinal fins in tubes and annuli has got more importance in the research work to enhance heat transfer performance. Many research investigations have been carried out both analytically and experimentally on this topic.

2.1 Experimental Study.

The work on heat transfer and pressure loss in finned tubes began since 1964. Hilding, Coogan and Bargles, et al are pioneers of experimental investigations for this type of augmented surface. Hilding and Coogan (5) presented air data for a variety of tubes with internal fins. Bargles and Morton (6) reworked the data of Hilding and Coogan (5) into the constant pumping power, R_3 comparison.

Bergles et al carried out several investigations and research work (1,4,6) and he made remarkable contribution in this field.

During 1973-75 Watkinson did several research work with finned tube. Watkinson,Milette and Trassof carried out research work (2) for turbulent flow of water in finned tube in heating mode. In 1975,Watkinson (3) reported on turbulent air flow in the heating mode for tubes 9 and 14 (Table 2.1 in Appendix - A) as well as the other tubes of (2). Watkinson (7) also conducted experiment in laminar oil flow and presented data for eighteen

12.7 to 32 mm diameter tubes containing from 6 to 50 straight or spiral fins over the Prandtl number range of 180 to 250 and the Reynolds number range of 50 to 3000, based on inside tube diameter and nominal area. At a Reynolds number of 500, heat transfer was enhanced over smooth tubes values by 8 to 224% depending on tube geometry. Watkinson used the modified equation of Bergles (1) to calculate the equivalent Reynolds number for the constant pumping power case. At constant pumping power and the same Reynolds number, the increase in heat transfer ranged from 1 to 187%.

During 1976-77, Carnavos (8,9) experimentally determined the heat transfer performance for cooling air in turbulent flow in tubes having integral internal spiral and straight longitudinal fins. He used "Forged Fin" tubes that have integral internal fins manufactured by the Forge-Fin Division of Noranda Metal Industries, Inc. Previous works (1,2,3,7) tested with several fluids in tubes having 6 to 18 integral internal fins, all made by same process with varying fin height, fin helix angles and tube diameters. Carnavos (9) conducted experiments with 21 tubes having integral internal spiral and longitudinal fins. He found that these tubes were potentially cabable of increasing the capacity of an existing heat exchanger, at constant pumping power, by 12% to 66% by direct substitution of an inner fin tube for a smooth tube. Table (2.1) in Appendix - A presents the configurational details of tube nos. 9,14 of Watkinson (3) and 16,24 of Carnavos (9).

Carnavos presented heat transfer and pressure loss results for all the fin tubes on hydraulic diameter and average bulk physical property basis. All the heat transfer data were correlated by Eq (2.1) to within $\pm$ 6% as follows :

$$\frac{Nu}{Pr^{0.4}} = 0.023 \, (Re_h)^{0.8} \left(\frac{A_{xf}}{A_{xc}} \right)^{0.1} \left(\frac{A}{A_h} \right)^{0.5} (\sec \alpha)^3 \qquad (2.1)$$

Pressure data were correlated by Eq (2.2) for Fanning friction factor to with in 7% as follows :

$$F_h = \frac{0.046}{(Re_h)^{0.2}} \left(\frac{A_{xf}}{A_x} \right)^{0.5} (\cos \alpha)^{0.5} \qquad (2.2)$$

Where α = Spiral fin tube helix angle.

Bergles et al (4) outlined several practical criteria for evaluation of the performance of augmented tubes, relative to smooth tube; performance ratios R_3 and R_5 have been developed for criteria 3 and criteria 5. Criterion R_3 aims at improving the heat duty for the case of constant pumping power and for constant basic geometry of the tube (i.e. same diameter and length of the tube), whereas criterion R_5 aims at reducing the heat exchanger size while maintaining the same pumping power and same heat duty. Bergles (4), Watkinson (3) Carnavos (9) and others used criteria 3 and 5 to evaluate the over all tube performance. Table (2.2) in Appendix - A shows the values of R_3 grouped by tube type.

2.2 ANALYTICAL STUDY.

In 1979, Patankar, Ivanovic and Sparrow (10) studied turbulent flow heat transfer in internally finned tubes and annuli. Their research was undertaken to analyze the turbulent flow and heat transfer characteristics of circular tubes and annuli with longitudinal internal fins. They established analytical model to obtain both average and local heat transfer results, as well as friction factors. The local results are of special interest because they convey information that would be extremly difficult to obtain experimentally. This includes inparticular the distribution of the heat loss along the height of the fin and the distribution of the tube wall heat loss around the circumference. Since the velocity varies along the fin height (.e. in the radial direction), the local fin heat loss would be expected to reflect this variation. In particular, if the highest velocity was to occur in the neighbourhood of the tip of the fin, the heat loss might be largest in that neighborhood.

They found that the heat transfer coefficient increases monotonically along the fin height and the extent of variation is substantial. In the neighborhood of the fin tip, the local coefficient are large (but **finite**) and in the range of 2 to 2.5 times the average,while near the base they are virtually zero. These results stand in sharp contrast to the standard model used in fin analysis where the heat transfer coefficient is assumed to be uniform. The tube wall heat transfer coefficient is virtually zero at the corner and increases monotonically along the circumference of the tube attaining maximum at the inter fin midpoint. Analytical predictions for the average Nusselt number and the friction factor were not generated by them, because emprical correlations suggested by Carnavos (9) were available.

Prakash and Ye-Di Liu (11) worked on Laminar flow heat transfer in the entrance region of an internally finned circular duct. They analyzed steady,laminer forced convection flow and heat transfer in the entrance region of finned tube by numerically integrating the governing partial differential equations.Results, exhibit the expected large pressure gradients and heat transfer coefficients in the entrance region,approaching,asymptotically, the fully developed values away from the entrance region.

CHAPTER - III

HEAT TRANSFER PARAMETERS

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

3.1 Thermal Boundary Conditions.

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

(1) Uniform heat input per unit axial length with uniform temperature at any cross section.

(2) Uniform temperature both axially and peripherlly.

In the present study, we considered uniform heat input per unit axial length.

3.2 Definition of Parameters.

Hydraulic Diameter :

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

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

$$= \frac{\pi D_i^{2}}{\pi D_i + 2NH} \qquad (3.1)$$

(Assuming thin fin)

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

$$D_h = \frac{4\ A_{xf}}{\pi D_i + 2NH} \tag{3.2}$$

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

Reynolds Number :

Reynolds number based on inside diameter is defined as

$$Re_i = \frac{\rho V D_i}{\mu} \tag{3.3}$$

Reynolds number based on hydraulic diameter is defined as

$$Re_h = \frac{\rho V\ D_h}{\mu} \tag{3.4}$$

Pressure Drop and Fanning Friction Factor:

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

$$P^*(x) = -\Delta P\ (x)\ /\ \tfrac{1}{2}\rho V^2 \tag{3.5}$$

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

P_i = pressure at inlet

$P(x)$ = pressure at some axial location, x

The local friction factor based on hydraulic diameter is defined as

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

The local friction factor based on inside diameter is defined as

$$F_i = \frac{(-\Delta P / x)\, D_i}{2 \rho V^2} \tag{3.7}$$

Heat Transfer: Uniform Heat input per unit axial length:

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

$$T_b(x) = T_i + \frac{Q' x}{M C_p} \tag{3.8}$$

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

$$h_x = \frac{Q' \text{[illegible]}}{(T_w - T_b)_x A_{h\text{[illegible]}}} \tag{3.9}$$

Where $A_{h\text{[illegible]}} = (\pi D_i + 2NH)\ \text{[illegible]}$ (3.10)

The local Nusselt number can be defined as

$$Nu_x = \frac{h_x D_h}{k}$$

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

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,

$$\frac{(T_w - T_b)_{F.D}}{(T_w - T_b)_x}$$

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 cordinates (X / L). The entrance length L_t may be defined as the length required for

$$\frac{(T_w - T_b)_{F.D}}{(T_w - T_b)_x} = 1.05$$

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 upto 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

$$h = \frac{Q'}{A_h \, (T_w - T_b)_{F.D}} \qquad (3.12)$$

$$\text{Where } A_h = (\pi D_i + 2NH) \qquad (3.13)$$

$$Q' = M \, C_p \, (T_i - T_o) \, / \, L \qquad (3.14)$$

Nusselt number for thermally developed regime is given by

$$Nu = \frac{h \, D_h}{k} \qquad (3.15)$$

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 (heat transfer) area, A of an 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,

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

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

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 directly reported in many works and which is really a most important parameter.

Criterion 3 :

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

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

Pumping power can be defined as

$$P_m = (- \Delta P/\rho) M \tag{3.18}$$

$$= \frac{4 F_i L}{D_i} \frac{V^2}{2} A_x V \rho \tag{3.19}$$

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

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

Therefore,

$$A_{xo} F_{io} V_o^3 = A_{xf} F_{if} V_f^3 \tag{3.20}$$

$$\left(\frac{V_o}{V_f} \right)^3 = \frac{A_{xf} F_{if}}{A_{xo} F_{io}} \tag{3.21}$$

Reynolds number based on inside diameter for smooth tube is

$$Re_o = \frac{\rho V_o \; D_i}{\mu}$$

Reynolds number based on inside diameter for finned tube is

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

Therefore,

$$\frac{Re_o}{Re_{if}} = \frac{V_o}{V_f} \qquad (3.22)$$

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

$$A_{xo} \; F_{io} \; Re_o^3 = A_{xf} \; F_{if} \; Re_{if}^3 \qquad (3.23)$$

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

Blasius relation :

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

The expression (3.23) becomes,

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

Therefore,

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

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 calcualte the constant pumping power criteria, R_3 form the equation,

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

C H A P T E R - 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 longitudinal fins. The schematic diagram is shown in the figures (4.1 and 4.2). Air was used as the working fluid encompassing flow range of $10^4 < Re < 10^5$. The test section air was supplied by a centrifugal fan fitted at the end of the set up. It was driven by a 3 phase, 3 hp(440 V,4.1 A) motor. The maximum flow rate of the fan under free operation was approximately 30 m^3/min. A flow control valve was installed after the test section to control the air flow rate during the experiment. The set up consisted of (1) Inlet and flow measuring section (2) Heat transfer test section (3) Fan assembly.

4.1 Inlet Section :

The unheated inlet section (shaped Inlet) and heat transfer test section of same diameter (70 mm inside diameter) were cast from aluminum. 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 by fitting a shaped Inlet. The pipe and the shaped inlet, 530 mm long,(Fig 4.3B) were made integral to avoid any flow disturbances at upstream of the section of flow measurement. The co-ordinates of the curvature of the shaped inlet was suggested by Ower and Pankhurst (15) which is reproduced in Table (4.1),Appendix - B. The shaped inlet was fitted in the open end of the test section.

4.2 Test Section :

The internal fins of test section,(1520 mm long and 70 mm inside diameter) were made integral with circular tube to avoid contact resistance. The fin and tube assembly was cast from aluminum, because of its high thermal conductivity and easy machinability. Two halves of circular tube with integral fins were cast seperately and joined together to give the shape of a circular tube. Fig (4.3A) depicts the tube configuration and dimensions. As the two halves were joined at the line of symmetry, it did not effect thermal and hydraudynamic results of the experiment. The test section was wrapped with mica-sheet and insulation tape. Over the mica sheet Nichrome wire (of resistance 0.610 ohm/m) was spirally wound uniformly with spacing of 8 mm between the turns. The nichrome wire was covered with mica sheet and insulation tape to make it electrically insulated by covering with asbestos. The test sections were joined by bolted flanges, between which asbestos sheet were installed. The asbestos sheet between the flanges provided thermal insulation for the heat transfer section.

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

The electrical power to the test section was determined by measuring the current and voltage supplied to the heating element. The voltage was measured with a voltmeter and current was measured by an a.c.ammeter. Fig (4.9) shows the electrical circuit diagram of the heating system of the test section. The particulars of electric heater,temperature controller and fan are given in Table (4.2) in Appendix - B.

4.3 Fan Assembly :

A diffuser of cone angle 12° was made of 1/16 inch M.S plate and fitted to the suction side of the fan. The diffuser was used for minimizing head loss at the suction side. To arrest vibration of the fan a flexible duct was installed between the inlet section of the fan and 3 inch diameter pipe of the set up as shown in Fig (4.2). A flow control valve was installed at the suction side before flexible duct to control the air flow rate during experiment. Flow control valve is of butterfly type.

4.4 Flow Measurement by Traversing Pitot :

Flow of air through the experimental set up was measured at inlet section with the help of Traversing Pitot. A shaped inlet made of aluminum was installed at the inlet to the test section to have an easy entry and symmetrical flow. At 4 pipe diameters, according to Ower and Pankhurst (15),from inlet a traversing pitot was installed. A drawing of the Micro meter Traversing Pitot is given in Fig (4.6). The velocity was determined by means of a pitot tube in conjunction with a static pressure hole in the wall of the pipe. The pitot was traversed,along the diameter, in a plane about 1 pipe diameter down stream of the side hole so as not to disturb the static pressure readings. The difference between the static pressures at the two pipe sections one pipe diameter apart would not, as a rule,be large enough in reasonably smooth pipes.

Mean velocity was measured by log linear method by traversing pitot tube along the diameter of the pipe at ten measuring points. Table (4.3) in Appendix - B shows location of measuring points for the log linear method. The ten points log linear traverse, which has been specified by the B.S.I. for class A accuracy resulted in a mean square error of about 0.5 per cent. Initial calibration at the shaped inlet was done by traversing pitot tube and mean velocity was measured by graphical integration (Fig 5.2. 1-5.2.4). The curve V/V_c against log $V_c D$ / is shown in Fig (5.3). From Fig (5.3) mean velocity can be determined with a single measurement of the velocity at the axis of the pipe. Mean velocity, measurement by ten points log linear method, differs very less from that measured by graphical integration. Table 5.1 shows per centage of deviation in log linear method over graphical integration method.

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 locations of the test section as shown in Fig (4.5). Pipe wall pressure tappings (Fig 4.8) 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 (Araldide) was used for proper fixing of the static pressure tappings.

4.6 Measurement of Tamperatures :

The temperatures were measured with the help of thermocouples at the following locations :

(1) Fluid bulk temperature at inlet and outlet of the test section.

(2) Wall temperature at 10 axial locations of the test section.

(3) Fin-tip temperature at ten axial locations of the test section.

The bulk temperature of the air entering the test section was measured using thermocouple situated in the air stream just up stream of the test section inlet. The bulk temperature of the air at the outlet of the test section was measured using three thermocouples situated at three different radius at the outlet of the test section.

20 thermocouples were installed in 10 cross sections (Fig 4.4) with two in each cross section to measure the wall as well as fin-tip temperatures of the test section. The thermocouples were made from copper constantan wires. All the thermocouples were calibrated before installation. Thermal contact between the aluminum tube and the thermocouple junction was assured by peening thermocouples junction into grooves in the wall. 1/16 inch holes were drilled across the height of the fin in 10 cross sections to measure the temperature of the fin-tip. Thermocouples were inserted through the holes and peened into the grooves of the fin-tip.

Temperatures were measured by a multipoints system of thermocouples connected through ice bath and selector switches to microvoltmeter.

4.7 Procedure of Experiment :

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

Inclined tube manometer was chosen on the score of simplicity for indicating velocity head. Water was selected as the manometric liquid. All the pressure tappings for measuring static pressure were connected to the U-tube manometers. The readings of velocity head were taken by traversing pitot tube along the diameter of the pipe. Then the readings of static pressure were taken.

The electrical current was adjusted with the help of Regulating Transformer (or Variac) to attain steady state condition for a particular Reynolds number.

Steady state was defined according to D. L. Gee and R. L. Webb (16) 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 was established if the outlet air temperature did not deviate over a 10-15 minutes period. All the thermocouples readings were taken at steady state condition.

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

C H A P T E R – V

RESULTS AND DISCUSSIONS

This chapter presents the results obtained form experimental data along with discussions on results. Friction factor and heat transfer results have been shown in the entrance region as well as in the fully developed region. Heat transfer performance expressed in terms of the Nusselt number has been compared at both constant Reynolds number and at constant pumping power.

The dimensionless pressure drop, P^* along the length of the finned tube has been shown in Fig (5.4). The pressure gradient is high at 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 drop measured from 260 mm down stream of the inlet plane. Fig (5.5) shows the friction factor based on inside diameter and Fig (5.6) based on hydraulic diameter. Friction factor for finned tube is higher than that of unfinned tube, but slope of the friction factor of finned tube is very nearly equal to that of unfinned tube(smooth tube). Friction factor based on hydraulic diameter of finned tube increases over that of smooth tube by 48.15% to 100% for Reynolds number range from 1.56×10^4 to 4.42×10^4, shown in Table (5.2). Friction factor based on inside diameter of finned tube increases by 220% to 351.6% over that of smooth tube for Reynolds number range 2.66×10^4 to 7.86×10^4, shown in Table (5.3) in Appendix – C.

Pressure drop ΔP was measured for axial distance Δx at different axial location x, and from the above measurement friction factor was calculated to see the distribution of friction factor along the length of the finned tube shown in Fig (5.7). Fig (5.7) shows that the friction factor is higher near the entrance due to large pressure drop, then falls abruptly to the minimum value, after then rises a little before gradual falling off to fully developed value. The friction factor curve shows peculiar trend due to the effect of internal fins which need further investigation.

Fig (5.8-5.9) shows the wall and bulk temperature distribution along the length of the tube. The wall temperatures were determined by direct measurement as described earlier (in chapter-IV), while the bulk temperature distribution was obtained by calculation. At higher Reynolds number, wall temperature is low, because more heat is taken away by the air. Fig (5.8-5.9) shows that there is a portion of the test section where the wall and bulk temperature distribution are parallel, yielding a uniform value of $(T_w - T_b)$ as expected for constant heat rate. Just up stream of the exit, the slope of the wall temperature gradually falls due to end effect.

Fig (5.10) shows distribution of wall to bulk temperature difference along axial distance for different Reynolds number. The wall to bulk temperature difference first increases, then become axially unchanging in the fully developed regime. Near the exit, it decreases due to end effect.

Fig (5.11) shows distribution of temperature ratio $(T_w-T_b)_{F.D}/(T_w-T_b)_x$ along the length of finned tube, for different Reynolds number ratio $Nu_x/Nu_{F.D}$ along the length of the finned tube. Starting with a maximum value at the inlet, the distribution drop off with increasing down stream distance rapidly at first and then more gradually. At a sufficient distance from the inlet the distribution levels off, signaling the attainment of the thermally developed regime. Just upstream of the exit, the lift off of the data reflects an end effect.

Fig (5.12-5.16) shows variation of local Nusselt numbers along the length of the finned tube. Nusselt number is large in the entrance region. It decreases with increasing axial distance approaching the fully developed values. Fig (5.13-5.16) shows that at higher Reynolds number, the curve of local Nusselt number along the length of the tube is higher.

Fig (5.17) shows the variation of local Nusselt number of finned tube for Reynolds number 4.42×10^4 and that of smooth tube for Reynolds number 5×10^4. The data for smooth tube have been obtained from the smooth tube correlation for entrance region. The details of calculation of local Nusselt number for smooth tube is given in Appendix - E.

The thermal entrance length of finned tube, defined as the length required for $Nu_x/Nu_{F.D} = 1.05$, has been calculated from Fig (5.15). The thermal entrance length is 5.2d for Reynolds number 1.56×10^4 and 6.25 d for Reynolds number 4.24×10^4. The thermal entrance length for different Reynolds number, obtained from Fig (5.15) are shown in Table (5.4) in Appendix - C. The results exhibit the thermal entrance length increases with Reynolds number. Fig (5.17) shows that the thermal entrance length for smooth tube is 8.3 diameter for Reynolds number 5×10^4 and that of finned tube is 6.25 diameter for Reynolds number 4.42×10^4. The attainment of fully developed region is earlier in finned tube than that in smooth tube, because of enhancement of heat transfer in finned tube due to fins. But the curve of smooth tube in the fully developed region is higher than that of finned tube because of higher Reynolds number.

Fully developed Nusselt number based on hydraulic diameter and effective (heat transfer) area is shown in Fig (5.19). The slope of heat transfer data of finned tube is very nearly equal to that of smooth tube. The variation of heat transfer results from Carnavos correlation are 6.27% to 14.98% for Reynolds number range from 1.56×10^4 to 4.42×10^4, shown in Table (5.10).

Heat transfer results based on inside diameter and nominal area are shown in Table 5.5 and 5.6 in Appendix - C. Based on inside diameter and nominal area, Nusselt number for the finned tube exceeded smooth tube values by 97% to 112.2% for Reynolds number range from 2.66×10^4 to 7.86×10^4. The heat transfer coefficients for finned tube were in the range 1.97 to 2.12 times smooth tube values. Fig (5.20) shows the heat transfer results of finned tube based on inside diameter and nominal area as a function of Reynolds number and comparison with that of smooth tube. The heat transfer curve for finned tube is straight line but much above the smooth tube curve showing enhancement of heat transfer due to fin effect.

When compared with a smooth tube at constant pumping power and constant basic geometry, an improvement as high as 52% was obtained in heat capacity. Constant pumping power performance ratio, R_3 is shown as a function of Re_o in Fig (5.21). $R_3 = h_{if}/h_{op}$ are in the range of 1.36 to 1.52 for Reynolds number, Re_o from 4.49×10^4 to 1.19×10^5. Table (5.8) in Appendix - C shows the comparison with previous work. This work result is close to that of tube no 9 of Watkinson et al (3) and show difference of 6.5% to 13% only. Table (5.9) shows heat transfer performance ratio at constant Reynolds number and at constant pumping power.

C H A P T E R - VI

CONCLUSIONS

Steady state turbulent flow heat transfer performance of circular tube having integral internal fins was experimentally studied. The experimental study 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 enumerated below :

(1) Nusselt number is large in the entrance region. It decreases with increasing axial distance approaching asymptotically, the fully developed values.

(2) For fully developed region, the slope of $\frac{Nu}{Pr^{0.4}}$ vs Re_h curve of finned tube is very nearly equal to that of smooth tube.

(3) The heat transfer results are very close to Carnavos correlation for finned tube and per centage of deviations are in the range of 6.27% to 14.98%.

(4) Based on inside diameter and nominal area, heat transfer for the finned tube exceeded smooth tube values by 97% to 112% for Reynolds number range from 2.66×10^4 to 7.86×10^4.

(5) The heat transfer coefficient based on inside diameter and nominal area were in the range of 1.98 to 2.12 times smooth tube values.

(6) When compared with a smooth tube at constant pumping power and constant basic geometry of the tube an improvement as high as 52% was obtained in heat capacity.

Criterion $R_3 = h_{if}/h_{op}$ are in the range of 1.36 to 1.52 for Reynolds number, Re_o from 4.49×10^4 to 1.19×10^5.

(7) The results exhibit the thermal entrance length increases with Reynolds number. The thermal entrance lengths were 6.25 d for 4.42×10^4 Reynolds number and 5.2 d for 1.56×10^4 Reynolds number.

(8) The attainment of fully developed region is earlier in finned tube that in smooth tube.

(9) The results exhibit that the local friction factor is high near the inlet, drops suddenly to a minimum value,then rises a little before gradual falling off with axial distance,appr-oaching the fully developed value.

(10) Friction factor of finned tube based on hydraulic diameter is higher than that of smooth tube and in the range of 1.5 to 2 times the smooth tube values. But the slope of the friction factor of finned tube is very nearly equal to that of smooth tube.

(11) Friction factor based on inside diameter for finned tube is in the range of 3.2 to 4.5 times the smooth tube values.

Extension of the Present Work :

With some modifications of the experimental set up, the test section can be replaced by another one of varying tube diameter, tube length, number of fins, fin height etc. and more data can be obtained to analyse heat transfer performance of tube having internal fins.

FIGURES

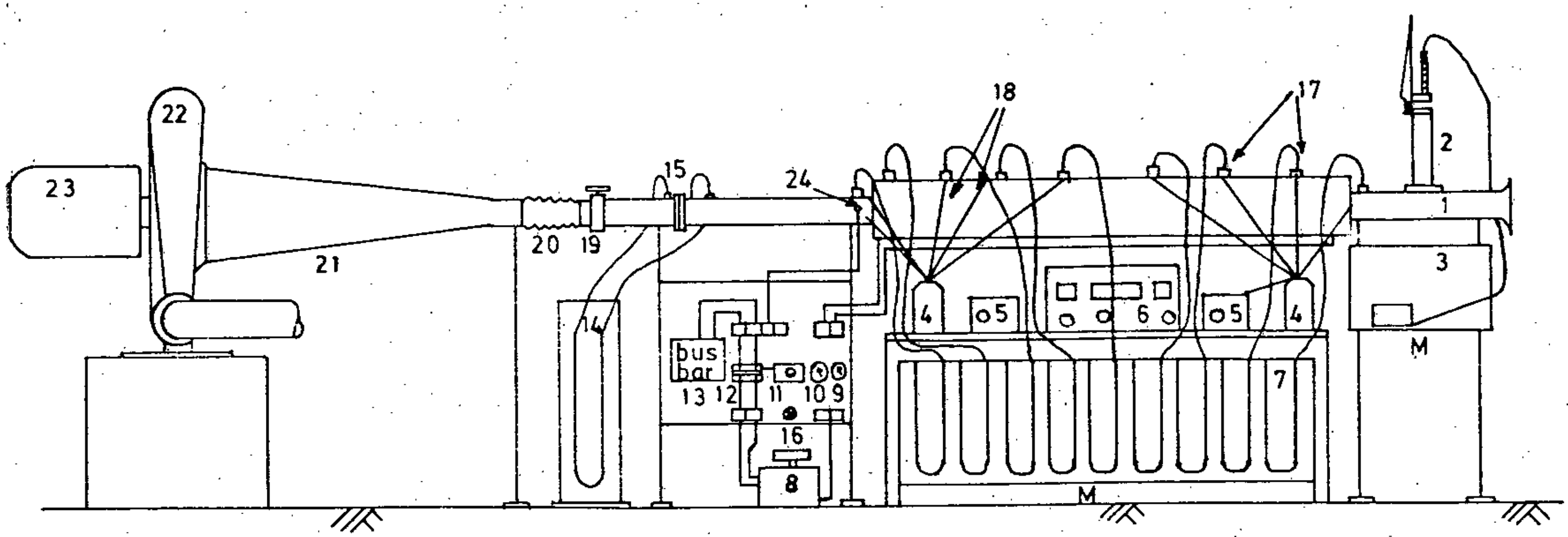
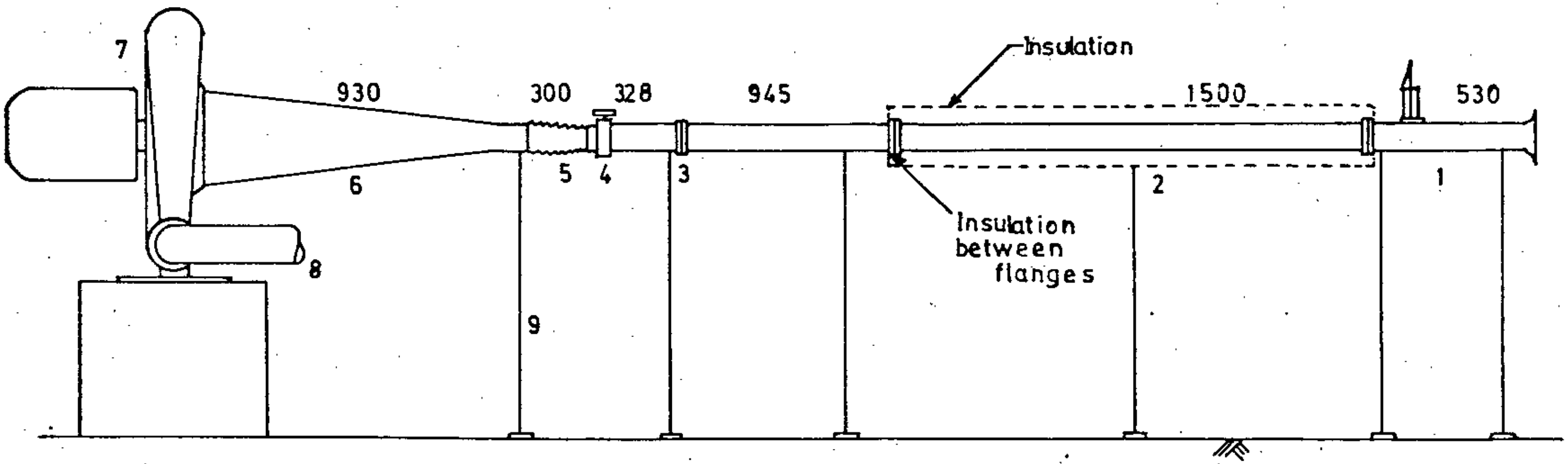


FIG: 4·1 SCHEMATIC DIAGRAM OF EXPERIMENTAL SET UP

Experimental Set-up

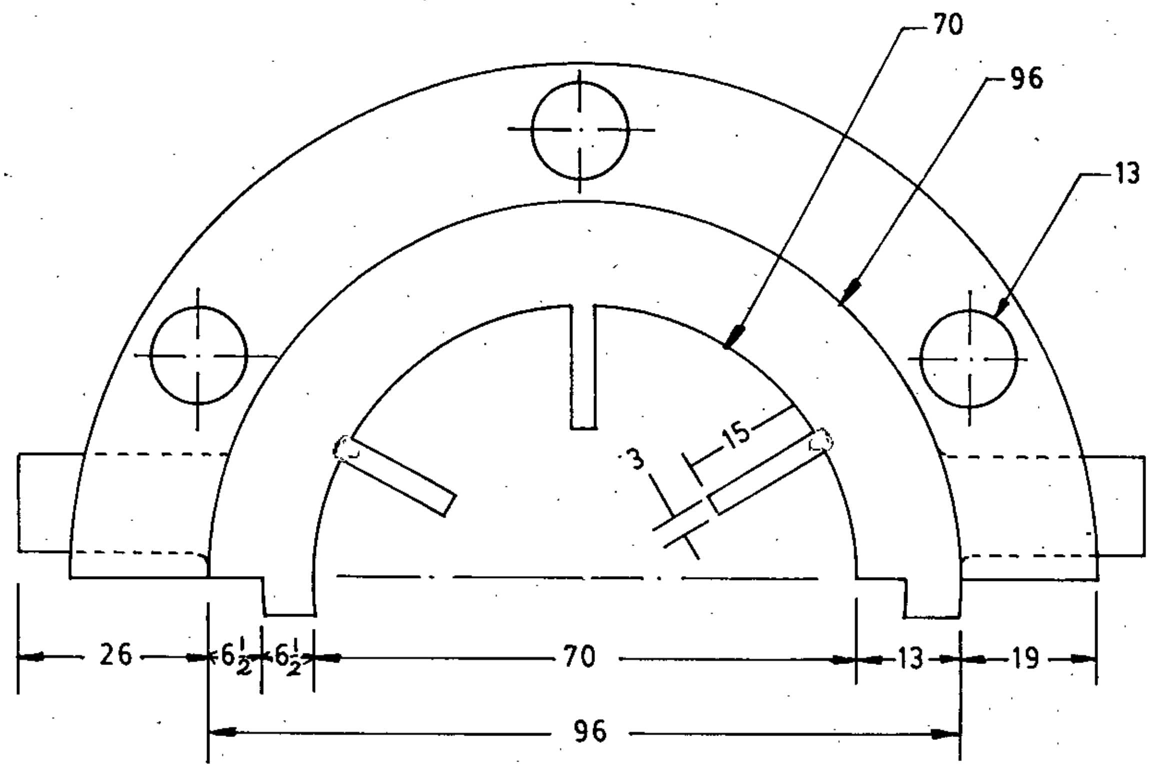
1. Shaped inlet
2. Traversing pitot
3. Inclined tube manometer
4. Ice bath
5. Selector switch
6. Microvoltmeter (DVM)
7. U-tube manometers
8. Variable voltage transformer
9. Ammeter
10. Voltmeter
11. Temperature controller
12. Magnetic contactor
13. Bus bar
14. U-tube manometer
15. Orifice meter
16. Heater on off lamp
17. Pressure tappings
18. Thermocouples
19. Flow control valve
20. Flexible pipe
21. Diffuser
22. Fan
23. Motor



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

All dimensions in mm.

FIG 4·2 SCHEMATIC DIAGRAM OF EXPERIMENTAL SET UP



All dimensions are in mm

FIG. 4.3A HALF OF THE CIRCULAR TUBE WITH FINS

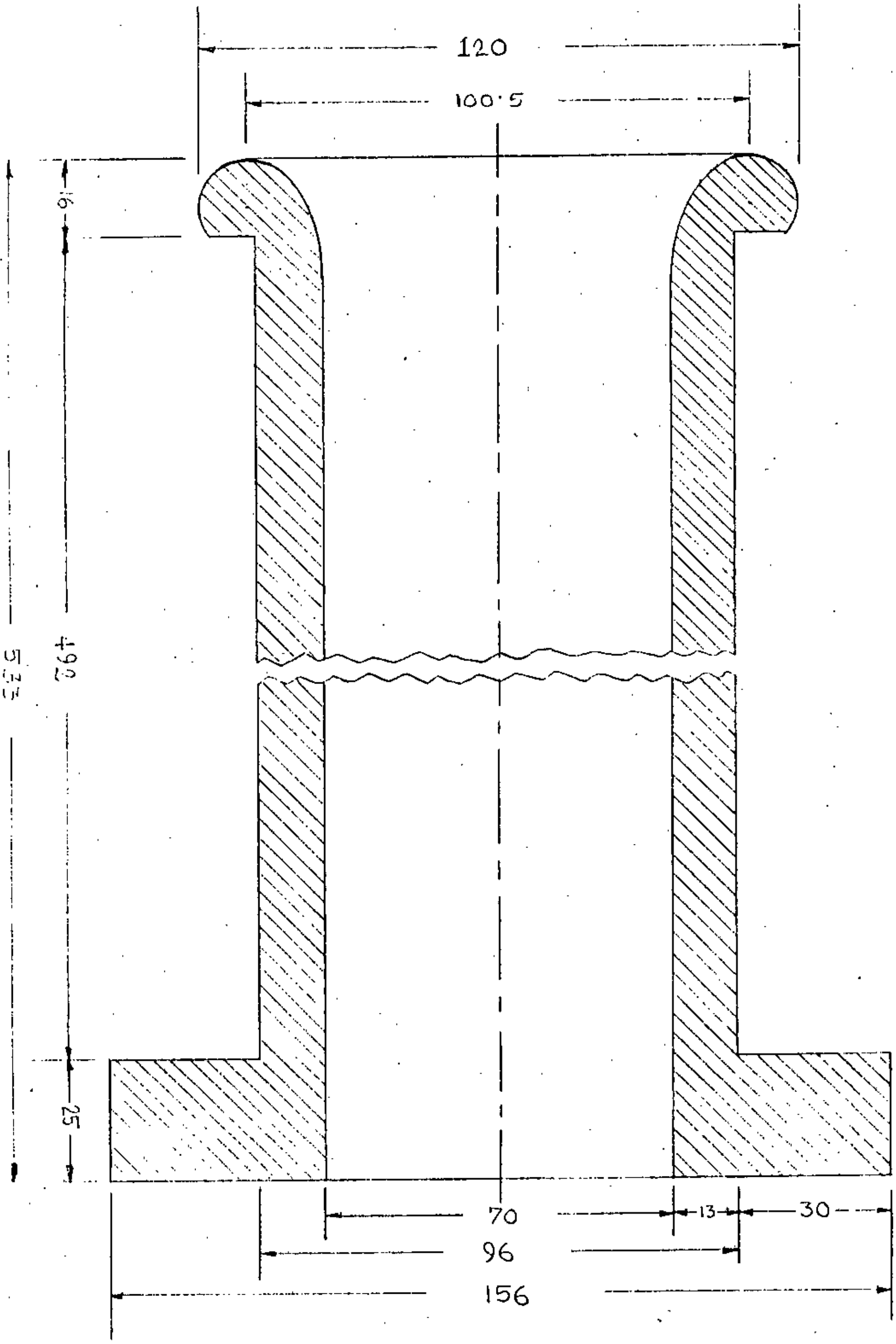
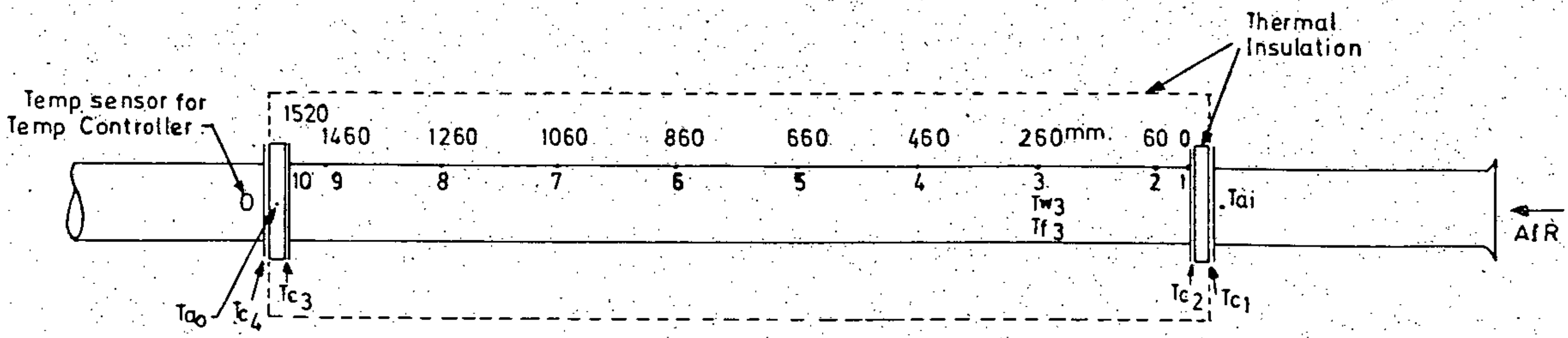


FIG. 4.3B SHAPED INLET



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

Tai ~ Air inlet temp
Tao ~ Air outlet temp
Tc ~ Temp at the flange
Tw_3 ~ Wall temp at 3
Tf3 ~ Fin-tip temp at 3

FIG: 4.4 LOCATION OF THERMOCOUPLES

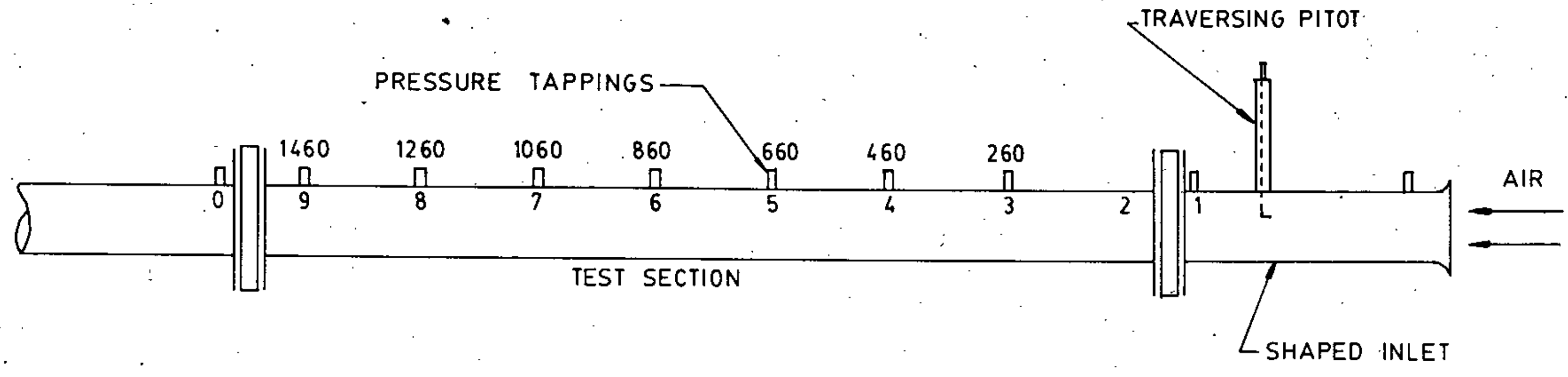
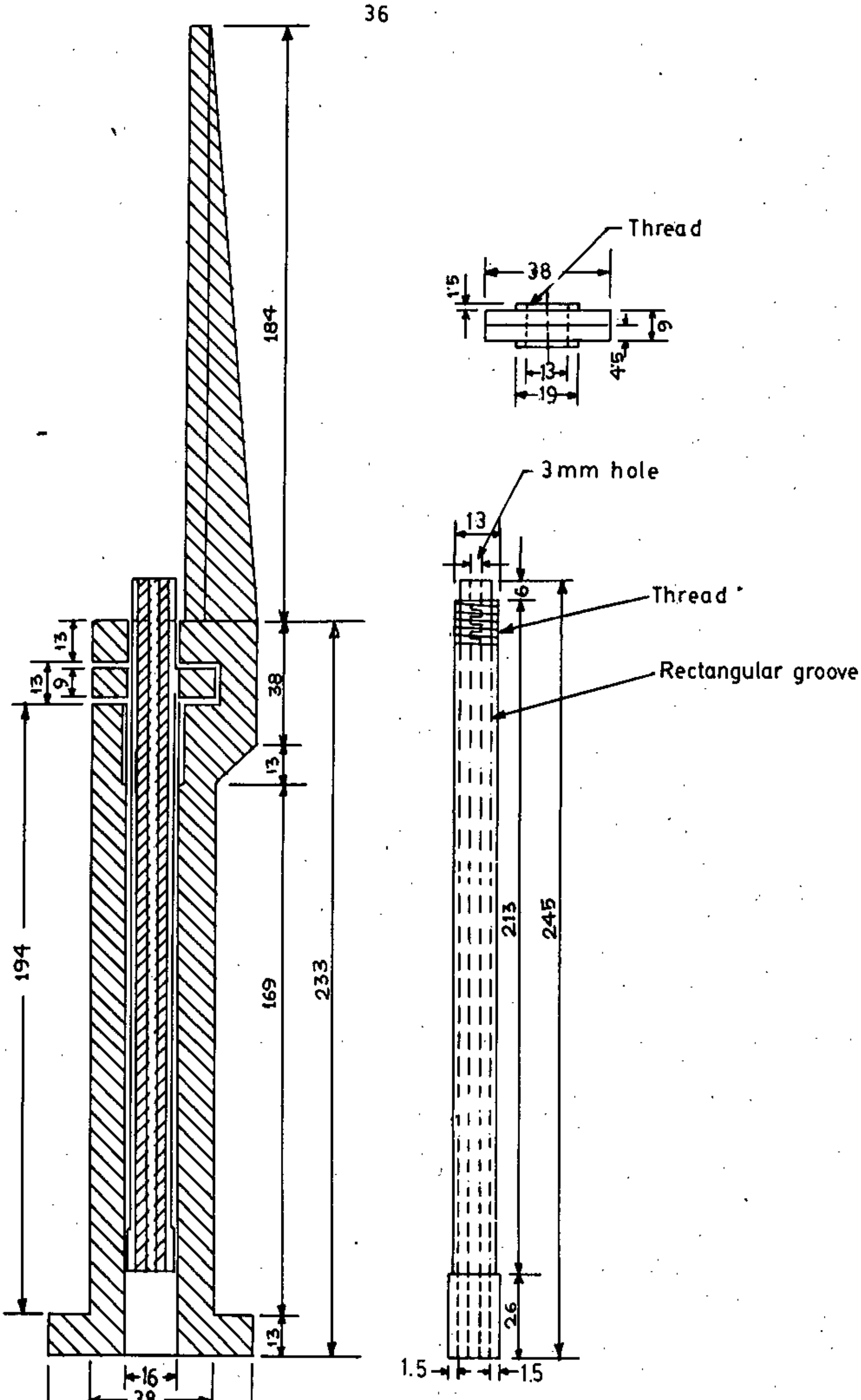


FIG. 4.5 LOCATION OF PRESSURE TAPPING



All dimension are in mm

FIG. 4.6 TRAVERSING PITOT

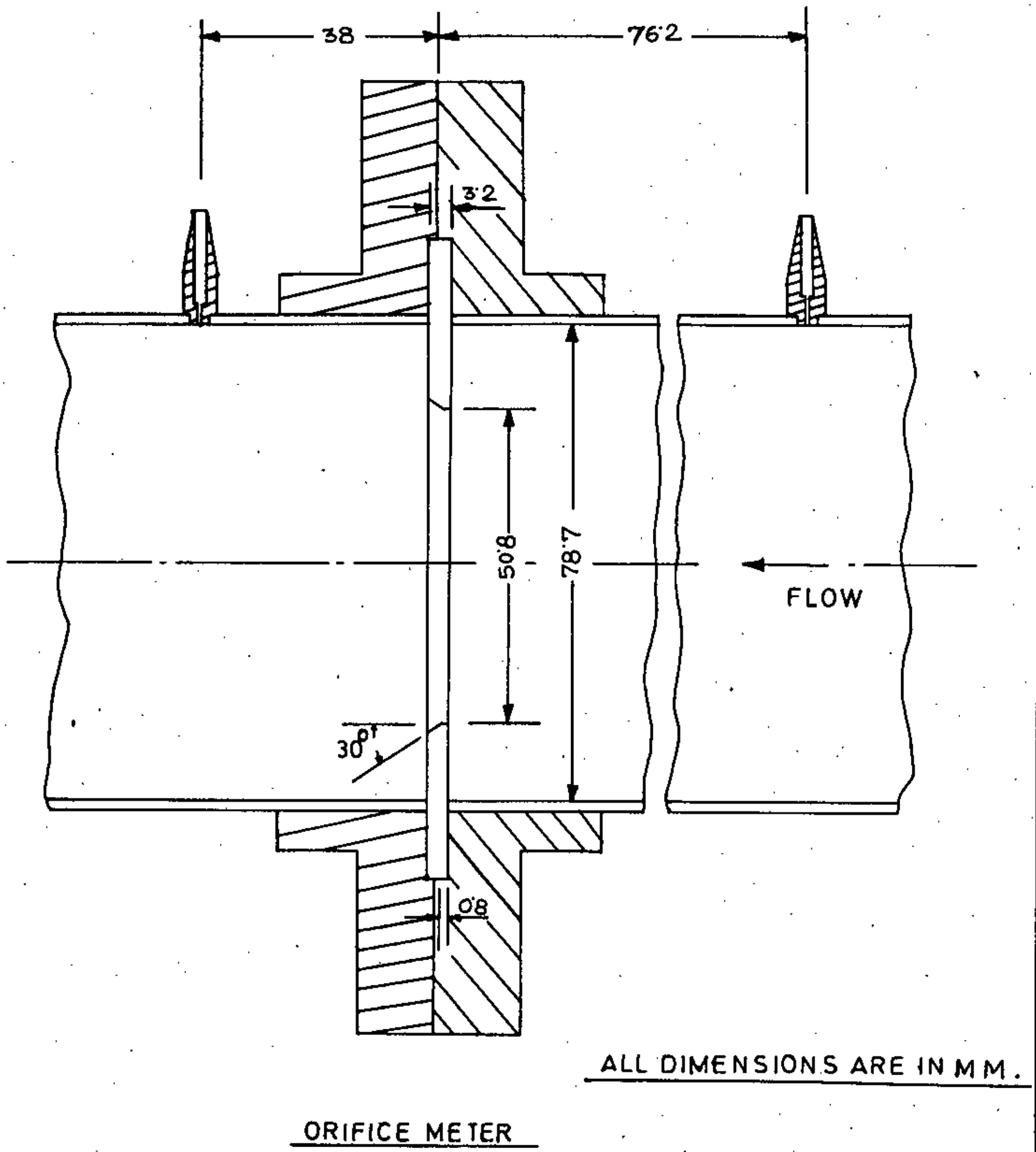
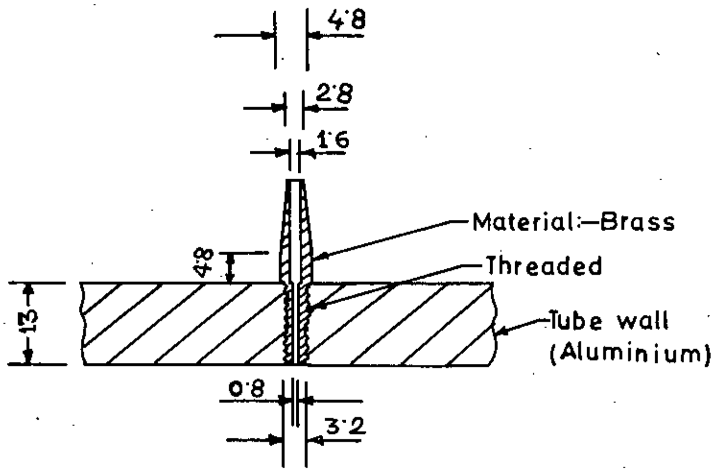
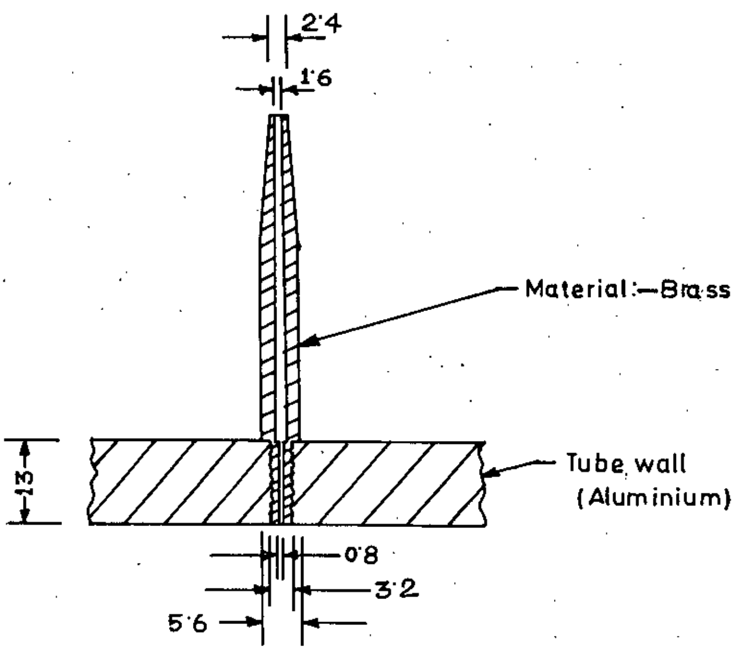


FIG: 4·7 ORIFICE METER



PRESSURE TAPPINGS AT SHAPED INLET



PRESSURE TAPPINGS AT THE TEST SECTION

ALL DIMENSIONS ARE IN MM.

FIG: 4·8 PRESSUQE TAPPINGS

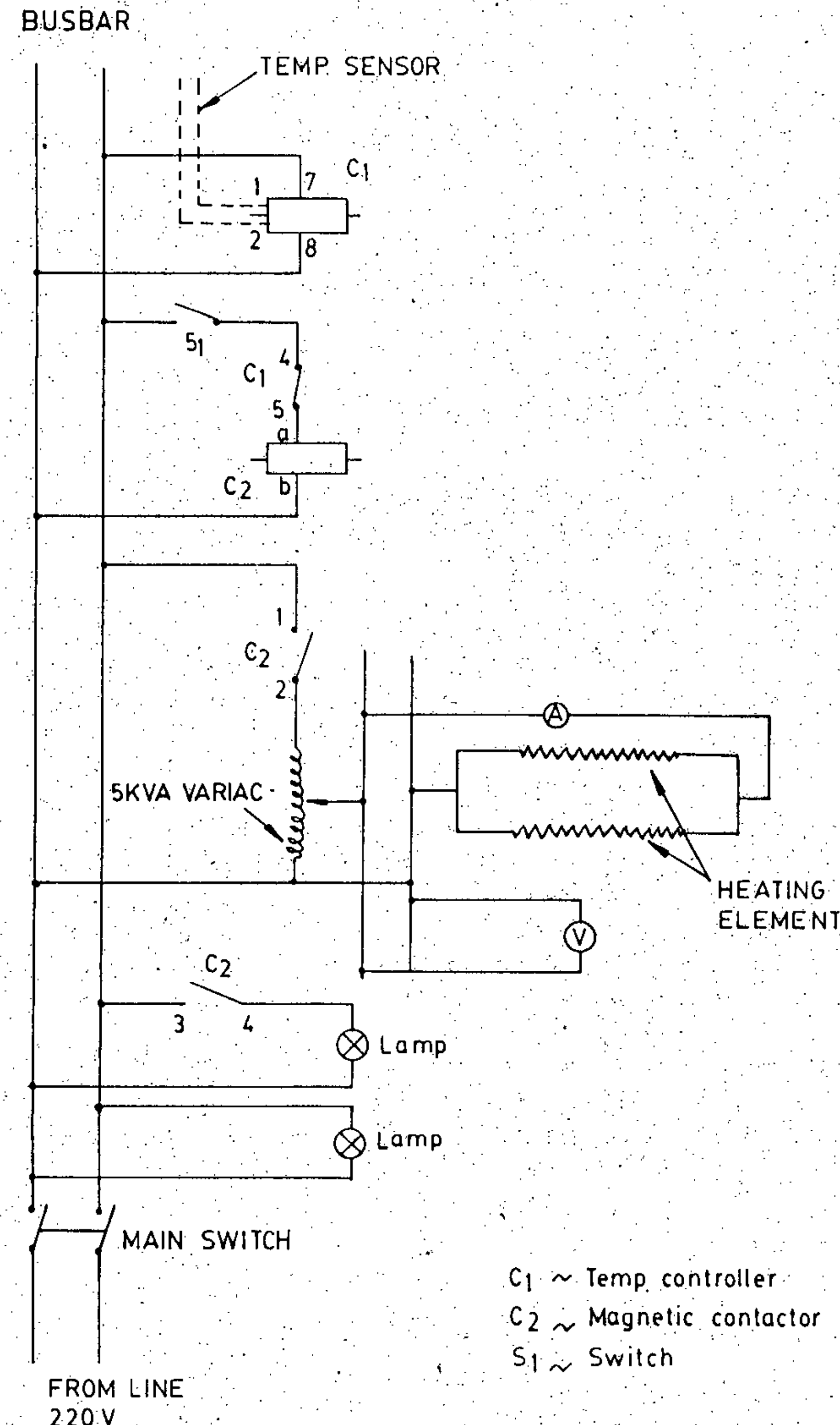


FIG. 4·9 ELECTRIC CIRCUIT DIAGRAM FOR HEATING SYSTEM

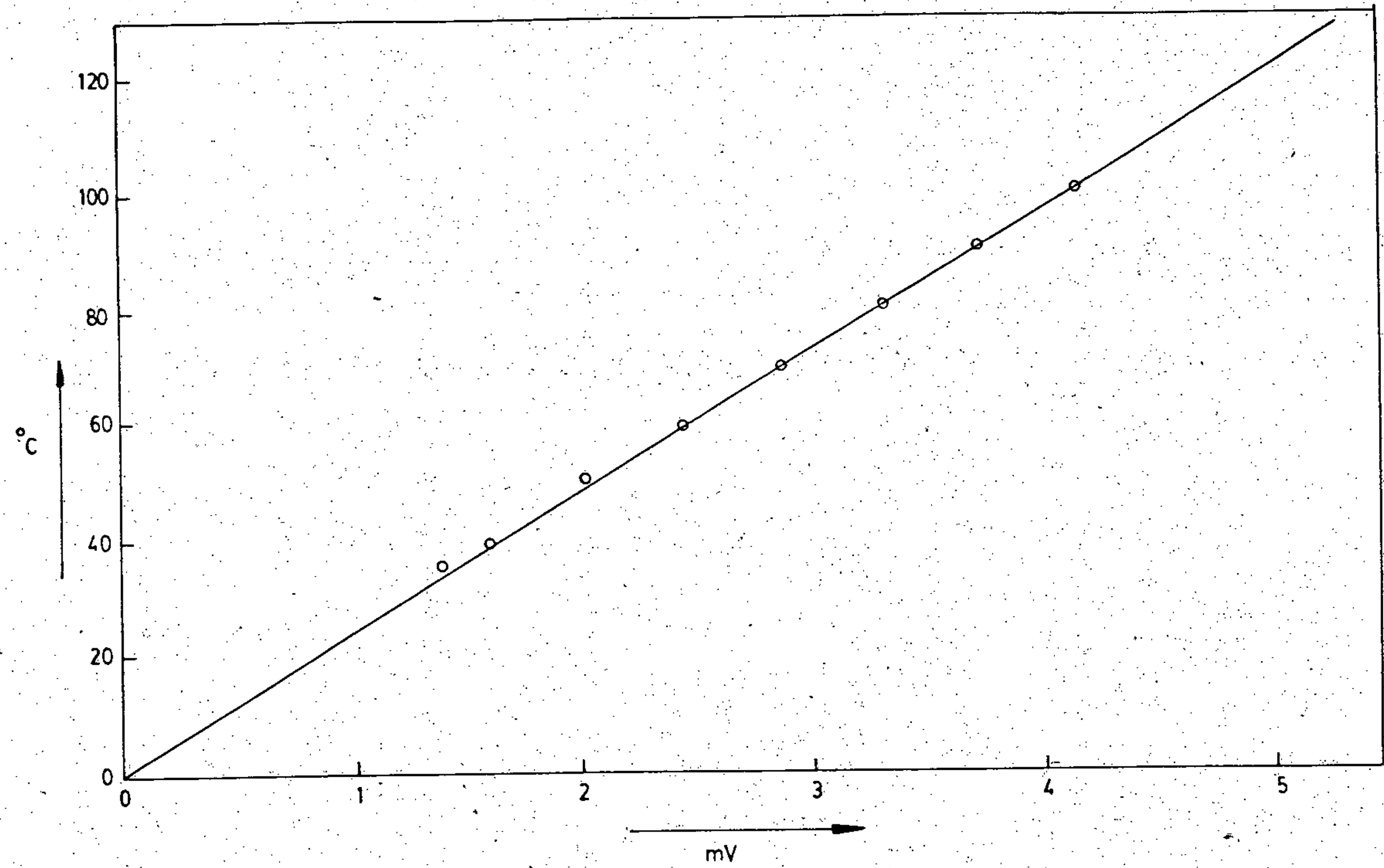


FIG. 5.1 CALIBRATION CURVE OF THERMOCOUPLE

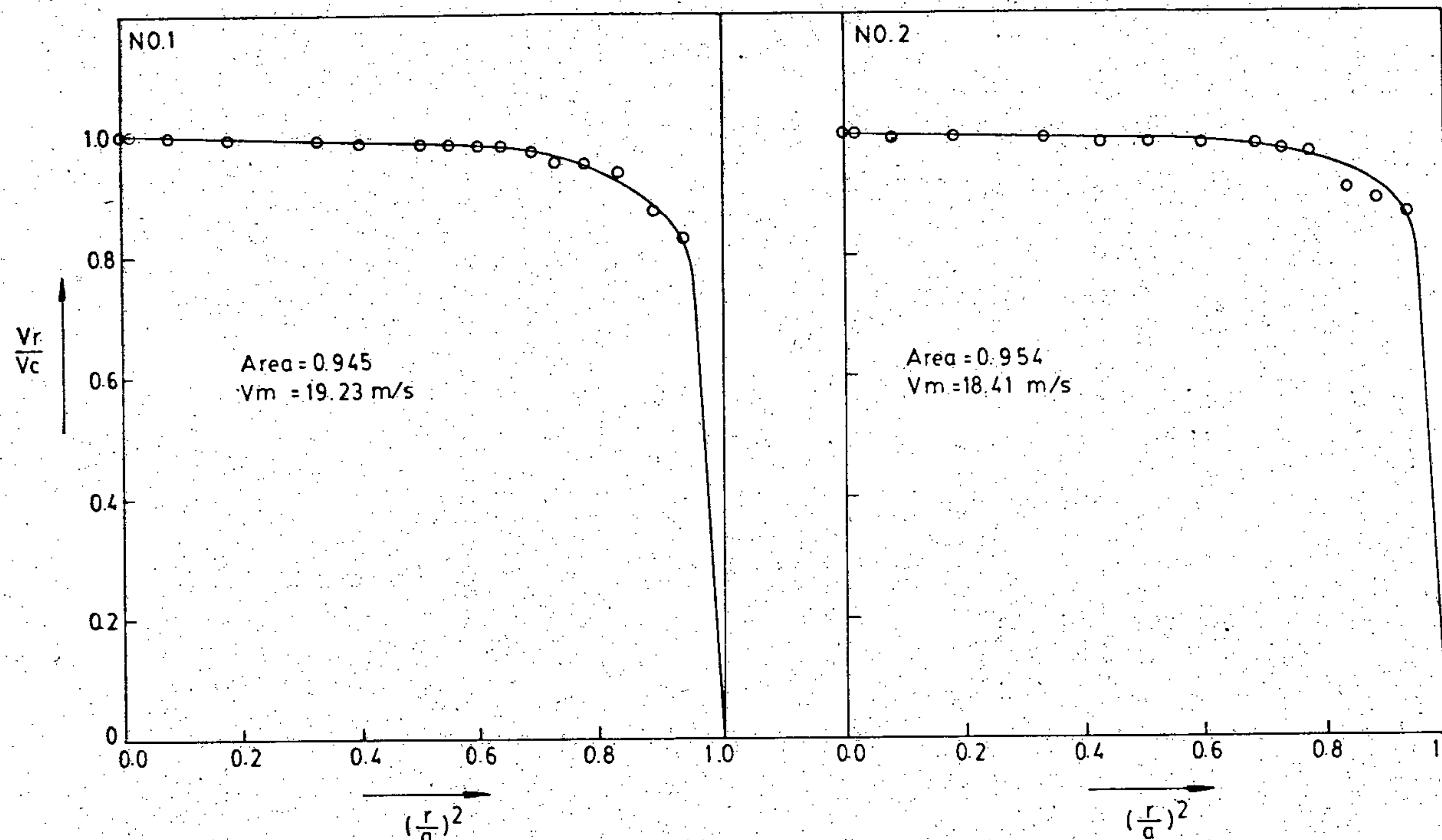


FIG. 5.2.1 CALCULATION OF MEAN VELOCITY BY GRAPHICAL INTEGRATION

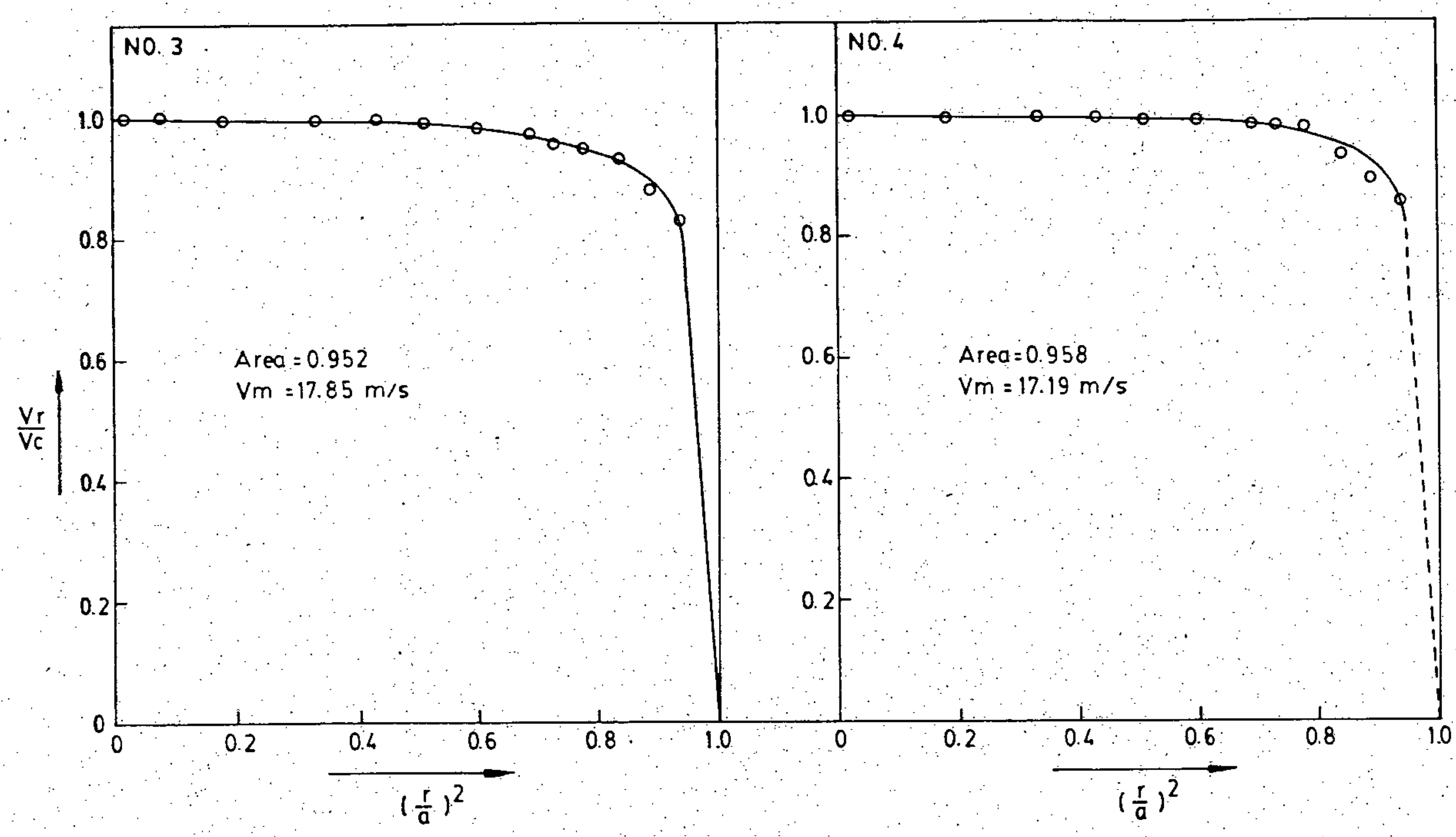


FIG. 5.2.2 CALCULATION OF MEAN VELOCITY BY GRAPHICAL INTEGRATION

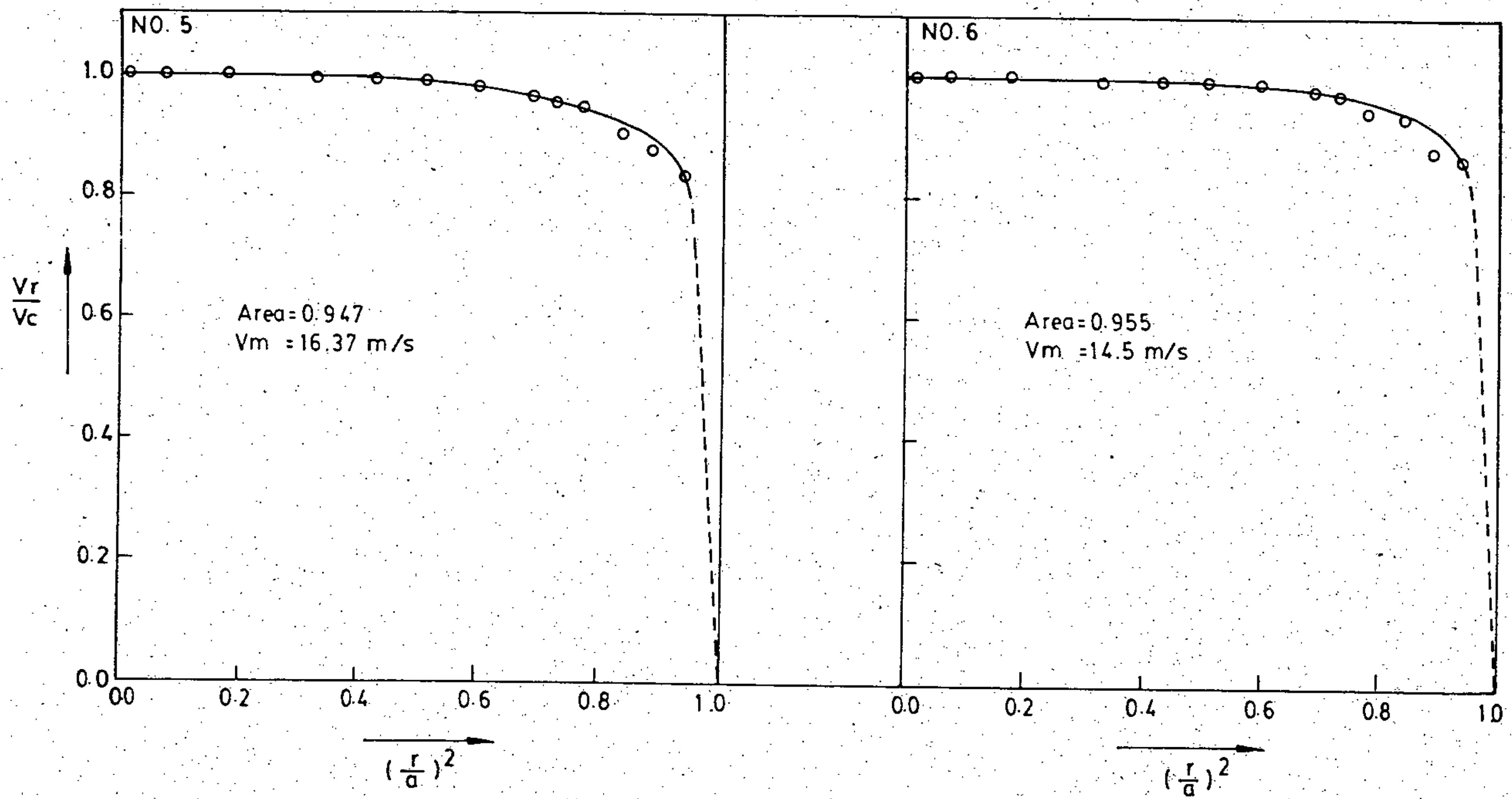


FIG. 5.2.3 CALCULATION OF MEAN VELOCITY BY GRAPHICAL INTEGRATION

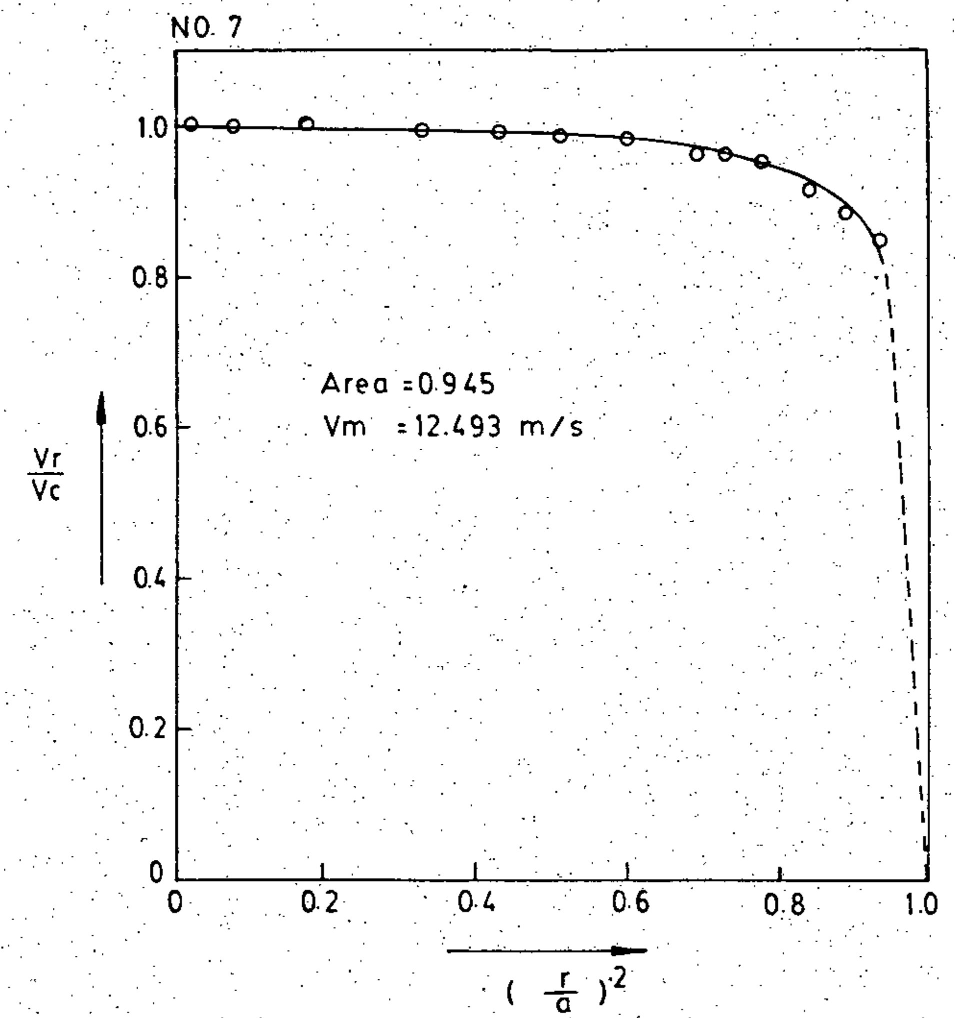


FIG. 5.2.4 CALCULATION OF MEAN VELOCITY BY GRAPHICAL INTEGRATION

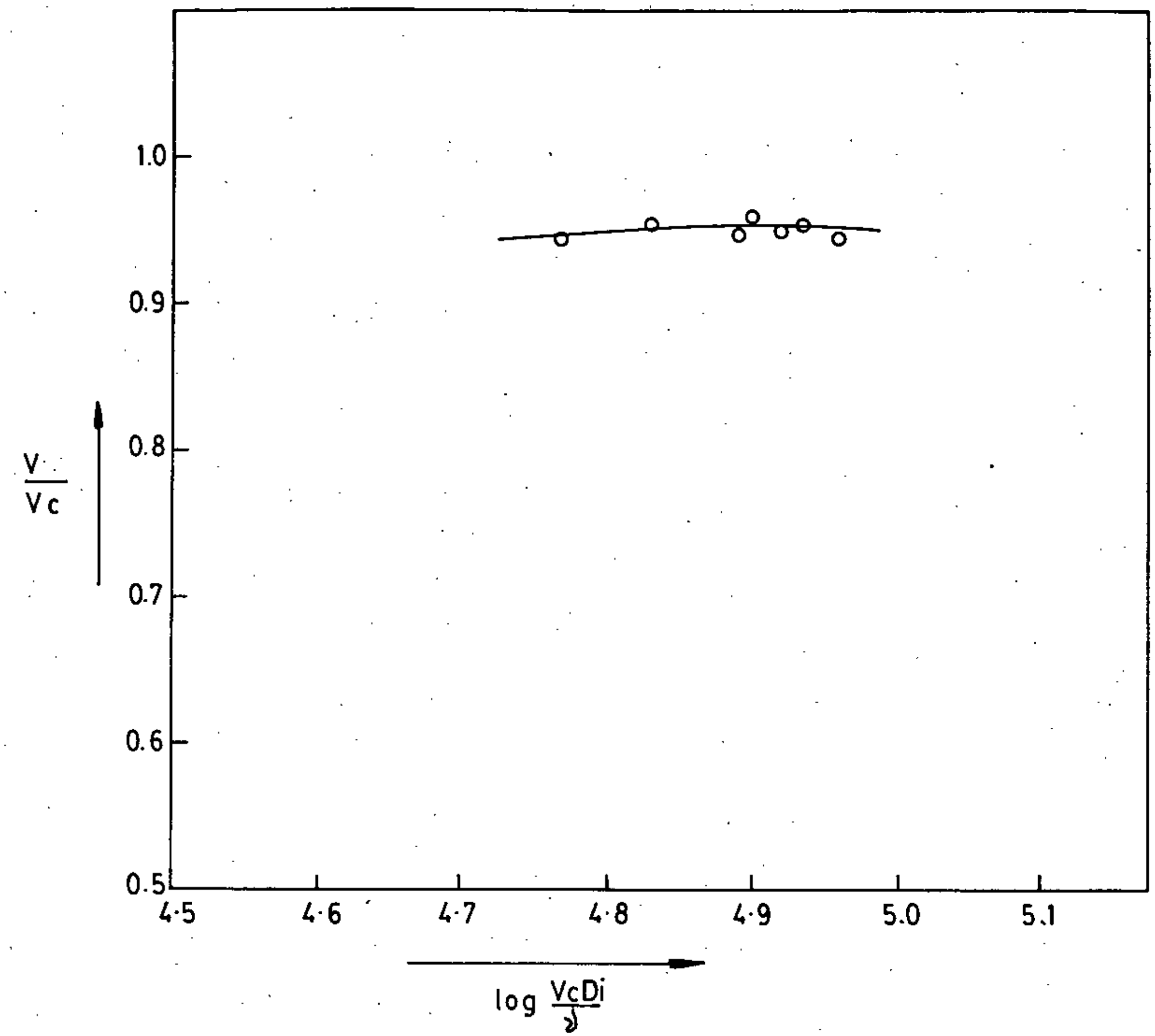


FIG.5.3 RELATION BETWEEN MEAN AND AXIAL VELOCITY IN SHAPED INLET

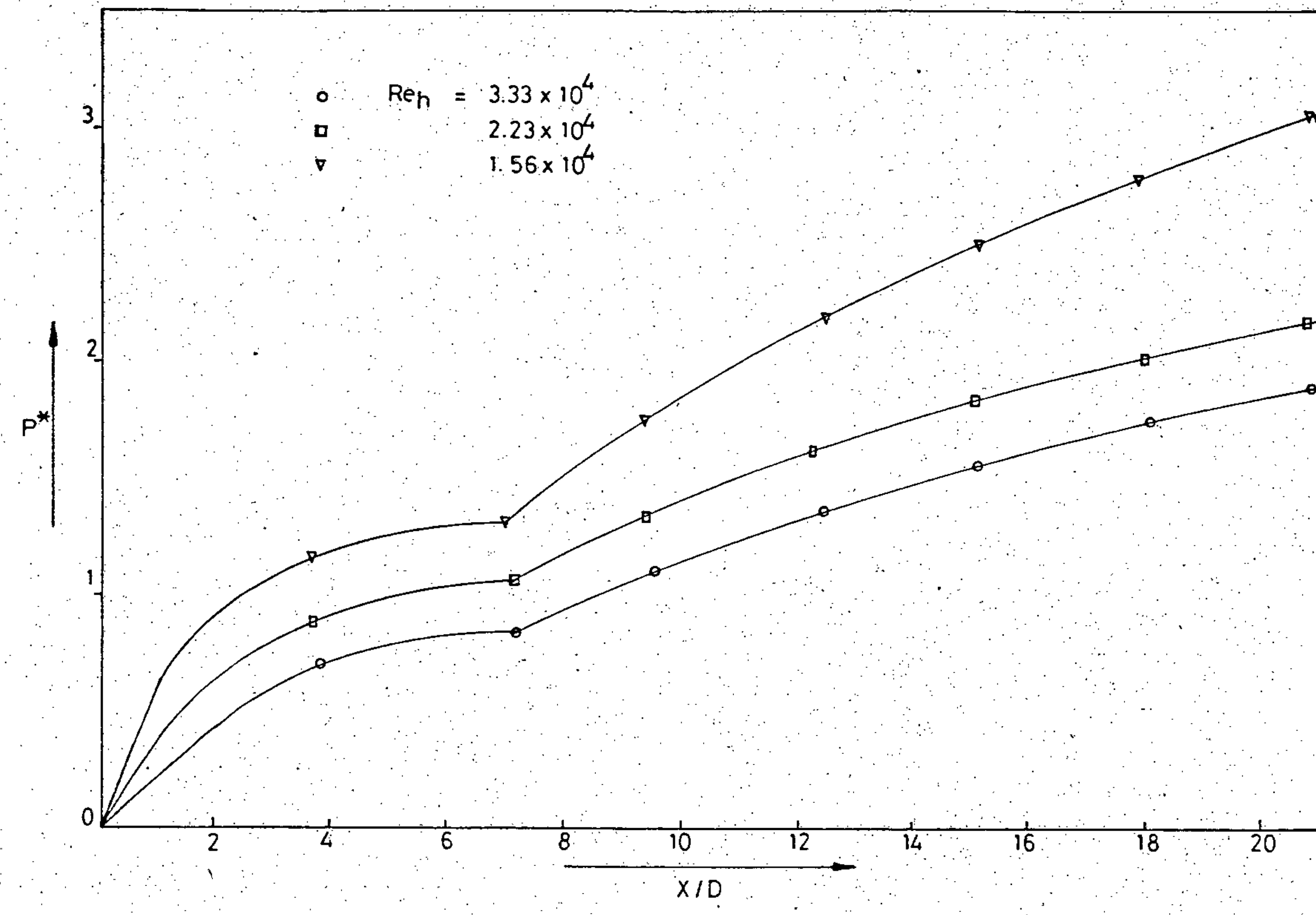


FIG. 5.4 PRESSURE DROP ALONG THE LENGTH OF THE TUBE

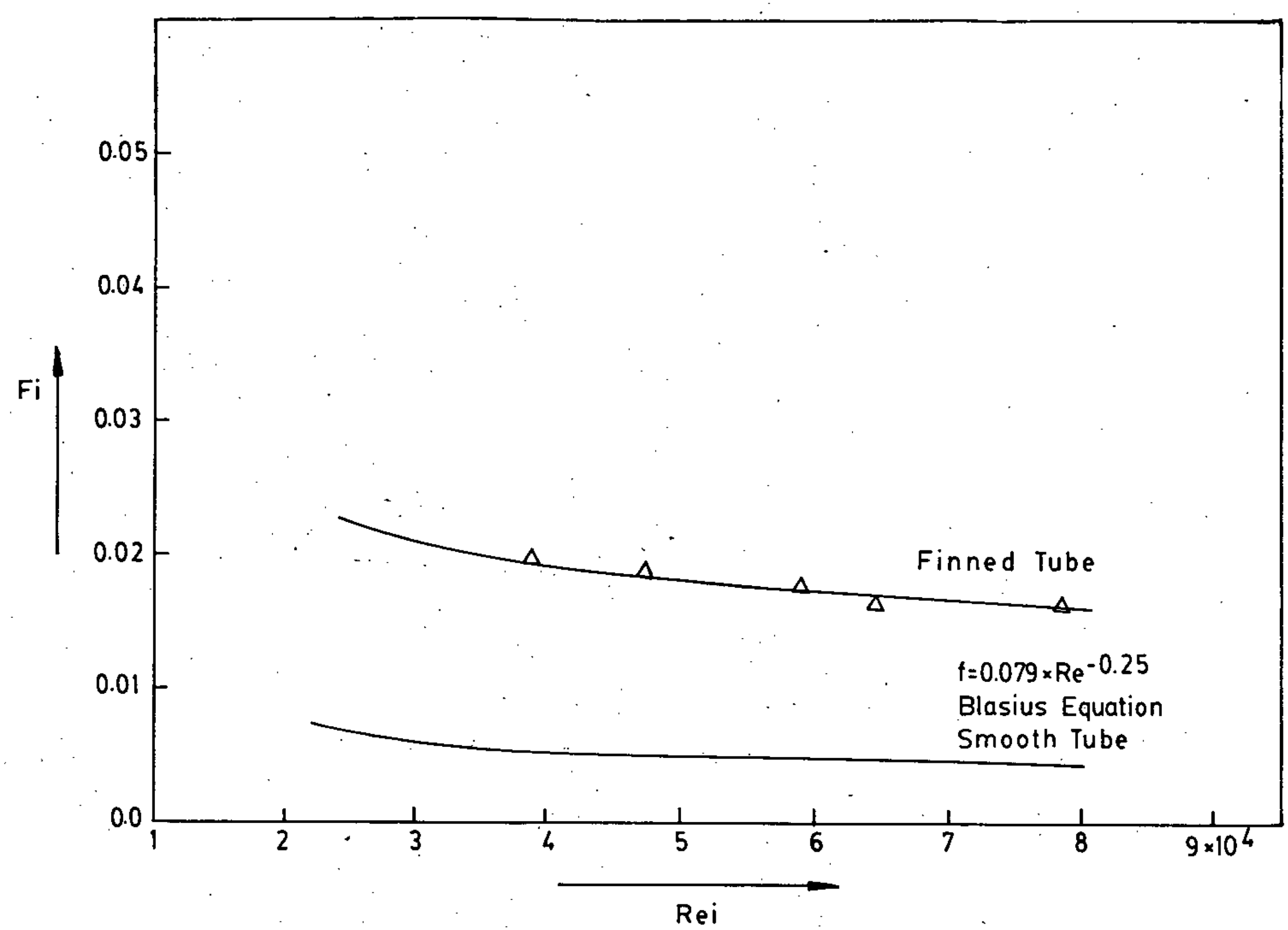


FIG. 5.5 FRICTION FACTOR BASED ON INSIDE DIAMETER

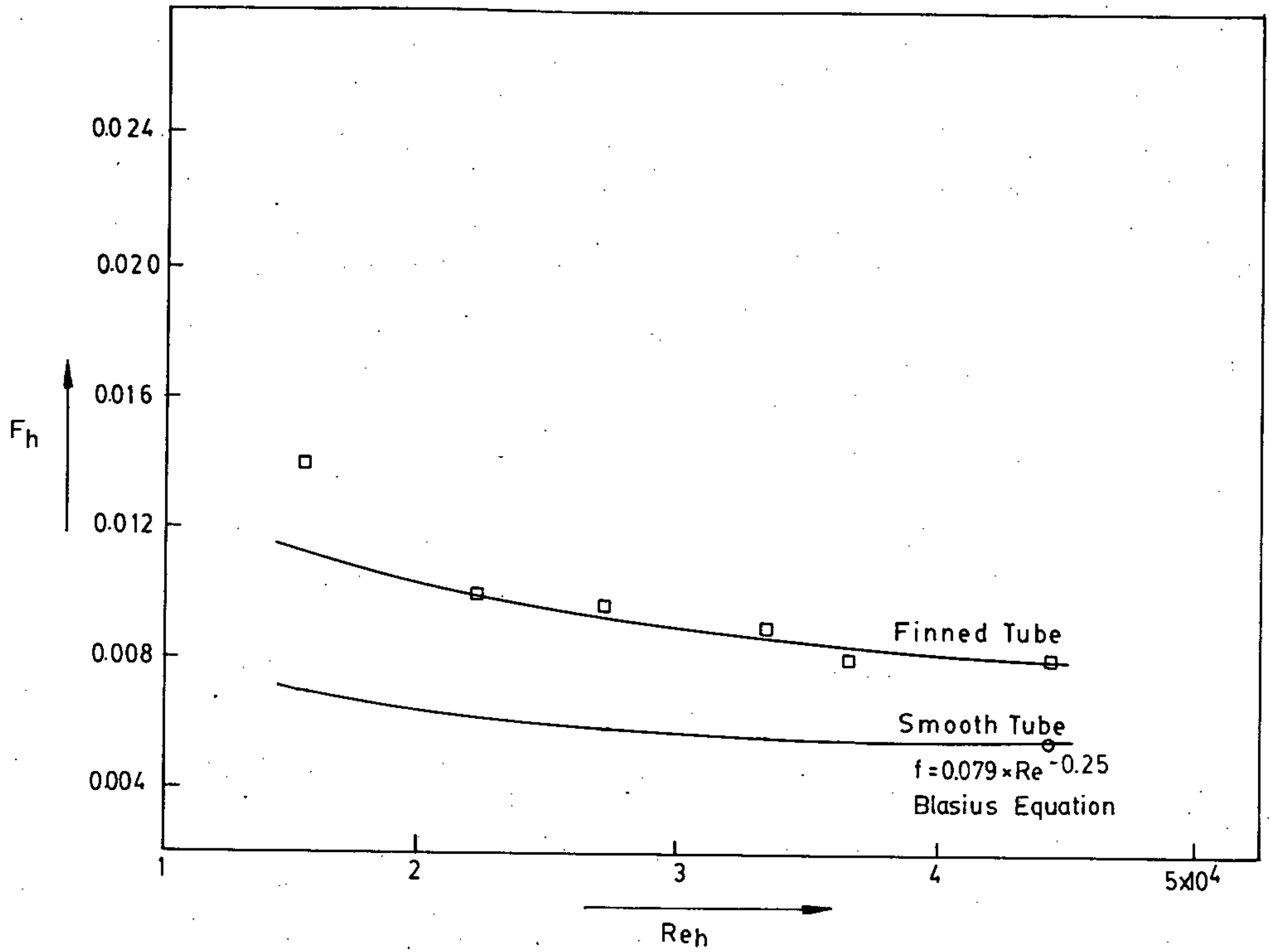


FIG. 5.6 FRICTION FACTOR BASED ON HYDRAULIC DIAMETER

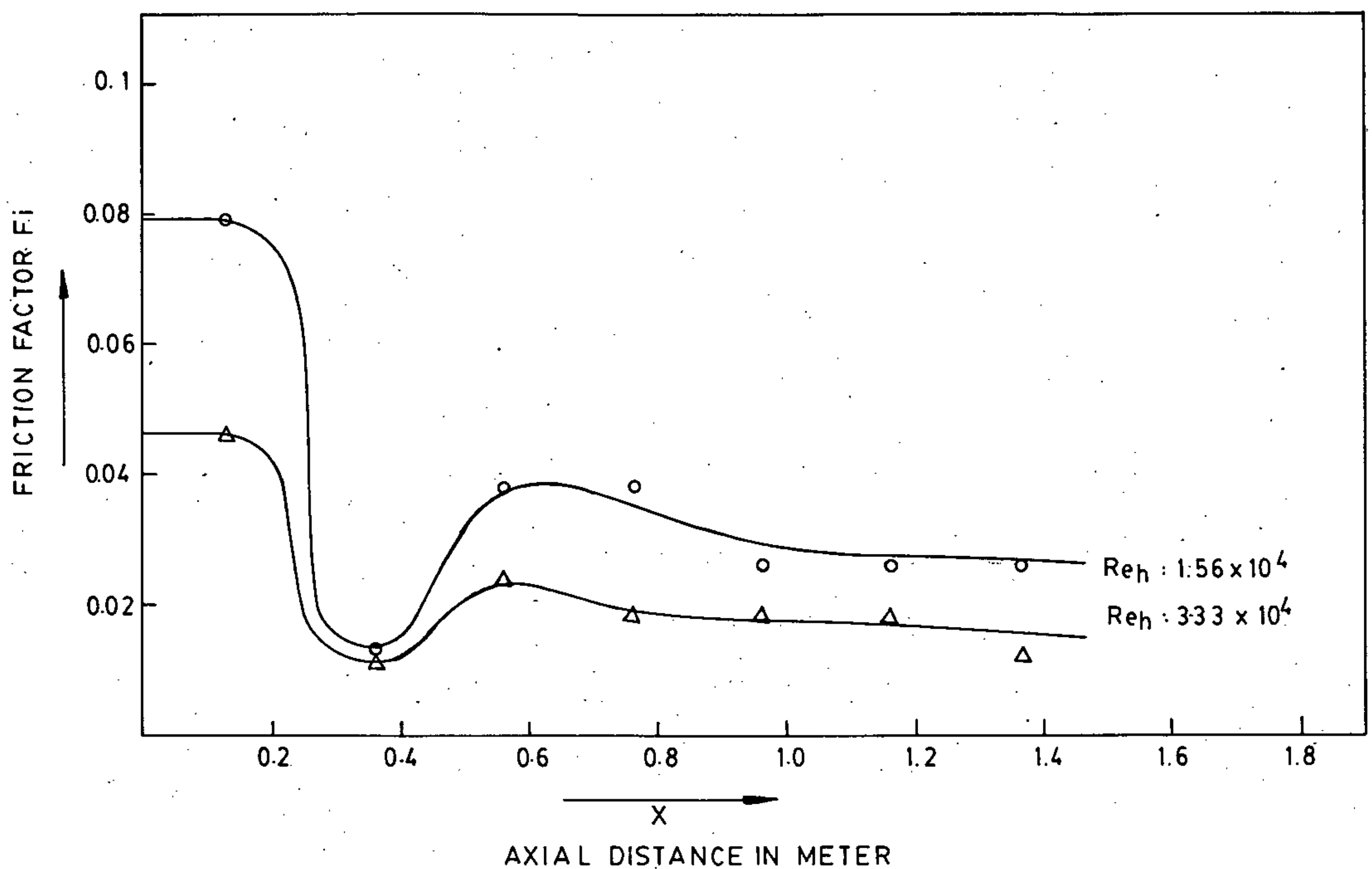


FIG. 5.7 DISTRIBUTION OF FRICTION FACTOR ALONG AXIAL DISTANCE

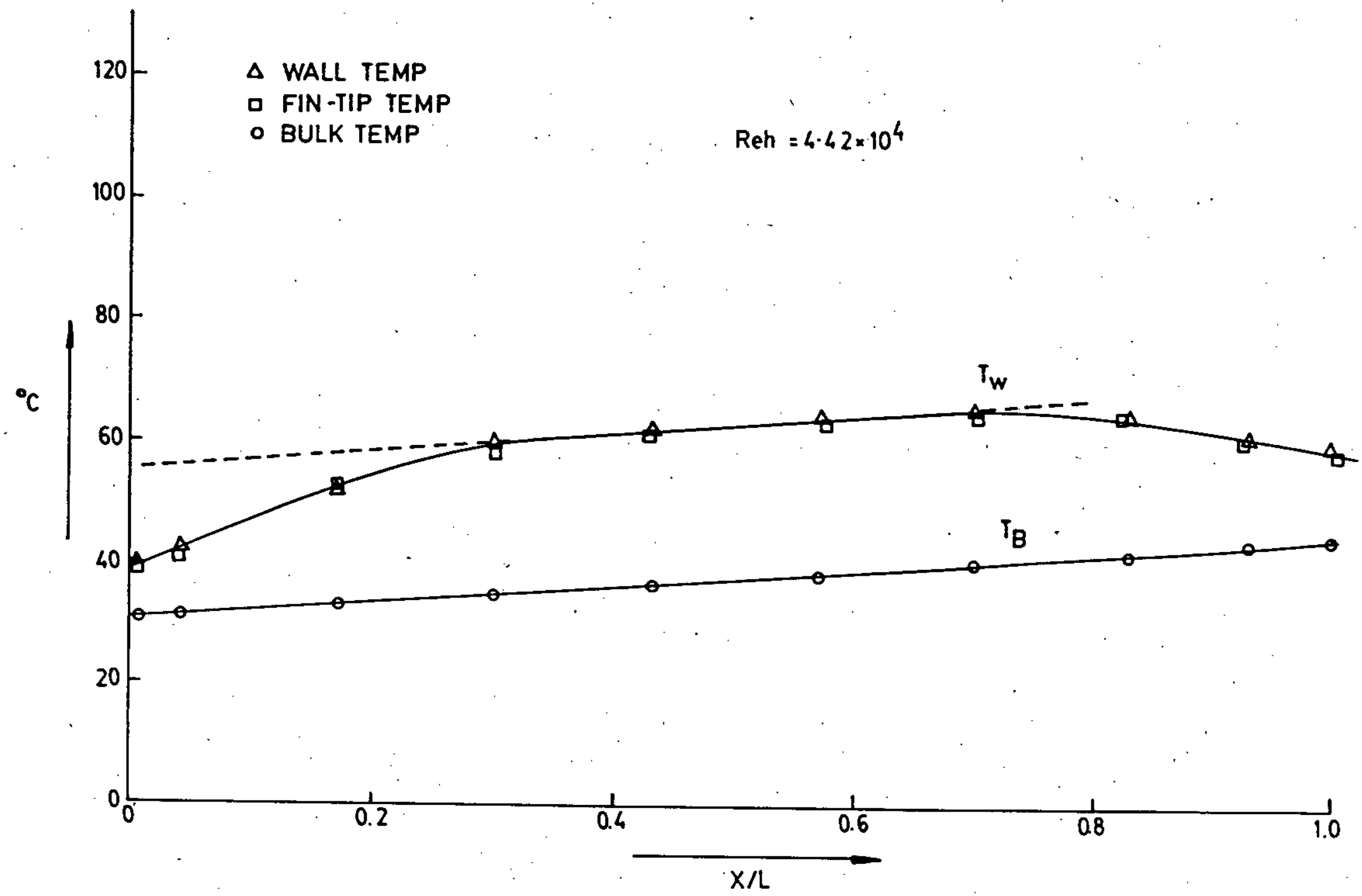


FIG. 5.8 WALL FIN-TIP AND BULK TEMP DISTRIBUTION ALONG AXIAL DISTANCE

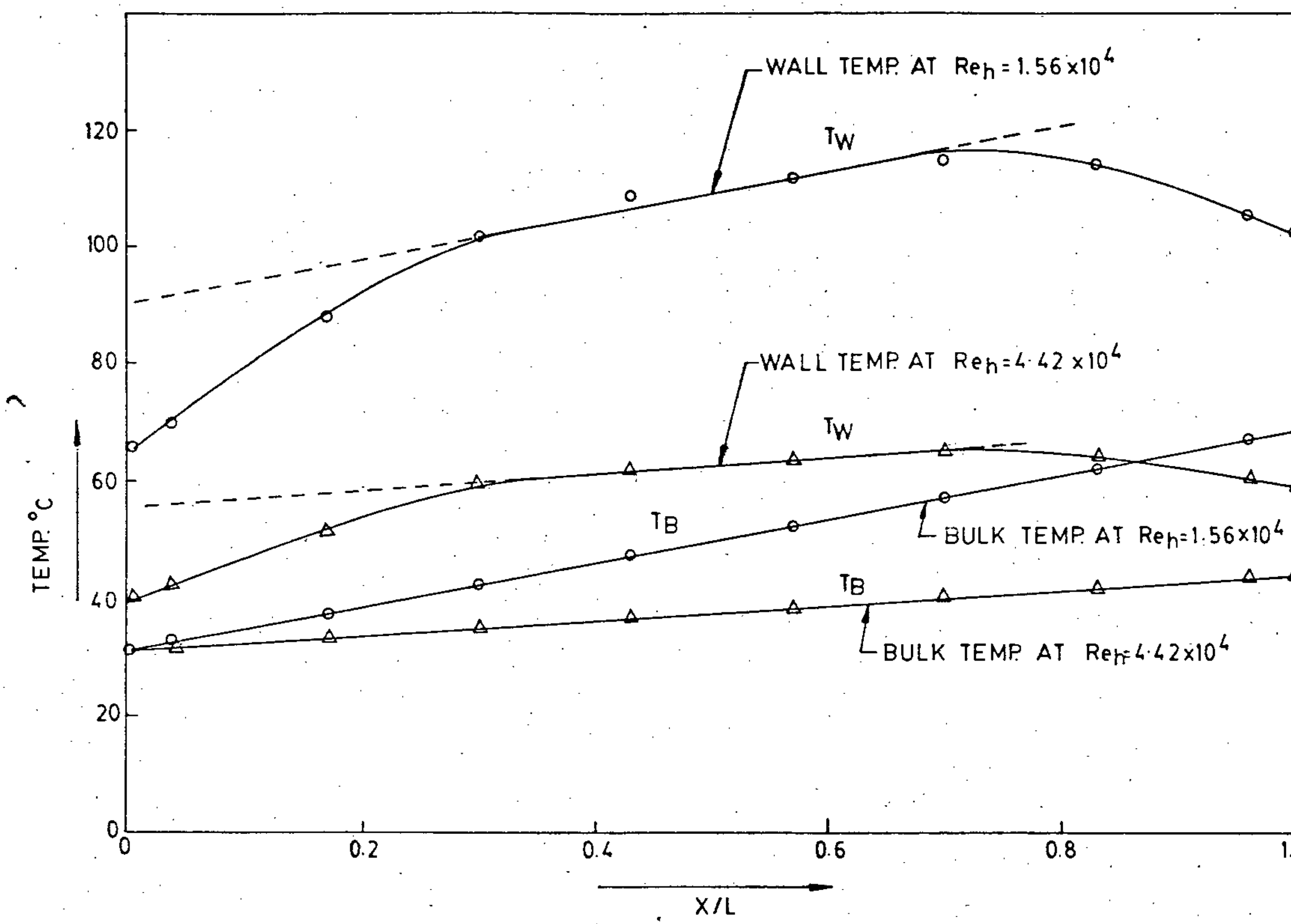


FIG. 5.9 WALL AND BULK TEMP. DISTRIBUTION ALONG AXIAL DISTANCE FOR REYNOLDS NUMBER 1.56×10^4 AND $4 \cdot 42 \times 10^4$

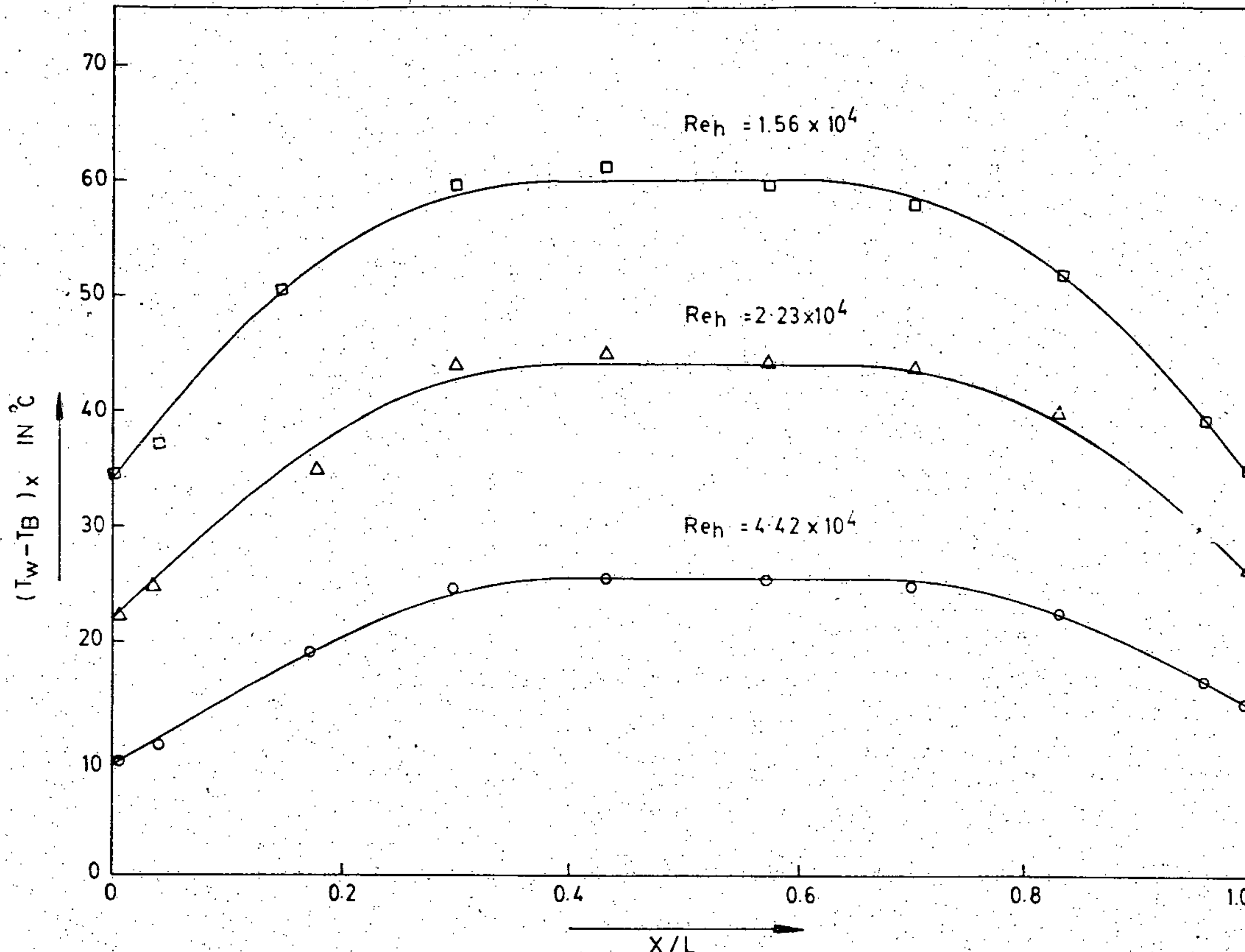


FIG.5.10 DISTRIBUTION OF WALL-TO-BULK TEMP. DIFFERENCE ALONG LENGTH OF THE TUBE

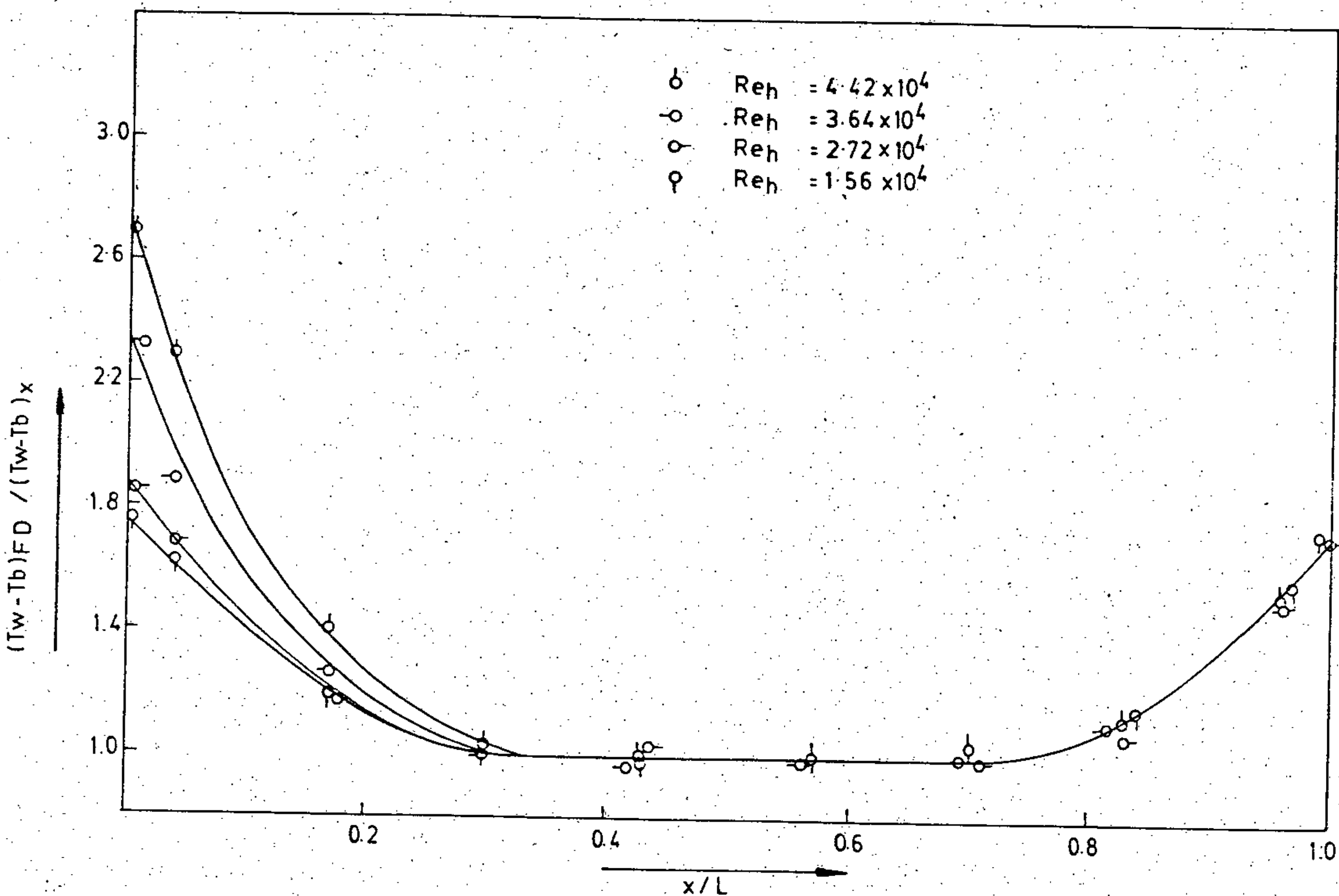


FIG. 5.11 DISTRIBUTION OF TEMP. RATIO $(T_w - T_b)_{FD} / (T_w - T_b)_x$ ALONG THE LENGTH OF FINNED TUBE

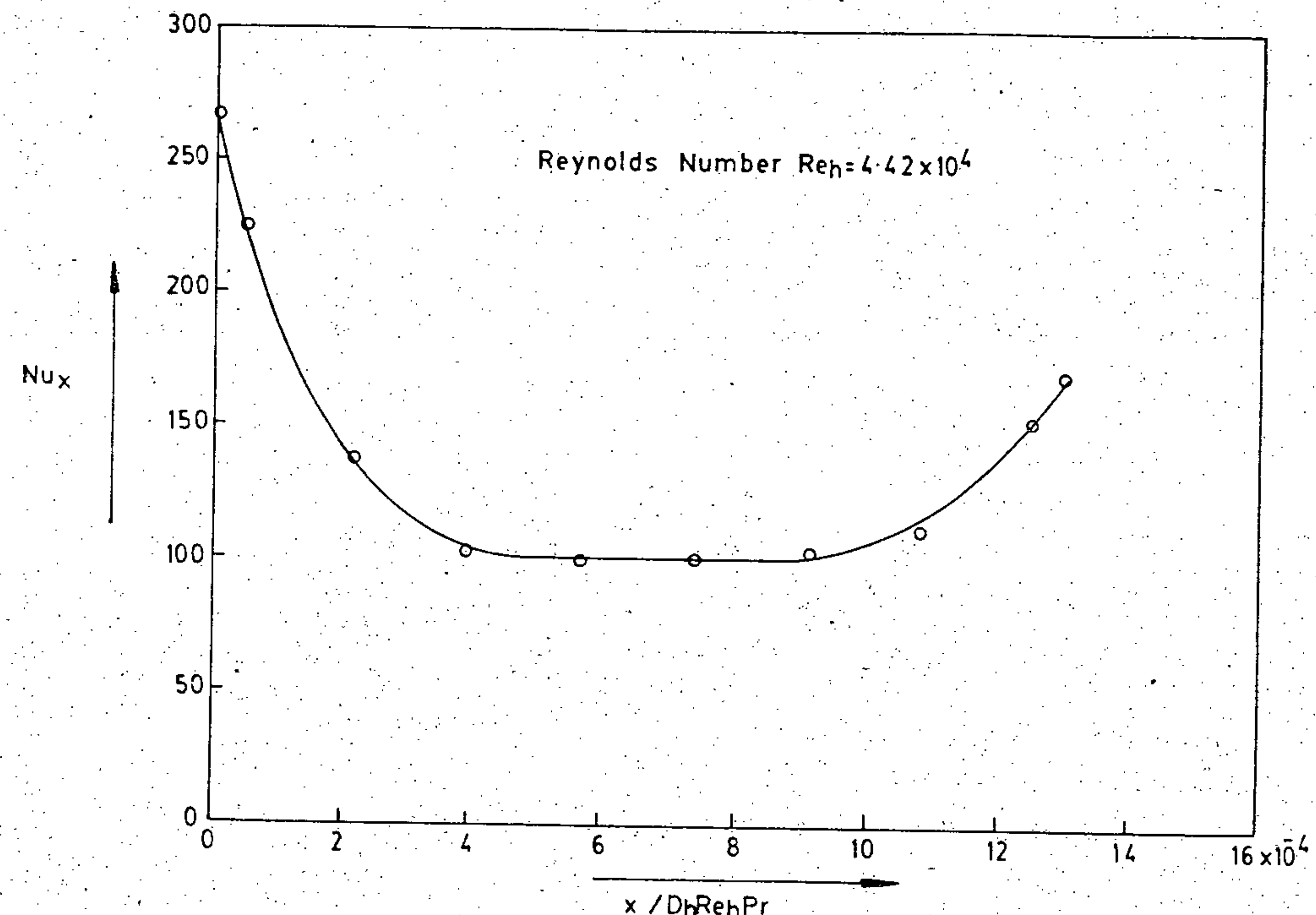


FIG. 5.12 LOCAL NUSSELT NUMBER ALONG AXIAL DISTANCE

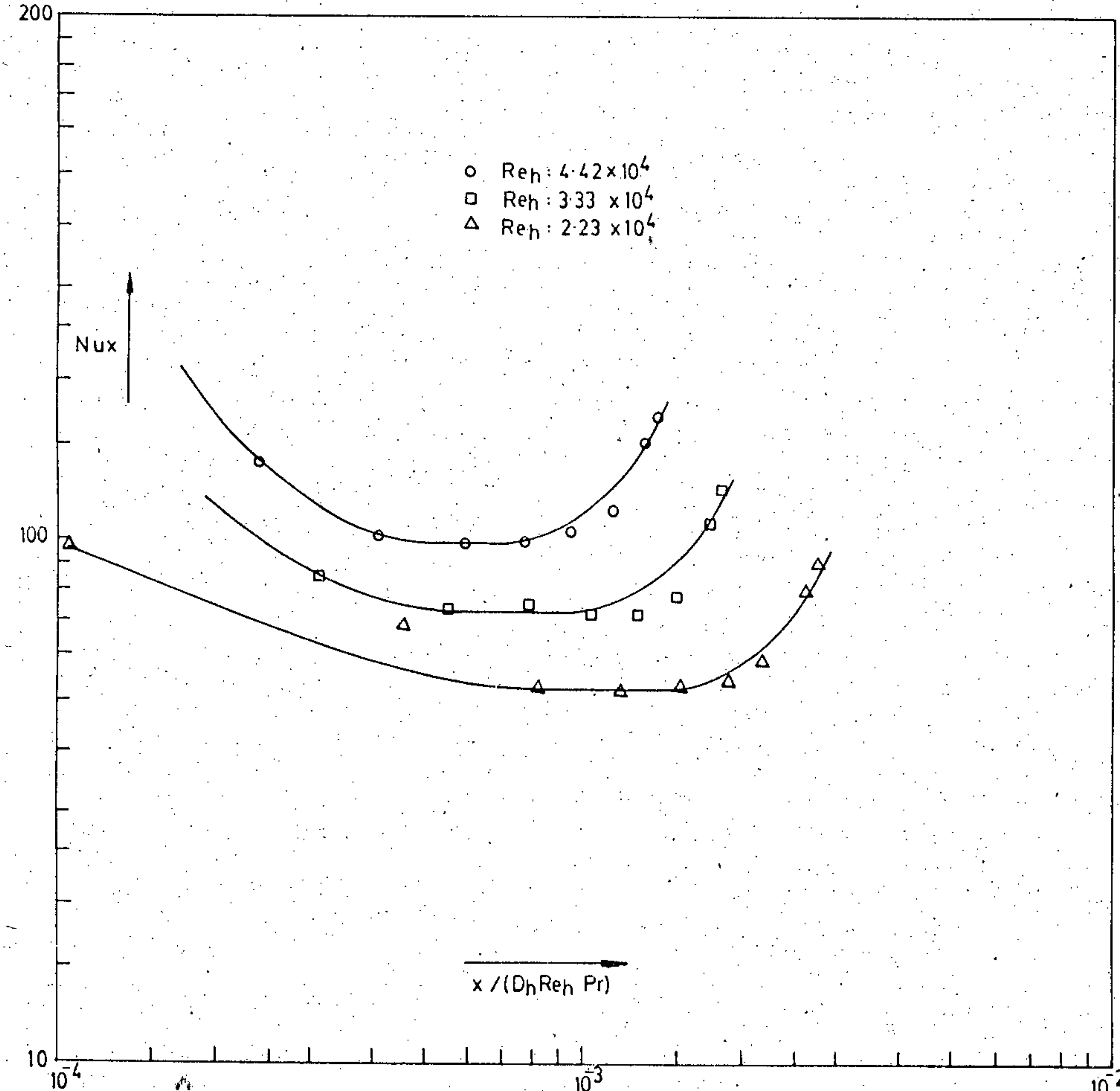


FIG. 5.13 LOCAL NUSSELT NUBER ALONG AXIAL DISTANCE FOR DIFFERENT REYNOLDS NUMBER

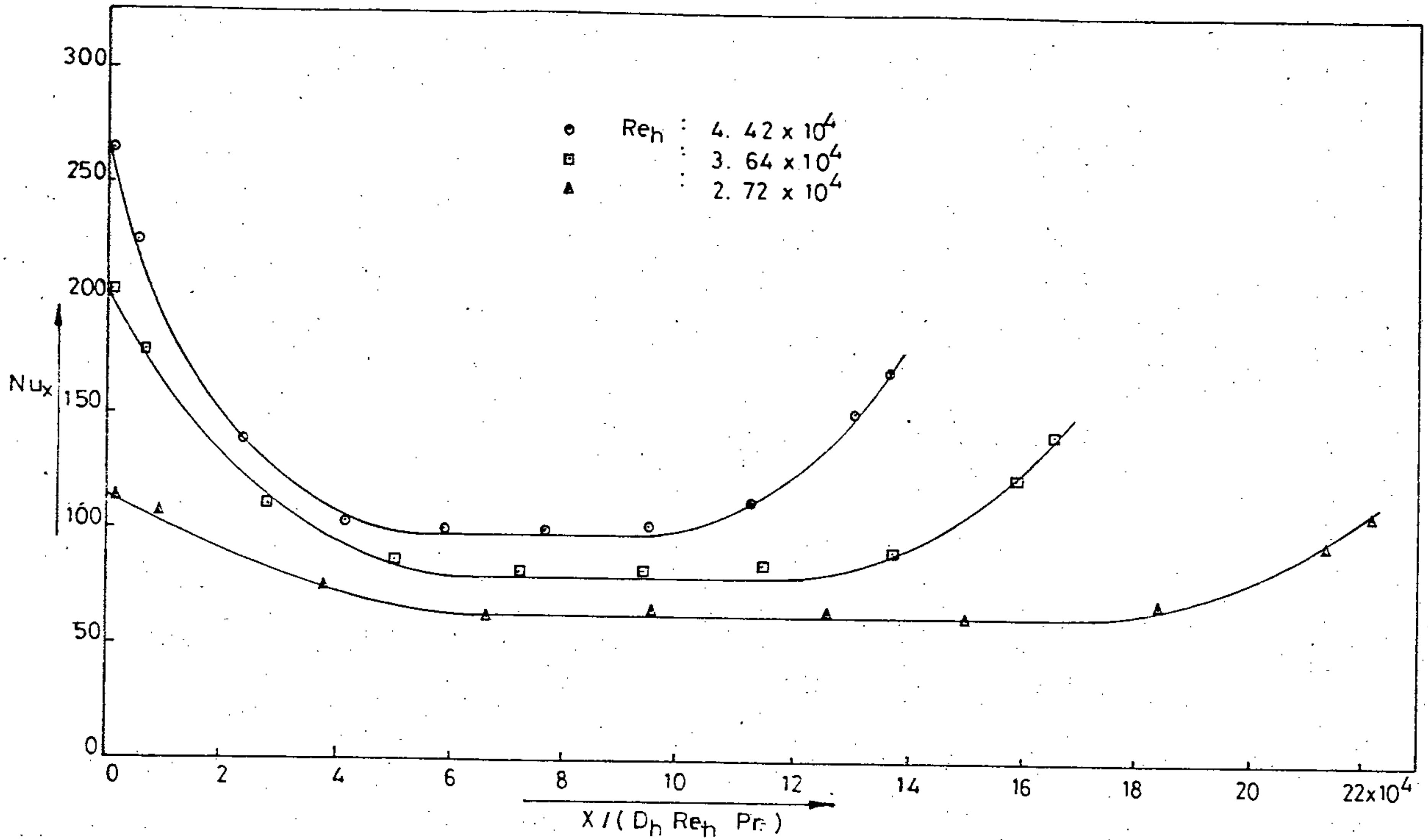


FIG. 5.14 LOCAL NUSSELT NUMBER ALONG THE LENGTH OF THE CIRCULAR TUBE HAVING INTERNAL LONGITUDINAL FINS FOR DIFFENT REYNOLDS NUMBERS.

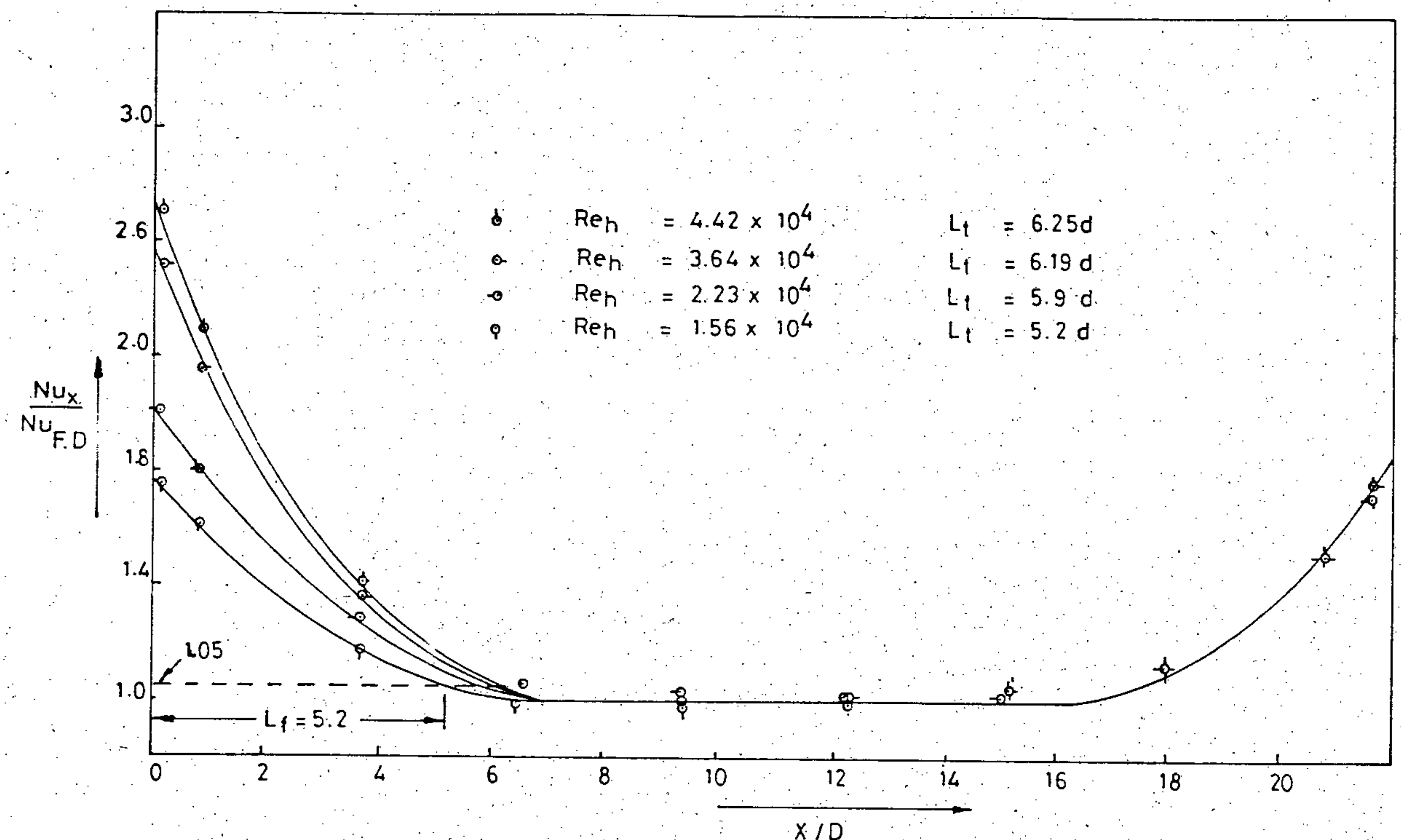


FIG. 5.15 DISTRIBUTION OF NUSSELT NUMBER RATIO Nux/Nu_{FD} ALONG THE LENGTH OF FINNED TUBE

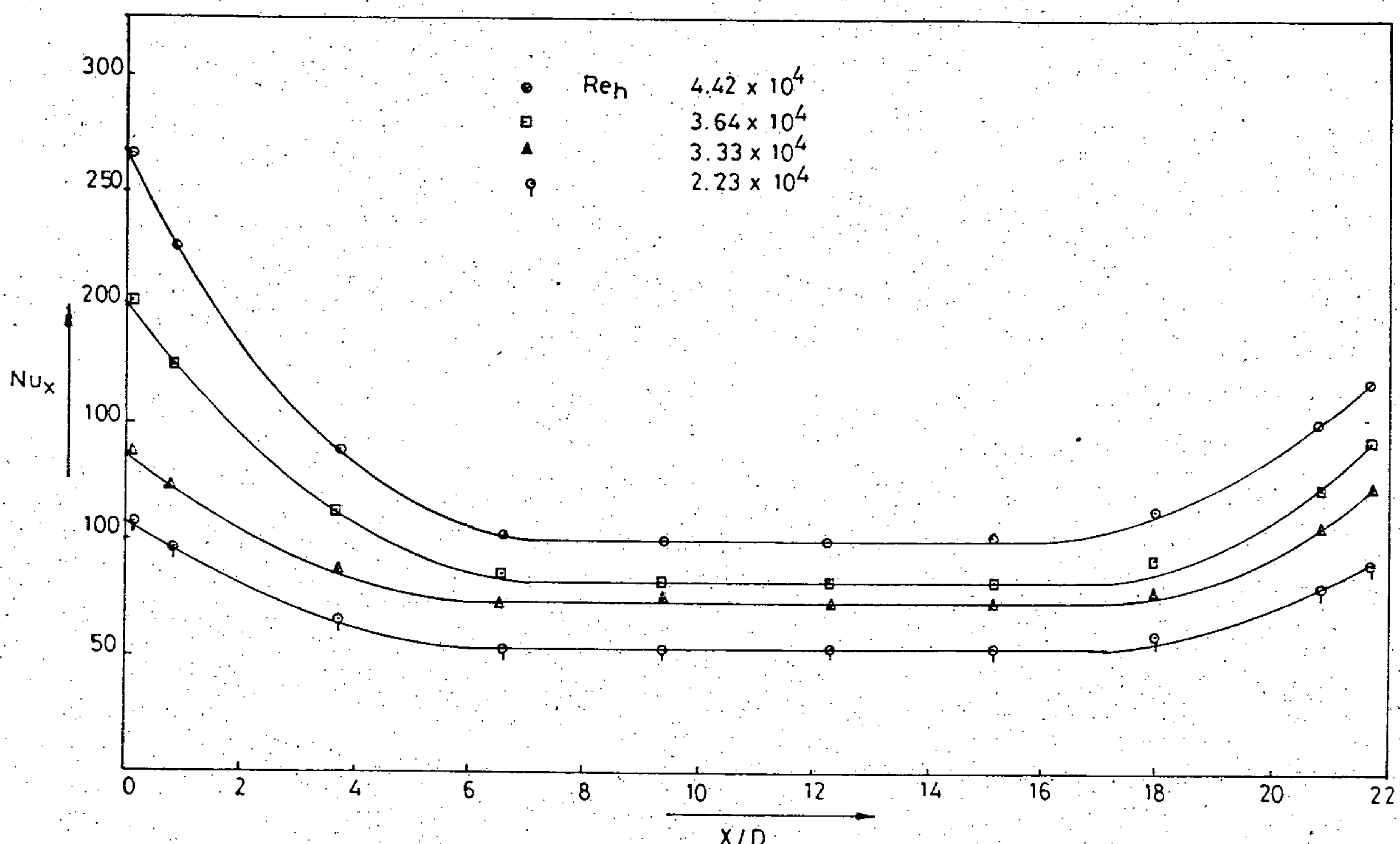


FIG. 5.16 LOCAL NUSSELT NO ALONG THE LENGTH OF FIN TUBE AT DIFFERENT REYNOLDS NUMBERS

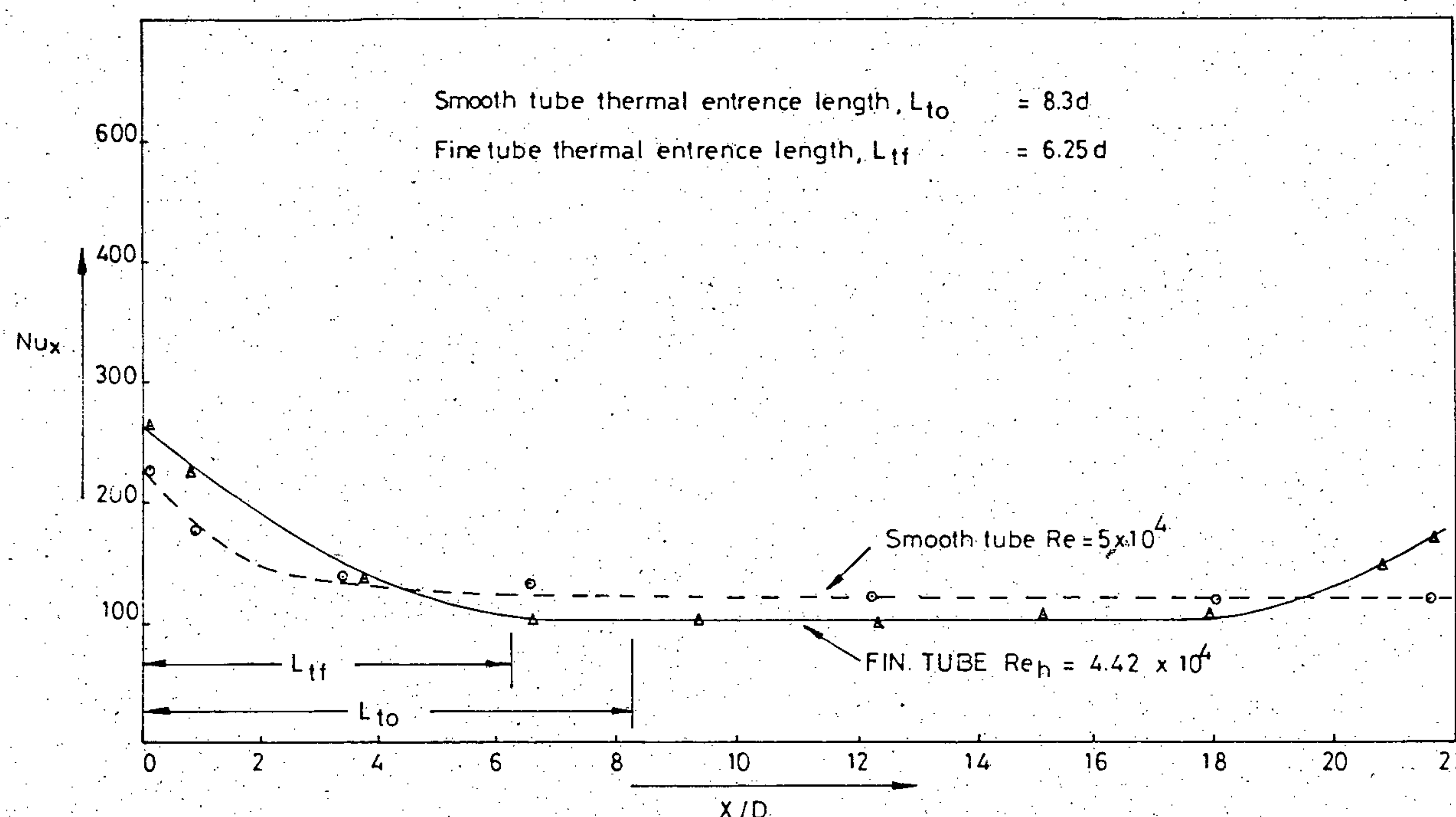


FIG. 5.17 LOCAL NUSSELT NUMBER BASED ON HYDRAULIC DIAMETER AND EFFECTIVE AREA AT DIFFERENT AXIAL DISTANCE AND COMPARISON WITH SMOOTH TUBE

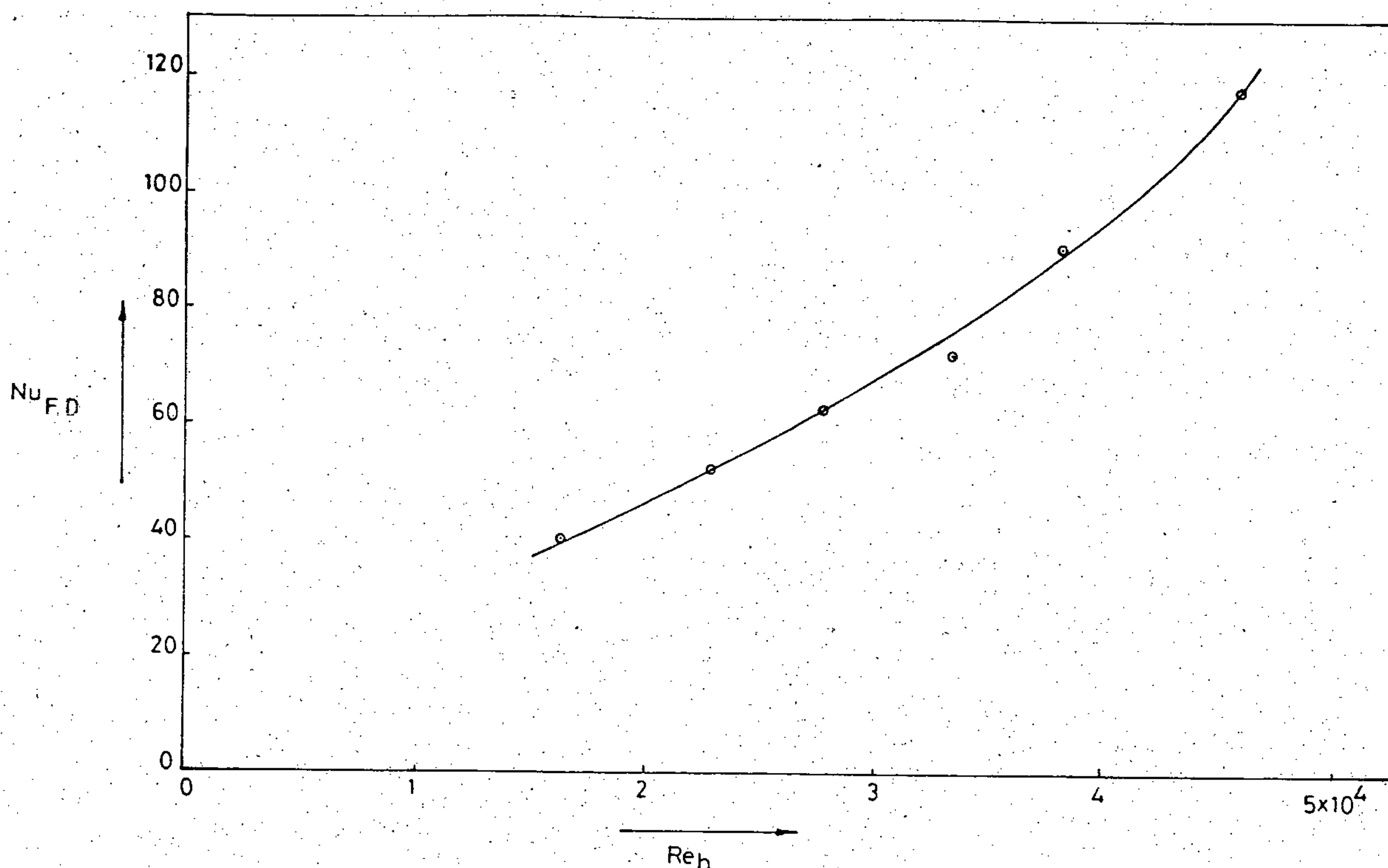


FIG. 5.18 FULLY DEVELOPED NUSSELT NUMBER AT DIFFERENT REYNOLDS NUMBERS

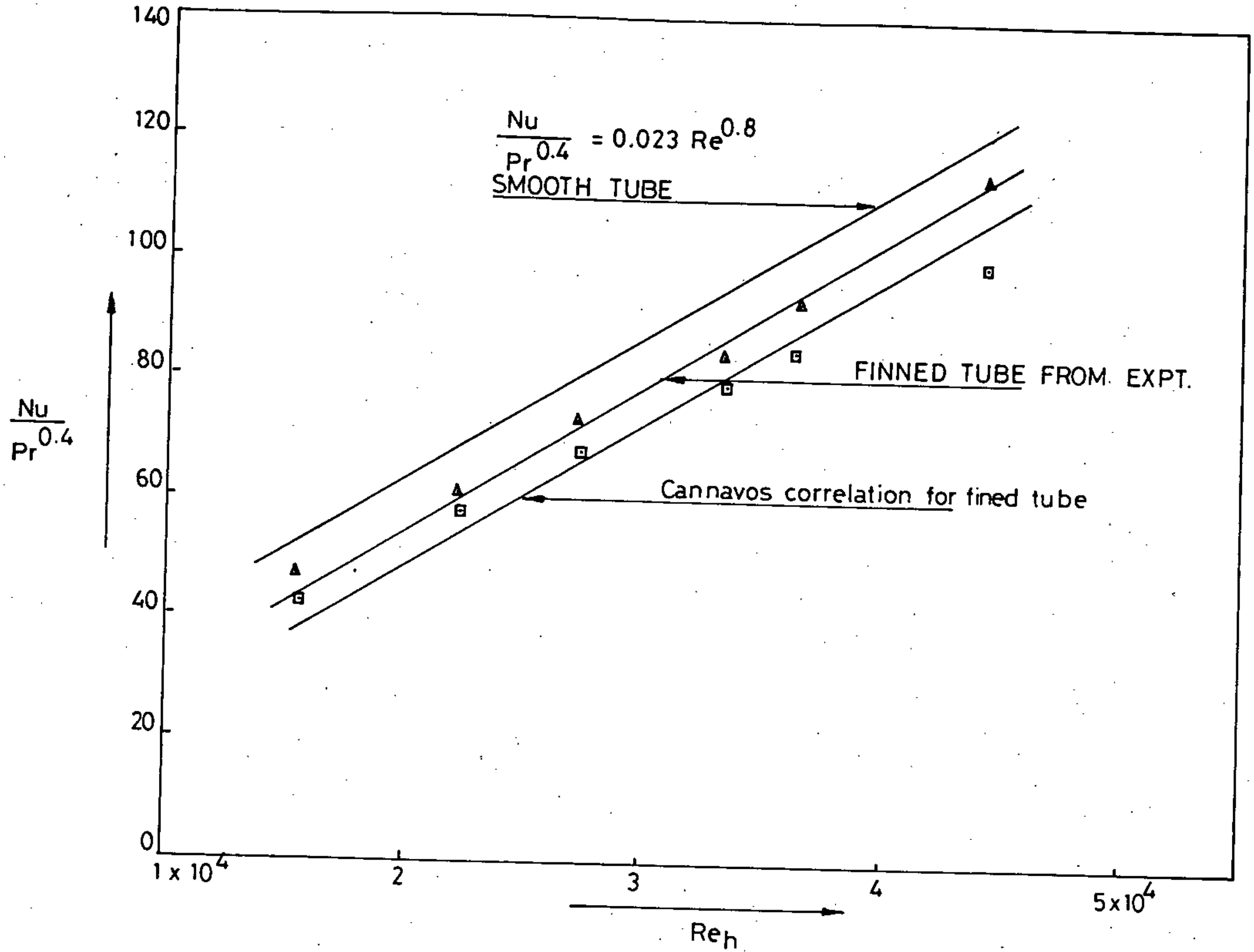


FIG. 5.19 HEAT TRANSFER RESULTS BASED ON HYDRAULIC DIAMETER AND EFFECTIVE AREA.

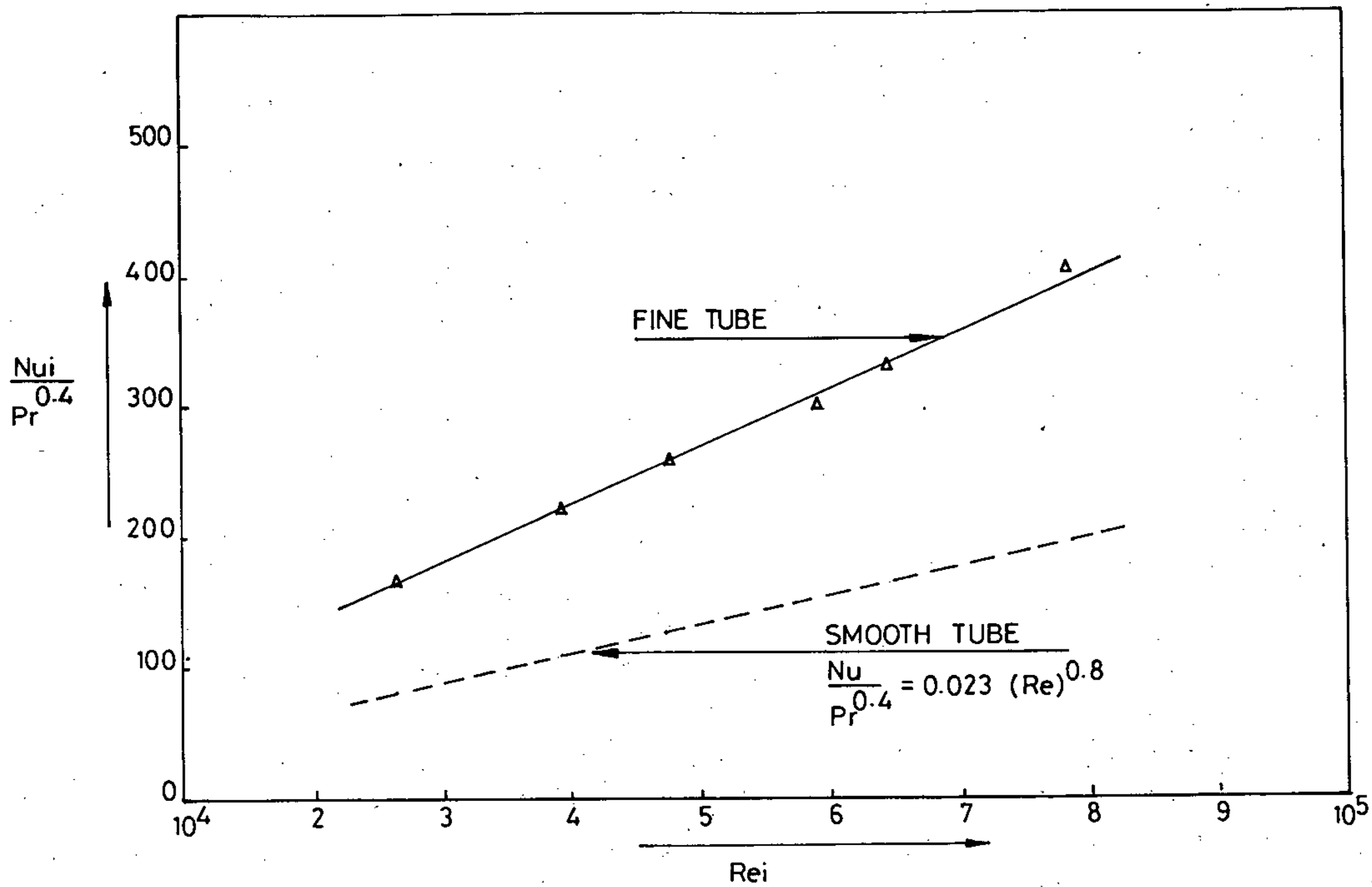


FIG. 5.20 HEAT TRANSFER RESULT BASED ON INSIDE DIAMETER AND NOMINAL AREA.

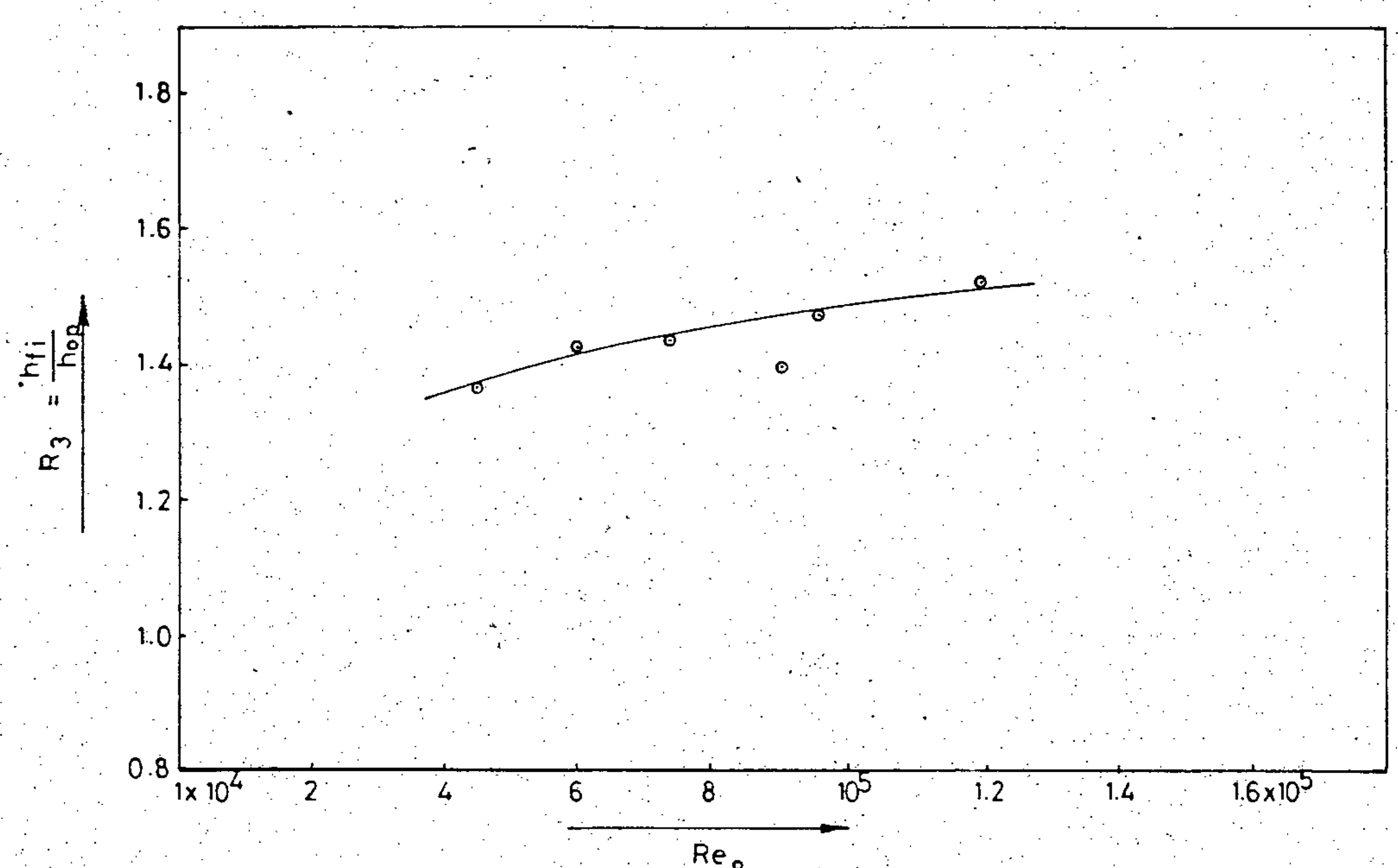


FIG. 5.21 CONSTANT PUMPING POWER COMPARISON WITH SMOOTH TUBE

PLATES

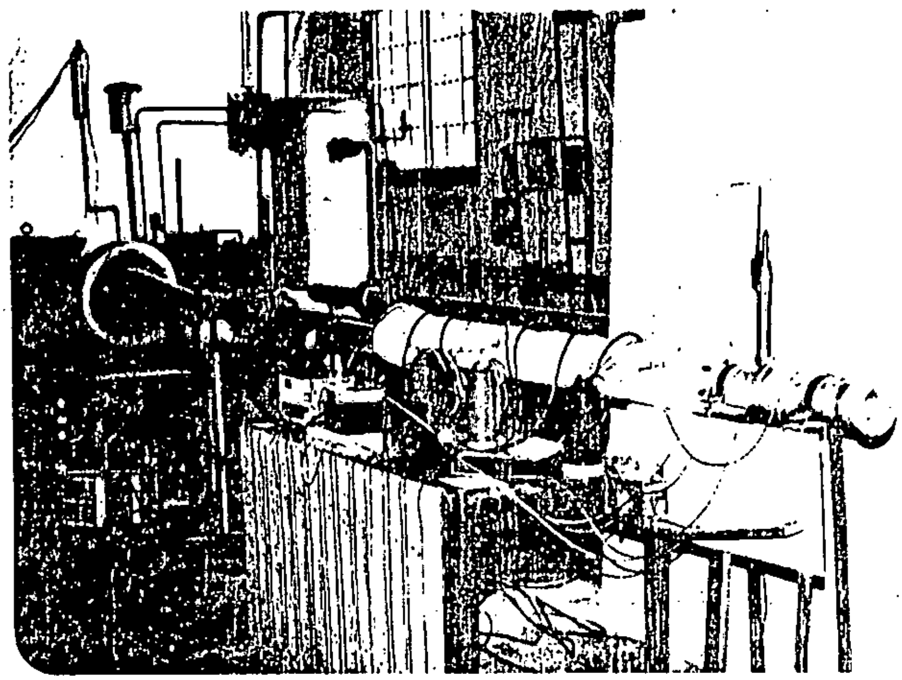


Plate 4.1 Experimental Set-up

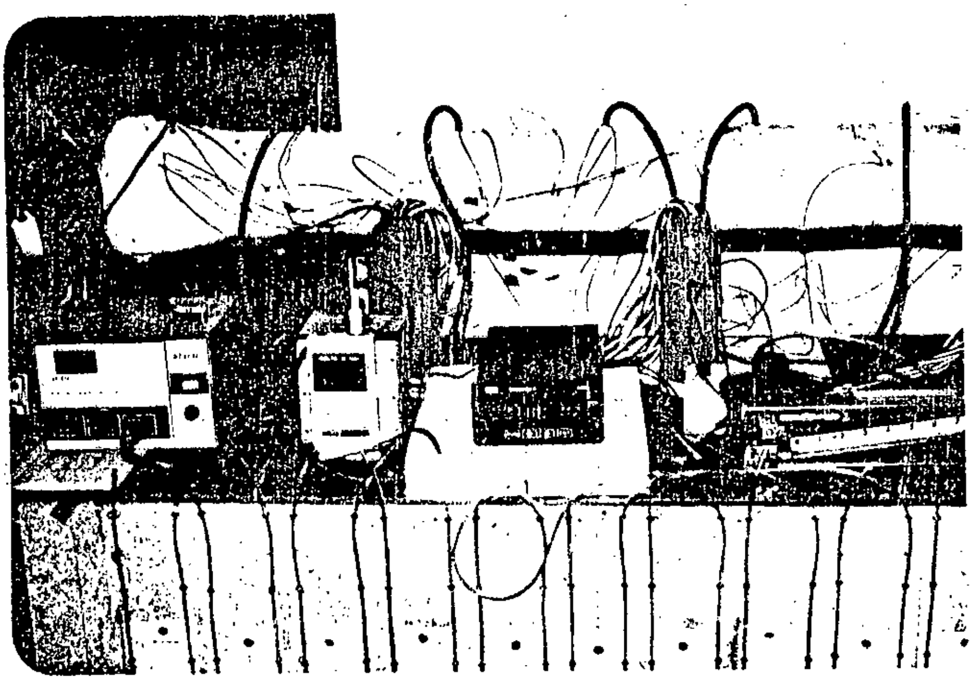


Plate 4.2 Measuring Instruments

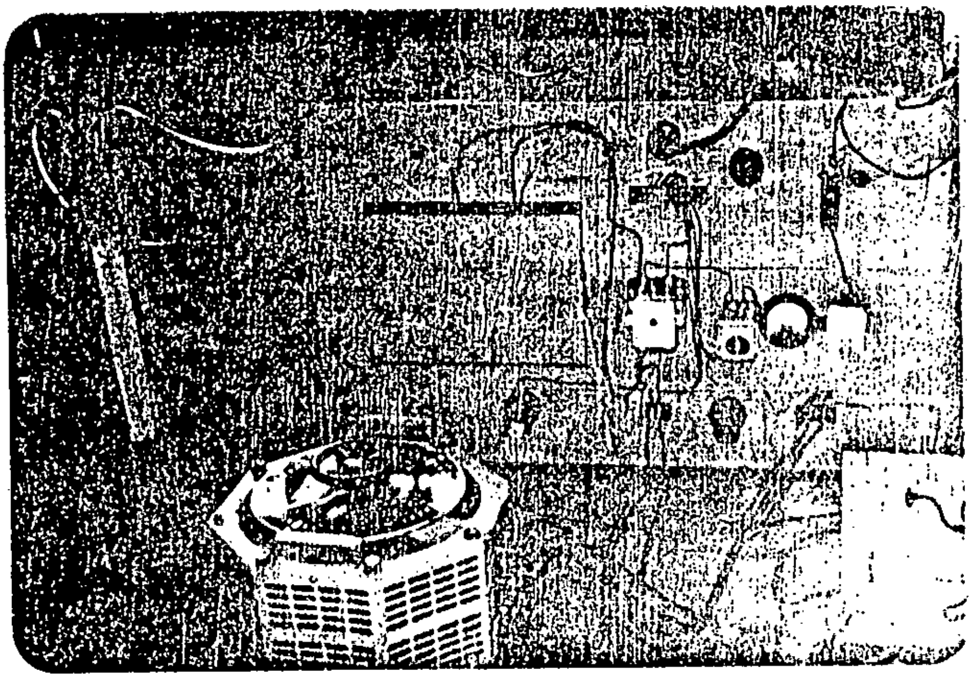


Plate 4.3 Voltage Regulating Transformer and Panel Board for Electrical Heating System.

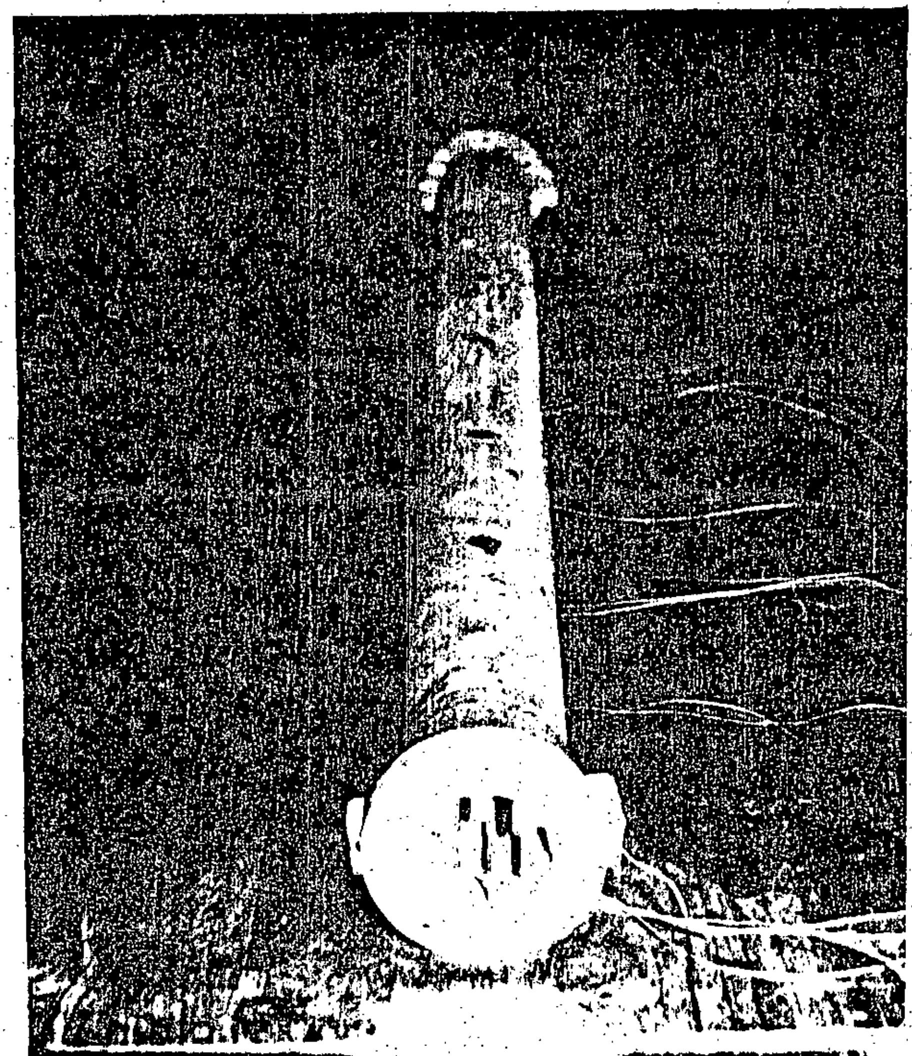


Plate 4.4 Test section

Plate 4.5   Two Halves of Test Section.

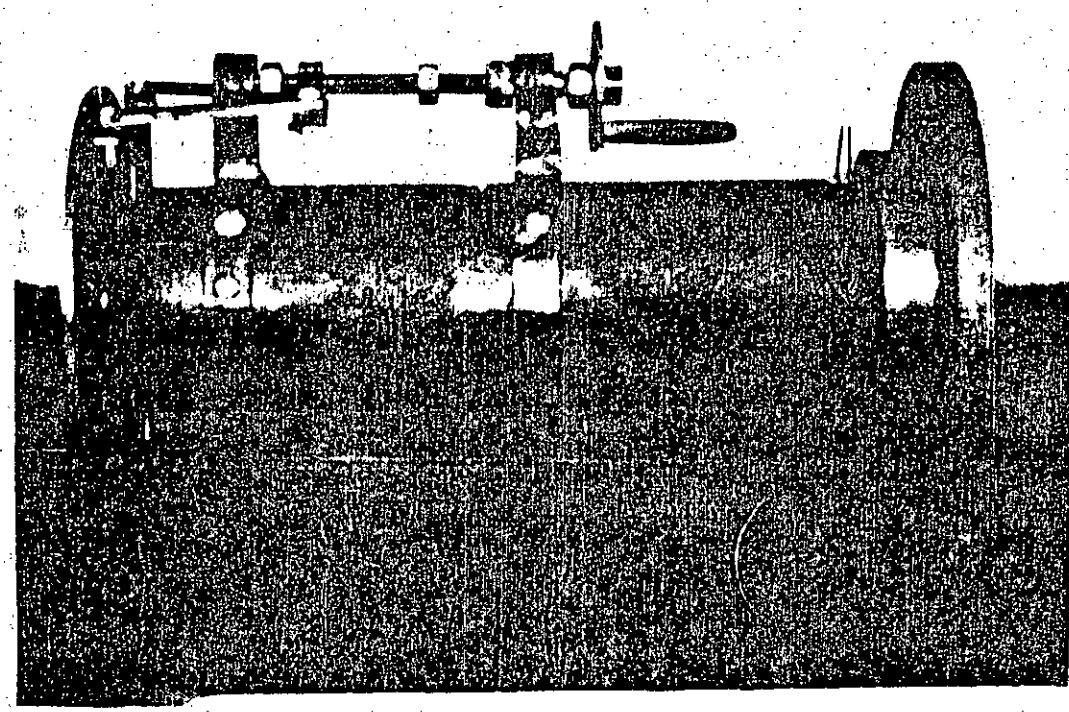
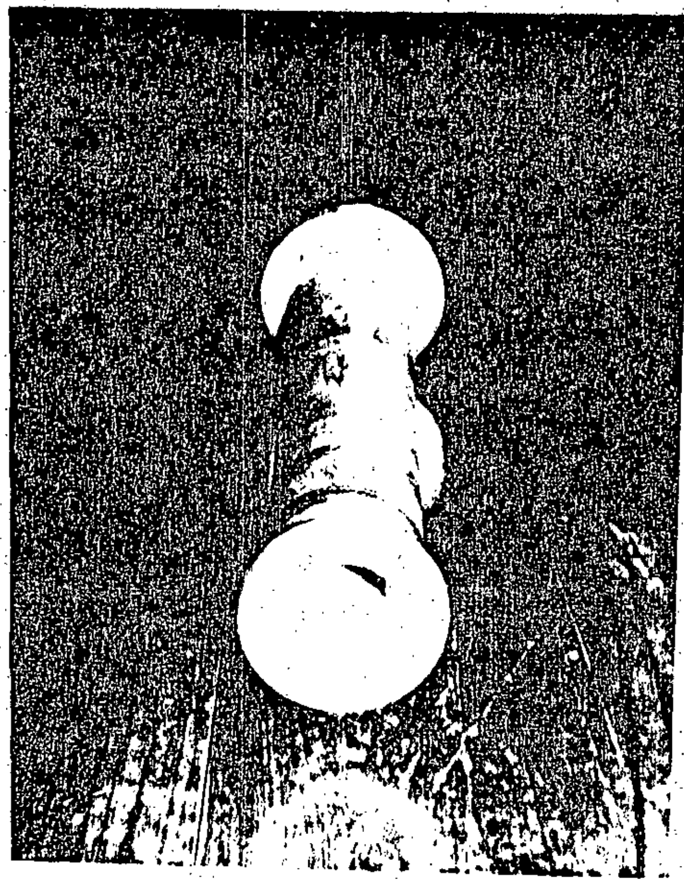


Plate 4.6 Flow Control Valve



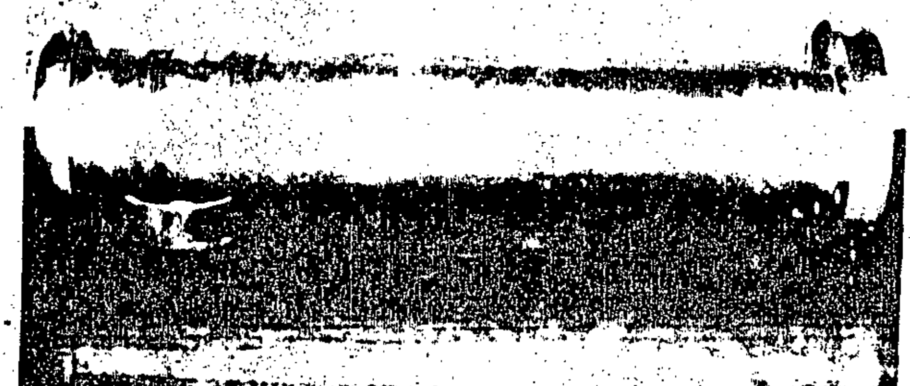


Plate 4.7 Shaped inlet

APPENDICS

APPENDIX - A

Table (2.1) : Tube Dimensions.

Tube No	9	14	16	24	
Reference	(3	(3)	9	9	Present work
Tube O.D. mm	12.7	15.9	9.53	25.4	96
Tube I.D. mm	10.3	13.9	7.64	23.8	70
Fin Tip Diameter mm	7.75	10.9	3.18	19.8	40
No. of Fins	10	10	6	16	6
Helix angle	0	0	0	20.0	0
Actual Flow Area, A_{xf} mm^2	73.6	137	33.4	415	3578.5
Nominal Flow A Area, A_x sq.mm	83.1	151	45.9	444	3848.5
Core Flow Area, A_{xc} sq.mm	47.2	93.7	7.92	309	1256.6
A_{xf} / A_x	0.886	.907	0.728	0.936	0.93
A_{xf} / A_{xc}	1.56	1.46	4.22	1.34	2.85
Actual heat transfer Area, A_h sq.mm / mm	54.0	67.3	50.8	132	399.9
Nominal heat transfer Area, A sq.mm / mm	32.3	43.6	24.0	74.7	219.9
A_h / A	1.67	1.55	2.12	1.76	1.82
Hydraulic Dia, D_h	5.45	8.15	2.63	12.6	35.79

Table (2. 2) Data Comparison.

Ref.	Tube No	Reynolds Numbers, $Re_{\theta 3}$				Remarks
		10^4	2.5×10^4	5×10^4	10^5	
1	3	2.75	2.20	2.10	No data	Water
2	9	1.63	1.46	1.31	1.10	Water
3	9	No data	1.67	1.48	1.32	Air
9	24	No data	1.65	1.62	1.53	Air
1	4	1.69	1.45	1.31	No data	Water
2	14	1.50	1.28	1.11	0.95	Water
3	14	No data	0.95	1.11	1.00	Air
9	14	No data	1.25	1.28	1.28	Air

APPENDIX - B

Table (4.1) Shaped Inlet.

Co-ordinates for shaped inlet

x/d	0	0.094	0.109	0.125	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	0.010	0.013	0.016	0.019	0.024	0.029	0.034	0.041	0.048
x/d	0.359	0.375	0.391	0.406	0.422	0.438	0.453	0.469	0.453
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	

Table (4.2)

1. **Fan :**		
Capacity :	:	30 Cu.m / 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		
Heater Resistance	:	8.75 Ohm
Maximum voltage	:	220 Volts
Maximum current	:	25 A
Power	:	5.5 KW.

Table (4.3) Location of Measuring points for the Log Linear Method.

Number of measuring points per diameter	Distance form wall in pipe diameters
4	0.043, 0.290, 0.710, 0.957
6	0.032, 0.135, 0.321, 0.679, 0.865, 0.968
8	0.021, 0.117, 0.184, 0.345 0.655, 0.816, 0.883, 0.979
10	0.019, 0.077, 0.153, 0.217, 0.361 0.639, 0.783, 0.847, 0.923, 0.981

APPENDIX - C

TABLE 5.1

Expt. Nos.	Mean Velocity V m / s By Graphical Integration	Mean Velocity V m / s By Ten points Log Linear Method.	% of error in Log Linear Method.
1	19.23	19.32	0.47
2	18.41	18.52	0.57
3	17.85	17.89	0.22
4	17.19	17.29	0.58
5	16.39	16.51	0.68
6	14.5	14.62	0.82

Table (5.2) comparison of friction factor of finned tube based on hydraulic diameter with that of smooth Tube.

Experiment Nos.	Reynolds Number Re_h	Friction Factor		$\frac{F_h}{F_o}$	Increase in Friction factor over smooth Tube.
		Finned Tube F_h	Smooth Tube F_o		
1.	4.42×10^4	0.008	0.0054	1.48	48.15 %
2.	3.64×10^4	0.008	0.0057	1.40	40.35 %
3.	3.33×10^4	0.009	0.0058	1.55	55.17 %
4.	2.72×10^4	0.0096	0.0061	1.57	57.38 %
5.	2.23×10^4	0.01	0.0065	1.54	53.38 %
6.	1.56×10^4	0.014	0.007	2.0	100.0 %

Table (5.3) Comparison of friction factor of finned tube based on inside diameter with that of smooth tube.

Reynolds Number Re_i	Friction Factor of Finned Tube F_i	Friction Factor of Smooth Tube F_o	$\frac{F_i}{F_o}$	Increase in Friction factor over smooth tube
7.86×10^4	0.016	0.0047	3.4	240.4%
6.46×10^4	0.016	0.00496	3.23	220.0%
5.88×10^4	0.018	0.0051	3.53	252.9%
4.77×10^4	0.019	0.0054	3.52	251.9%
3.88×10^4	0.02	0.0056	3.57	257.1%
2.66×10^4	0.028	0.0062	4.52	351.6%

Table (5.4) Estimate of thermal Entrance Length of finned tube.

Reynolds Number, Re_h	Thermal Entrance length, L_t
1.56×10^4	5.2 d
2.23×10^4	5.9 d
3.64×10^4	6.15 d
4.42×10^4	6.25 d

Table (5.5) Comparision of Heat Transfer Co-efficient 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 co-efficient of finned Tube h_{if} w/sq.m ^oC	Heat Transfer co-efficient of smooth tube h_o w/sq.m ^oC	$\frac{h_{if}}{h_o}$
7.86×10^4	135.13	63.67	2.122
6.46×10^4	110.22	54.5	2.02
5.88×10^4	100.60	50.94	1.975
4.77×10^4	88.06	43.37	2.03
3.88×10^4	74.59	36.64	2.04
2.66×10^4	57.36	27.67	2.07

Table (5.6) 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	Nusselt Number for finned tube Nu_{if}	Nusselt Number for smooth tube Nu_o	Percentage increase of Nusselt number over smooth tube
7.862×10^4	350.4	165.07	112.27%
6.46×10^4	285.1	140.99	102.21%
5.88×10^4	258.2	130.77	97.45%
4.77×10^4	224.5	110.56	103.06%
3.88×10^4	189.3	92.98	103.59%
2.66×10^4	143.6	69.25	107.36%

Table (5.7) 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 fin tube h_{if} $w/m^2, {}^oC$	Heat Transfer coefficient for smooth tube at constant pumping power, h_{op}	Criteria, $R_3 = \frac{h_{if}}{h_{op}}$
7.862×10^4	1.19×10^5	135.130	88.923	1.52
6.46×10^4	9.63×10^4	110.216	75.085	1.47
5.88×10^4	9.08×10^4	100.602	72.128	1.395
4.77×10^4	7.36×10^4	88.063	61.381	1.44
3.88×10^4	6.0×10^4	74.588	52.311	1.43
2.66×10^4	4.49×10^4	57.363	42.052	1.36

Table (5.8) Data comprison with previous work.

Ref.	Tube No	$R_3 = h_{if}/h_{op}$ at different Reynolds number Re_o				Remarks
		10^4	$2.5x10^4$	$5x10^4$	10^5	
3*	9	No data	1.67	1.48	1.32	Heating Air
3*	14	No data	0.95	1.11	1.00	Heating Air
9*	14	No data	1.25	1.28	1.28	Cooling Air
9*	24	No data	1.65	1.62	1.53	Cooling Air
This work	-	No data	No data	1.39	1.49	Heating Air

* Obtained from Carnavos (9).

Table (5.9)

Reynolds Number, Re_i	At constant Reynolds Number, $\frac{h_{if}}{h_o}$	At constant pumping power, $\frac{h_{if}}{h_{op}}$
4.77×10^4	2.03	1.44
5.88×10^4	1.98	1.395
6.46×10^4	2.02	1.47
7.86×10^4	2.12	1.52

Table (5.10)

Reynolds number Re_h	$\frac{Nu_f}{P_r^{0.4}}$		Percentage of variation from Carnavos correlation %
	This work	Carnavos correlation	
4.42×10^4	113.233	98.48	14.98
3.64×10^4	92.2	84.33	9.33
3.33×10^4	83.51	78.58	6.27
2.72×10^4	72.63	66.83	8.68
2.23×10^4	61.24	56.99	7.46
1.56×10^4	46.48	42.78	8.65

APPENDIX – D

SAMPLE CALCULATIONS

$$A_x = \frac{\pi D_i^2}{4} = 0.00385 \text{ sq.m.}$$

$$A_{xf} = (\pi D_i^2 / 4) - W H N$$

$$= 0.00358 \text{ sq.m.}$$

$$A_h = \pi D_i + 2 H N = 0.3999 \text{ m}$$

Hydraulic Diameter.

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

$$= 0.03579 \text{ m.}$$

Determination of Mean Velocity.

The velocity was calculated form the relation,

$$\Delta P = \tfrac{1}{2} \rho V^2 \qquad (D - 1)$$

If V is to be m/s, ρ must be expressed in kg/m^3 and ΔP in Pascals (N / m^2). If h is the velocity head expressed in cm of water,

$$\Delta P = 98.1 \times h \quad Pa \qquad (D-2)$$

Standard Atmospheric properties at sea level are

Pressure : 760 mm Hg

Temperature : $15^{\circ}C$

Density, : 1.225 kg / m^3

For any other temperature $t^{\circ}C$ and pressure, b mm Hg, the value of the density in kilograms per cubic meter is

$$= 1.225 \times \frac{288}{273+t} \times \frac{b}{760}$$

$$= 0.4642 \left(\frac{b}{273+t}\right) \quad \ldots \qquad (D-3)$$

Substituting (D - 2) and (D-3) in (D-1),

$$V = 20.56 \left(\frac{273+t}{b}\right)^{1/2} (h)^{1/2} \quad \ldots \qquad (D-4)$$

$$V = C (h)^{1/2} \quad \ldots$$

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

h = Velocity head, cm of water.

Room temp, t = 30.0 °C

Atm.pressure, b = 760 mm Hg.

C = 12.98188

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

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

$$V_i = 18.76 \text{ m / s}$$

$$\text{Mass Flow rate, } M = \rho A_x V$$

$$= 0.08423 \text{ m}^3 / \text{s}$$

$$V = \frac{M}{\rho A_{xf}}$$

$$= 20.62 \text{ m / s}$$

<u>Reynolds Numbers.</u>

$$Re_i = 78615.9$$

$$Re_h = 44190.28$$

Friction Factor:

Local friction factor based on inside diameter is given by,

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

$$= 7.073066 \times 10^{-4} \ (-\Delta P/x) \quad \ldots \quad (D-5)$$

Local friction factor based on hydraulic diameter is given by,

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

$$= 3.61566 \times 10^{-4} \ (-\Delta P/x) \quad \ldots \quad (D-6)$$

x	0.0	0.26	0.46	0.66	0.86	1.06	1.26	1.46
P (x)mm of water	0	50.0	53.0	59.0	65.0	69.0	73.0	77.0
$-\Delta P$ mm of water			3.0	9.0	15.0	19.0	23.0	27.0
$-\Delta P$ in kg/m^2			2.99	8.96	14.93	18.91	22.9	26.88
$-\Delta P/\Delta x$			14.95	22.4	24.89	23.64	22.9	22.4
F_i			0.011	0.016	0.018	0.017	0.016	.016
F_h			0.005	0.008	0.009	0.009	0.008	.008

Heat Transfer Calculation :

$T_i = 30.341\ ^{\circ}C$

$T_o = 43.9\ ^{\circ}C$

Properties of Air are evaluated at 310.14 K

$C_p = 1.00636\ kj / kg\ ^{\circ}C$

$k = 0.026998\ w / m\ ^{\circ}C$

$\nu = 16.704 \times 10^{-6}\ m^2 / s$

$\rho = 1.14152\ kg / m^3.$

Total heat in put to the air,

$Q = M\,C_p\ (T_o - T_i)$

$= 1151.88$ J/s or Watt

$Q' =$ heat input per unit axial length

$= \frac{Q}{L}$

$= 757.82$ W/m.

The local bulk temperature T_b (x) of the fluid is,

$$T_{bx} = T_i + \frac{Q' x}{M\,C_p}$$

$$T_{bx} = 30.341 + 8.94\,x \quad \ldots \quad (D-7)$$

Local heat transfer coefficient is given by,

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

$$= \frac{757.82}{(T_w - T_b)_x \, 0.3999}$$

$$h_x = \frac{1895.0237}{(T_w - T_b)_x} \; W/m^2 \, {}^oC \qquad \ldots \qquad (D-8)$$

Local Nusselt number is,

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

$$= 1.326 \, h_x \qquad \ldots \qquad (D-9)$$

X	0	0.06	0.26	0.46	0.66	0.86	1.06	1.26	1.46	1.52
T_w	39.76	42.0	50.86	59.16	61.71	63.50	64.39	64.25	60.11	58.79
T_{bx}	30.34	30.88	32.67	34.45	36.24	38.03	39.82	41.61	43.39	43.93
$(T_w-T_b)_x$	9.42	11.12	18.20	24.71	25.46	25.47	24.57	22.64	16.72	14.86
h_x	201	170.4	104	76.7	74.4	74.4	77.1	83.7	113.3	127.5
Nu_x	266.6	226	138	101.7	98.7	98.6	102	111	150	169

From the graph of $(T_w - T_b)_x$ vs x / L, Fig (5.10),

$$(T_w - T_b)_{FD} = 25.5\ ^{o}C$$

Fully Developed heat transfer coefficient,

$$h = \frac{Q'}{A_h\ (T_w - T_b)_{FD}} = 74.31 \quad W / sq.\ m\ ^{o}C$$

Fully Developed Nusselt number,

$$Nu = \frac{h\ D_h}{k} = 98.513$$

X/L	0.0	0.04	0.17	0.3	0.43	0.57	0.7	0.83	0.96	1.0
$\frac{(T_w - T_b)_{FD}}{(T_w - T_b)_x}$	2.71	2.29	1.4	1.03	1.0	1.0	1.04	1.13	1.53	1.72
X/D	0.0	0.86	3.71	6.57	9.43	12.29	15.14	18.0	20.86	21.71
$Nu_x/Nu_{F.D}$	2.71	2.29	1.4	1.03	1.0	1.0	1.04	1.13	1.53	1.72

Heat Transfer coefficient based on inside diameter and nominal area,

$$h_{if} = \frac{h\ A_h}{A} = 135.13\ \ W / m^2\ \ ^{o}C$$

Nusselt number based on inside diameter and nominal area,

$$Nu_{if} = \frac{h_{if}\ D_i}{k} = 350.363$$

Constant pumping power Result :

Constant pumping power smooth tube Reynolds number is given by,

$$Re_o = 2.517\ (Re_{if})^{12/11}\ (A_{xf}\ F_{if}/A_{xo})^{4/11}$$
$$= 1.19 \times 10^5$$

For smooth tube,

$$Nu_{op} = 0.023\ (Re_o)^{0.8}\ (Pr)^{0.4}$$
$$= 229.98$$

Heat transfer coefficient for smooth tube at $Re_o = 1.19\text{x}10^5$,

$$h_{op} = 88.7 \quad \text{W/Sq. m C}$$

$$R_3 = \frac{h_{if} \text{ at } Re_{if}}{h_{op} \text{ at } Re_o}$$
$$= 1.52$$

APPENDIX - E

<u>**Calculation of Local Nusselt Number for Smooth Tube in the Entrance Region :**</u>

We consider the temperature profile is developing but the velocity profile is already developed in the turbulent flow of air. The following equation for constant heat rate, discussed at length in kays (18), is directly applicable.

$$Nu_x = \left(\frac{1}{Nu} - \tfrac{1}{2} \sum \frac{\exp(- \gamma_m^2 \, x^+)}{Am \; \gamma_m^4} \right)^{-1} \qquad \text{(E-1)}$$

Where, $$X^+ = \frac{x \, / \, r}{Re \; Pr}$$

For $Pr = 0.7$, $Re = 5 \times 10^4$.

$$x^+ = \frac{x}{1225}$$

$$Nu = 0.023 \; Re^{0.8} \, P_r^{\,0.4}$$

$$= 114.9$$

Eigenvalues and Constants are obtained from kays (18). The local Nusselt number have been calculated from the equation (E-1) and written in the Table below :

X	x / D	x^+	Nu_x
0	0	0	223.3
0.06	0.857	4.9×10^{-5}	177.4
0.26	3.7	2.12×10^{-4}	140.13
0.46	6.57	3.76×10^{-4}	129.7
0.86	12.29	7.02×10^{-4}	121.5
1.26	18.0	1.029×10^{-3}	118.17
1.52	21.71	1.24×10^{-3}	117.02

APPENDIX F

PROGRAM LISTING

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C      MAFIZUL HUQ, STUDENT OF M.SC. ENGG.                                    HE
C      PROGRAM FOR HEAT TRANSFER CALCULATION OF THE THESIS                    HE
C      "STUDY OF HEAT TRANSFER PERFORMANCE OF TUBE HAVING INTERNAL FINS"      HE
C                                                                             HE
C      NOMENCLATURE USED IN COMPUTER PROGRAM            EXPERIMENT NO. 01     HE
C                                                                             HE
C      AX      =CROSS-SECTIONAL AREA OF TUBE WITHOUT FIN., SQ.M               HE
C      AXF     =CROSS-SECTIONAL AREA OF TUBE HAVING INTERNAL FINS.,SQ.M       HE
C      DI      =INSIDE DIAMETER OF TUBE,  M                                   HE
C      DH      =HYDRAULIC DIAMETER OF TUBE, M                                 HE
C      AH      =EFFECTIVE HEAT TRANSFER AREA OF FIN TUBE PER UNIT LENGTH      HE
C              =PAI.DI+(2.*NF*HF), SQ.M                                       HE
C      NF      =NO OF FINS                                                    HE
C      HF      =HEIGHT OF FIN, M                                              HE
C      WF      =THICKNESS OF FIN, M                                           HE
C      ROW     =DENSITY OF AIR AT ROOM TEMP., KG/CU.M                         HE
C      ROWF    =  "           " AT AVERAGE BULK TEMP OF FIN-TUBE, KG/CU.M     HE
C      TR      =ROOM TEMP., C                                                 HE
C      BR      =ATMOPHERIC PRESSURE,  MM OF HG.                               HE
C      P       =STATIC PRESSURE AT AXIAL DISTANCE X,  MM OF WATER             HE
C      H       =VELOCITY HEAD ,MM OF WATER                                    HE
C      FNUE    =KINEMATIC VISCOSITY OF AIR X 10**6., SQ.M/S                   HE
C      ROWF    =DENSITY OF AIR IN FIN TUBE AT AV. BULK TEMP., C               HE
C      ROW     =    "        "      "        AT ROOM TEMP., C                 HE
C      CP      =SP. HEAT OF AIR,KJ/KG*C                                       HE
C      FK      =THERMAL CONDUCTIVITY OF AIR, W/M *C                           HE
C      PR      =PRANDTL NUMBER OF AIR                                         HE
C      V       =MEAN VELOCITY AT INLET TO THE FIN TUBE, M/S                   HE
C      VF      =MEAN VELOCITY IN FIN TUBE, M/S                                HE
C      M       =MASS FLOW RATE, KG/S                                          HE
C      REI     =REYNOLDS NO. BASED ON INSIDE DIAMETER                         HE
C      REH     =REYNOLDS NO. BASED ON HYDRAULIC DIAMETER                      HE
C      RED     =EQUIVALENT REYNOLDS NO. FOR SMOOTH TUBE AT CONSTANT           HE
C               PUMPING POWER                                                 HE
C      X,X1    =AXIAL DISTANCE IN M                                           HE
C      DELH    =PRESSURE DROP AT DIST. X, MM OF WATER                         HE
C      DELP    =PRESSURE DROP AT DIST. X, KG/SQ.M                             HE
C      DPD     =DIMENTIONLESS PRESSURS DROP,DELP/0.5*ROWF*VF**2.              HE
C      DELP/X  =PRESSURE DROP PER UNIT LENGTH AT DIST. X, KG/SQ.M             HE
C      DP      =P(I)-P(I+1),INCREMENTAL PRESSURE DROP                         HE
C      DP/DX   =INCREMENTAL PRESSURE DROP PER UNIT LENGTH,                    HE
C              =(P(I)-P(I+1))/(X(I+1)-X(I))                                   HE
C      DFI     =FRICTION FACTOR AT DISTANCE X(I) BASED ON DI                  HE
C      DFH     =   "        "         "        "      "     DH                HE
C      FI      =FRICTION FACTOR BASED ON INSIDE DIAMETER OF THE TUBE,DI       HE
C      FFI1    =   "       "       "      "     "       "          "    "     HE
C      FH      =   "       "       "      " HYDRAULIC DIAMETER                HE
C      Q       =TOTAL HEAT INPUT TO AIR J/S OR W                              HE
C      QL      =HEAT INPUT PER UNIT LENGTH OF THE TUBE, W/M                   HE
C      TE      =AIR TEMP. AT THE ENTRANCE OF FIN TUBE IN * C                  HE
C      TO      =AIR OUTLET TEMP. IN *C                                        HE
C      TW      =WALL TEMPERATURE IN DEGREE CENTIGRADE                         HE
C      TB      =BULK TEMPERATURE *C                                           HE
C      TBX     =BULK TEMPERATURE AT AXIAL DISTANCE X  *C                      HE
C      FDTB1   =DIFFERENCE BETWEEN WALL TEMP. AND BULK TEMP. IN THERMAL       HE
```

```
C                 FULLY DEVELOPED REGION.
C       HX        =LOCAL HEAT TRANSFER COEFFICIENT AT AXIAL DIST.,X
C       FHE       =HEAT TRANSFER COEFF. IN THERMAL FULLY DEVELOPED REGION
C                  BASED ON EFFECTIVE HEAT TRANSFER AREA.
C       FHI       =HEAT TRANSFER COEF. IN THERMAL FULLY DEVELOPED REGION
C                  BASED ON INSIDE DIAMETER & NOMINAL AREA OF THE TUBE.
C       HO        =HEAT TRANSFER COEFF.FOR SMOOTH TUBE AT CONSTANT
C                  PUMPING POWER EQUIVALENT REYNOLDS NO.,REO
C       NUX       =LOCAL NUSSELT NO. AT AXIAL DIST.,X
C       FNU       =NUSSELT NO IN THERMAL FULLY DEVELOPED REGION.
C       FNUI      =     "            "       "      "          "    BASED ON
C                  INSIDE DIAMETER AND NOMINAL AREA.
C       R3        =CONSTANT PUMING POWER COMPARISON, FHI/HO
C
C
C
      REAL DI,DH,AX,AXF,V,VF,M,REI,REH,BB,FK
      DIMENSION H(10),X1(8),P(8),DELH(8),DELP(8),DELPX(7),FI(7),FH(7),
     +K(10),TW(10),TBX(10),DTB(10),XL(10),XD(10),HX(10),XNU(10),CC(10),
     +RT(10),RNU(10),XDRPH(10),XDR(8),XDRH(8),DP(7),DX(7),DPDX(7),DFI(
     +7),DFH(7),XD1(8),DELP1(8),DPD(8)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA EXNO,AMP,OHM/1.0,14.3,8.75/
      DATA ROW,ROWF/1.1666,1.14152/
      DATA FNUE/16.7040/
      DATA DI,WF,HF,NF/0.07,0.003,0.015,6.0/
      DATA PAI/3.141592654/
      DATA TR,BR/30.0,760.0/
      DATA H/1.473,2.007,2.184,2.235,2.311,2.311,2.286,2.286,2.159,
     *1.727/
      DATA X1/0.0,0.26,0.46,0.66,0.86,1.06,1.26,1.46/
      DATA P/27.0,50.0,53.0,59.0,65.0,69.0,73.0,77.0/
      DATA CP,FK/1.00636,0.026998/
      DATA PR/0.706/
      DATA TE,TO/30.041,43.930/
      DATA X/0.0,0.06,0.26,0.46,0.66,0.86,1.06,1.26,1.46,1.52/
      DATA TW/39.752,42.00,50.860,59.163,61.705,63.500,64.386,64.250,
     +60.114,58.795/
      DATA FDTB1/25.5/
      DATA FFI1/0.016/
      AX=PAI*(DI**2.0)/4.0
      WRITE(3,9)EXNO,AMP,OHM
 9    FORMAT(2X,'EXPERIMENT NO =',1X,F4.1,2X,'CURRENT=',1X,F4.1,1X,'A',
     +2X,'HEATER RESISTANCE =',1X,F5.2,1X,'OHM'///)
      WRITE (3,10)AX
 10   FORMAT(4X,'CROSS-SECTIONAL AREA OF THE TUBE,AX',4X,'=',F9.5,2X,
     *'SQ.M')
C     END
      AXF=AX-WF*HF*NF
      WRITE (3,13)AXF
      DH=4.0*AXF/(PAI*DI+2.0*HF*NF)
      WRITE(3,23)DH
 13   FORMAT(4X,'CROSS-SECTIONAL AREA OF FIN-TUBE,AXF',3X,'=',F9.5,2X,
     *'SQ.M')
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```
 23    FORMAT(4X,'HYDRAULIC DIAMETER,DH',18X,'=',F9.5,2X,'M')          H
       C=20.55*((273+TR)/BR)**0.5                                       H
       N=10.0                                                           H
       V=(C/N)*(H(1)**0.5+H(2)**0.5+H(3)**0.5+H(4)**0.5+H(5)**0.5+H(6)  H
      +**0.5+H(7)**0.5+H(8)**0.5+H(9)**0.5+H(10)**0.5)                 H
       WRITE(3,33)V                                                     H
       WRITE(3,43)C                                                     H
 33    FORMAT(4X,'MEAN VELOCITY,V',24X,'=',F9.5,2X,'M/S')              H
 43    FORMAT(20X,'C',22X,'=',F9.5)                                     H
C      MASS FLOW RATE,M:                                                H
       M=ROW*AX*V                                                       H
       WRITE(3,44)M                                                     H
 44    FORMAT(4X,'MASS FLOW RATE,M',23X,'=',F9.5,2X,'KG/S')            H
C      MEAN VELOCITY IN FIN-TUBE,VF                                     H
       VF=V*AX*ROW/(AXF*ROWF)                                           H
       WRITE(3,45)VF                                                    H
 45    FORMAT(4X,'MEAN VELOCITY IN FIN-TUBE,VF',11X,'=',F9.5,2X,'M/S') H
C      REYNOLDS NUMBER BASED ON INSIDE DIAMETER, REI                    H
C      REYNOLDS NUMBER BASED ON HYDRALIC DIAMETER, REH                  H
C      NUE=NUEF/1000000.0                                               H
       REI=V*DI*1000000.0/FNUF                                          H
       REH=VF*DH*1000000.0/FNUE                                         H
       WRITE(3,46)REI                                                   H
       WRITE(3,47)REH                                                   H
 46    FORMAT(4X,'REYNOLDS NO.BASED ON INSIDE DIA.,REI',3X,'=',F10.2)  H
 47    FORMAT(4X,'REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH','=',F10.2//) H
C      FRICTION FACTOR :                                                H
       WRITE(3,11)                                                      H
 11    FORMAT(2X,'FRICTION FACTOR CALCULATION:')                        H
       WRITE(3,48)(X1(I),I=1,8)                                         H
 48    FORMAT(2X,'AXIAL DIST,X',4X,8F7.2)                               H
       WRITE(3,333)(P(J),J=1,8)                                         H
 333   FORMAT(2X,'P',15X,8F7.2)                                         H
       DO 100 I=2,8                                                     H
       XDR(I)=X1(I)/(DI*REI)                                            H
       XDRH(I)=X1(I)/(DH*REH)                                           H
       DELH(I)=P(I)-P(2)                                                H
       DELP(I)=DELH(I)*0.99548                                          H
 100   CONTINUE                                                         H
       WRITE(3,49)(DELH(I),I=2,8)                                       H
 49    FORMAT(2X,'DELH(MM WATER)',9X,7F7.2)                             H
       WRITE(3,12)(DELP(I),I=2,8)                                       H
 12    FORMAT(2X,'DELP IN KG/SQ.M',8X,7F7.2)                            H
       DO 200 J=3,8                                                     H
       DELPX(J)=DELP(J)/(X1(J)-0.26)                                    H
 200   CONTINUE                                                         H
       WRITE(3,50)(DELPX(J),J=3,8)                                      H
 50    FORMAT(2X,'DELPX',25X,6F7.2)                                     H
       S=ROWF/9.81                                                      H
       S1=2.0*S*VF**2.0                                                 H
       S2=DI/S1                                                         H
       S3=DH/S1                                                         H
C      WRITE(3,55)S,S1,S2,S3                                            H
C55    FORMAT(4X,'S =',F7.3,2X,'S1 =',F7.3,2X,'S2 =',F7.5,2X,F7.5)     H
       DO 300 J=2,8                                                     H
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      FI(J)=DELPX(J)*S2
      FH(J)=DELPX(J)*S3
  300 CONTINUE
      WRITE(3,60)(FI(J),J=3,8)
      WRITE(3,65)(FH(J),J=3,8)
   60 FORMAT(2X,'FI BASED ON DI',16X,6F7.3)
   65 FORMAT(2X,'FH BASED ON DH',16X,6F7.3)
      WRITE(3,211)
  211 FORMAT(5X,'X',5X,'X/DH REH',5X,'X/DI REI',7X,'FH',9X,'FI')
      DO 501 J=3,8
      WRITE(3,213)X1(J),XDRH(J),XDR(J),FH(J),FI(J)
  213 FORMAT(4X,F4.2,3X,E10.3,3X,E10.3,5X,F7.4,4X,F7.4)
  501 CONTINUE
      WRITE(3,215)(X1(I),I=1,8)
  215 FORMAT(//2X,'AXIAL DIST,X',4X,8F7.2)
      WRITE(3,217)(P(J),J=1,8)
  217 FORMAT(2X,'P',15X,8F7.2)
C     CALCULATION OF INCREMENTAL PRESSURE DROP
      DO 502 L=1,7
      K=L+1
      DP(L)=P(K)-P(L)
      DX(L)=X1(K)-X1(L)
      DPDX(L)=DP(L)/DX(L)
      DFI(L)=DPDX(L)*S2
      DFH(L)=DPDX(L)*S3
  502 CONTINUE
      WRITE(3,219)(DP(L),L=1,7)
  219 FORMAT(2X,'INCREMENTAL PRESSURE'/2X,'DROP,  DP(X)',11X,7F7.2)
      WRITE(3,221)(DX(L),L=1,7)
  221 FORMAT(2X,'DX',21X,7F7.2)
      WRITE(3,223)(DPDX(L),L=1,7)
  223 FORMAT(2X,'DP/DX',18X,7F7.2)
      WRITE(3,225)(DFI(L),L=1,7)
      WRITE(3,227)(DFH(L),L=1,7)
  225 FORMAT(2X,'DFI(X)',17X,7F7.3)
  227 FORMAT(2X,'DFH(X)',17X,7F7.3)
C     CALCULATION OF DIMENTIONLESS PRESSURE DROP, DPD
      S4=0.5*S*VFA**2.0
      DO 503 K=1,8
      XD1(K)=X1(K)/DI
      DELP1(K)=P(K)-P(1)
      DPD(K)=DELP1(K)/S4
  503 CONTINUE
      WRITE(3,229)(XD1(K),K=1,8)
  229 FORMAT(//2X,'X/D',11X,8F7.2)
      WRITE(3,230)(DELP1(K),K=1,8)
  230 FORMAT(2X,'DELP1(X)',6X,8F7.2)
      WRITE(3,231)(DPD(K),K=1,8)
  231 FORMAT(2X,'DIMENTION LESS PRESSURE DROP'/1X,'DP/.5*ROW*VFA*2',
     +8F7.2)
C     THERMAL RESULT:
      WRITE(3,67)
   67 FORMAT(///2X,'THERMAL RESULT:'/)
      WRITE(3,69)TE,TO
   69 FORMAT(2X,'AIR INLET TEMP =',2X,F6.3,1X,'C',2X,'AIR OUTLET TEMP.
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     +=',2X,F5.3,1X,' C')
      TF=(TE+TO)/2.
      WRITE(3,73)TF
 73   FORMAT(2X,'PROPERTIES OF AIR EVALUATED AT TF =',2X,F7.3,2X,'C')
C     TOTAL HEAT INPUT TO AIR
      QA=0.0
      QA=QA+M*CP*1000.0*(TO-TE)
      WRITE(3,75)QA
 75   FORMAT(2X,'HEAT INPUT TO AIR',2X,F8.3,2X,'J/S OR WATT')
      QL=QA/1.52
      WRITE(3,80)QL
 80   FORMAT(2X,'HEAT INPUT PER UNIT LENGTH QL =',2X,F7.3,'W/M')
C     LOCAL BULK TEMP.
      C1=0.0
      C1=C1+QL/(M*CP*1000)
C     WRITE(3,81)C1
C81   FORMAT(2X,'C1 =',2X,F7.3)
      DO 400 I=1,10
      TBX(I)=(C1*X(I))+TE
      DTB(I)=TW(I)-TBX(I)
 400  CONTINUE
      WRITE(3,83)(X(I),I=1,10)
 83   FORMAT(2X,'DIST X',4X,10F6.3)
      WRITE(3,85)(TW(I),I=1,10)
 85   FORMAT(2X,'TW',8X,10F6.2)
      WRITE(3,87)(TBX(I),I=1,10)
      WRITE(3,90)(DTB(I),I=1,10)
 87   FORMAT(2X,'TBX',7X,10F6.2)
 90   FORMAT(2X,'(TW-TB)X',3X,10F6.2)
C     LOCAL HEAT TRANSFER COEFFICIENT AND NUSSELT NO.
      AH=PAI*DI+2.*NF*HF
      WRITE(3,92)AH
 92   FORMAT(2X,'EFFECTIVE HEAT TRANSFER AREA AH =',2X,F7.4,2X,'SQ.M')
      BB=0.0
      BB=BB+DH/FK
C     WRITE(3,93)FK
C93   FORMAT(4X,'FK=',2X,F9.6)
C     WRITE(3,94)BB
C94   FORMAT(4X,'BB =',2X,F9.6)
      DO 500 J=1,10
      CC(J)=DTB(J)*AH
      HX(J)=QL/CC(J)
      XNU(J)=BB*HX(J)
      XL(J)=X(J)/1.52
      XD(J)=X(J)/DI
C     XDR(J)=X(J)/(DI*REI)
C     XDRH(J)=X(J)/(DH*REH)
      XDRPH(J)=X(J)/(DH*REH*PR)
 500  CONTINUE
      WRITE(3,95)(XL(J),J=1,10)
 95   FORMAT(2X,'X/L',5X,10F6.2)
      WRITE(3,110)(XD(J),J=1,10)
 110  FORMAT(2X,'X/D',5X,10F6.2)
C     WRITE(3,111)(XDR(J),J=1,8)
C111  FORMAT('X/DR',2X,9E7.2)
```

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C       WRITE(3,112)(XDRH(J),J=1,8)
C112    FORMAT('X/DRH',1X,9E7.2)
C       WRITE(3,113)(XDRPH(J),J=2,10)
C113    FORMAT('X/DRPH',1X,'0.0',1X,9E7.2)
        WRITE(3,120)(HX(J),J=1,10)
 120    FORMAT(2X,'HX',5X,10F6.1)
        WRITE(3,135)(XNU(J),J=1,10)
 135    FORMAT(2X,'NUX',5X,10F6.1)
        WRITE(3,140)
 140    FORMAT(/2X,'NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE
       +WE FIND AXIALLY'/2X,' UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE
       +IN THERMAL FULLY DEVELOPED'/2X,'REGIME'/)
        FDTB=FDTB1
        IF(FDTB.EQ.0.0)GO TO 888
        DO 600 K=1,10
        RT(K)=FDTB/DTB(K)
 600    CONTINUE
        WRITE(3,145)
 145    FORMAT(2X,'(TW-TB)F/')
        WRITE(3,150)(RT(K),K=1,10)
 150    FORMAT(2X,'(TW-TBX)',1X,10F6.2/)
        FHE=QL/(AH*FDTB)
        FNU1=(FHE*DH)/FK
        WRITE(3,153)FHE
 153    FORMAT(/2X,'HEAT TRANSFER COEFF.IN THERMAL'/2X,'FULLY DEVELOPED
       +REGION, FHE =',5X,F8.3,2X,'W/SQ.M C')
        WRITE(3,157)FNU1
 157    FORMAT(/2X,'NUSSELT NO. IN THERMAL FULLY DEV. REGION,NUF =',F8.3/)
C       CACULATION OF HEAT TRANSFER COEFF. AND NUSSELT NO. BASED ON
C       INSIDE DIAMETER AND NOMINAL AREA :
        FHI=(FHE*AH)/(PAI*DI)
        WRITE(3,222)
 222    FORMAT(2X,'HEAT TRANSFER COEFFICIENT BASED ON,')
        WRITE(3,155)FHI
 155    FORMAT(2X,'INSIDE DIA.& NOMIAL AREA,FHI',5X,'=',3X,F8.3,1X,'W/SQ.
       +M C'/)
        FNUI=(FHI*DI)/FK
        WRITE(3,156)FNUI
 156    FORMAT(/2X,'NUSSELT NO. BASED ON INSIDE DIA'/2X,'AND NOMINAL
       +AREA',13X,'=',2X,F8.3/)
C       WRITE(3,157)FNU1
C157    FORMAT(/2X,'NUSSELT NO.IN THERMAL FULLY DEV. REGION,NUF =',F8.3)
        FNU=FNU1
 888    IF(FNU.EQ.0.0)GO TO 999
        DO 700 L=1,10
        RNU(L)=XNU(L)/FNU
 700    CONTINUE
        WRITE(3,311)(XD(L),L=1,10)
 311    FORMAT(//2X,'X/D',6X,10F6.2)
        WRITE(3,160)(RNU(L),L=1,10)
 160    FORMAT(2X,'NUX/FNU',2X,10F6.2//)
C999    WRITE(3,170)
C170    FORMAT(5X,'X',5X,'X/DH REH',5X,'X/DI REI',7X,'FH',9X,'FI')
C       DO 900 J=1,8
C       WRITE(3,175)X1(J),XDPH(J),XDR(J),FH(J),FI(J)
```

```
C175  FORMAT(4X,F4.2,3X,E10.3,3X,E10.3,5X,F7.4,4X,F7.4)
C900  CONTINUE
 999  WRITE(3,180)
 180  FORMAT(4X,'X',7X,'X/DH REH PR',4X,'NUX')
      DO 913 J=1,10
      WRITE(3,185)X(J),XDRPH(J),XNU(J)
 185  FORMAT(3X,F4.2,4X,E11.4,4X,F7.3)
 913  CONTINUE
C     CONSTANT PUMPING POWER COMPARISON: CRITERIA R3
      FFI=FFI1
      IF(FFI.EQ.0.0)GO TO 915
      RED=2.5169*(REI**1.0909)*((AX*FFI/AX)**0.3636)
      WRITE(3,190)RED
 190  FORMAT(//2X,'CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS,RED=',
     +E12.3)
      DNU=0.023*((RED)**0.8)*((PR)**0.4)
      HO=(DNU*FK)/DI
      WRITE(3,194)
 194  FORMAT(2X,'HEAT TRANSFER COEFF.FOR SMOOTH')
      WRITE(3,195)HO
 195  FORMAT(2X,'TUBE AT CONSTANT PUMPING POWER,HO =',2X,F7.3,1X,'W/SQ.
     +M C')
      R3=FHI/HO
      WRITE(3,199)R3
 199  FORMAT(/2X,'CRITERIA R3 = FHI/HO =',2X,F7.3)
 915  STOP
      END
```

EXPERIMENT NO = 1.0 CURRENT= 14.3 A HEATER RESISTANCE = 8.75 OHM

CROSS-SECTIONAL AREA OF THE TUBE,AX	=	0.00385	SQ.M
CROSS-SECTIONAL AREA OF FIN-TUBE,AXF	=	0.00358	SQ.M
HYDRAULIC DIAMETER,DH	=	0.03579	M
MEAN VELOCITY,V	=	18.76009	M/S
C	=	12.98188	
MASS FLOW RATE,M	=	0.08423	KG/S
MEAN VELOCITY IN FIN-TUBE,VF	=	20.61884	M/S
REYNOLDS NO.BASED ON INSIDE DIA.,REI	=	78616.19	
REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH	=	44180.94	

FRICTION FACTOR CALCULATION:

AXIAL DIST,X	0.00	0.26	0.46	0.65	0.86	1.06	1.25	1.46
P	27.00	50.00	53.00	59.00	65.00	69.00	73.00	77.00
DELH(MM WATER)		0.00	3.00	9.00	15.00	19.00	23.00	27.00
DELP IN KG/SQ.M		0.00	2.99	8.96	14.93	18.91	22.90	26.88
DELPX			14.93	22.40	24.89	23.64	22.90	22.40
FI BASED ON DI			0.011	0.016	0.018	0.017	0.016	0.016
FH BASED ON DH			0.005	0.008	0.009	0.009	0.008	0.008

X	X/DH REH	X/DI REI	FH	FI
0.46	0.291E-03	0.836E-04	0.0054	0.0106
0.65	0.417E-03	0.120E-03	0.0081	0.0158
0.85	0.544E-03	0.156E-03	0.0090	0.0176
1.05	0.670E-03	0.193E-03	0.0086	0.0167
1.25	0.797E-03	0.229E-03	0.0083	0.0162
1.46	0.923E-03	0.265E-03	0.0081	0.0158

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.86	1.06	1.26	1.46
P	27.00	50.00	53.00	59.00	65.00	69.00	73.00	77.00
INCREMENTAL PRESSURE								
DROP, DP(X)		23.00	3.00	6.00	6.00	4.00	4.00	4.00
DX		0.26	0.20	0.20	0.20	0.20	0.20	0.20
DP/DX		88.46	15.00	30.00	30.00	20.00	20.00	20.00
DFI(X)		0.063	0.011	0.021	0.021	0.014	0.014	0.014
DFH(X)		0.032	0.005	0.011	0.011	0.007	0.007	0.007

X/D	0.00	3.71	6.57	9.43	12.29	15.14	18.00	20.86
DELP1(X)	0.00	23.00	26.00	32.00	38.00	42.00	46.00	50.00
DIMENTION LESS PRESSURE DROP								
DP/.5*ROW*VF**2	0.00	0.93	1.05	1.29	1.54	1.70	1.86	2.02

THERMAL RESULT:

AIR INLET TEMP = 30.341 C AIR OUTLET TEMP. = 43.930 C
PROPERTIES OF AIR EVALUATED AT TF = 37.135 C
HEAT INPUT TO AIR 1151.816 J/S OR WATT
HEAT INPUT PER UNIT LENGTH QL = 757.774W/M

DIST X	0.000	0.060	0.260	0.460	0.660	0.860	1.060	1.260	1.460	1.520
TW	39.76	42.00	50.86	59.16	61.71	63.50	64.39	64.25	60.11	58.79
TBX	30.34	30.88	32.67	34.45	36.24	38.03	39.82	41.61	43.39	43.93
(TW-TB)X	9.42	11.12	18.19	24.71	25.46	25.47	24.57	22.64	16.72	14.87

EFFECTIVE HEAT TRANSFER AREA AH = 0.3999 SQ.M

X/L	0.00	0.04	0.17	0.30	0.43	0.57	0.70	0.83	0.96	1.00
X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
HX	201.1	170.4	104.1	76.7	74.4	74.4	77.1	83.7	113.3	127.5
NUX	266.6	225.9	138.1	101.7	98.7	98.6	102.2	110.9	150.2	169.0

NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE WE FIND AXIALLY UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE IN THERMAL FULLY DEVELOPED REGIME

(TW-TB)F/ (TW-TBX)	2.71	2.29	1.40	1.03	1.00	1.00	1.04	1.13	1.53	1.72

HEAT TRANSFER COEFF. IN THERMAL
FULLY DEVELOPED REGION, FHF = 74.308 W/SQ.M C

NUSSELT NO. IN THERMAL FULLY DEV. REGION, NUF = 98.513

HEAT TRANSFER COEFFICIENT BASED ON
INSIDE DIA.& NOMIAL AREA, FHI = 135.130 W/SQ. M C

NUSSELT NO. BASED ON INSIDE DIA
AND NOMINAL AREA = 350.363

X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
NUX/FNU	2.71	2.29	1.40	1.03	1.00	1.00	1.04	1.13	1.53	1.72

X	X/DH REH PR	NUX
0.00	0.0000E+00	265.648
0.06	0.5374E-04	225.859
0.26	0.2329E-03	138.068
0.46	0.4120E-03	101.665
0.66	0.5912E-03	98.654
0.86	0.7703E-03	98.627
1.06	0.9495E-03	102.248
1.26	0.1129E-02	110.936
1.46	0.1308E-02	150.241
1.52	0.1361E-02	168.993

CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS, REO= 0.119E+06
HEAT TRANSFER COEFF. FOR SMOOTH
TUBE AT CONSTANT PUMPING POWER, HO = 88.923 W/SQ. M C

CRITERIA R3 = FHI/HO = 1.520

DATA FOR EXPT NO 2

```
C
      REAL DI,DH,AX,AXF,V,VF,M,REI,REH,BB,FK
      DIMENSION H(10),X1(8),P(8),DELH(8),DELP(8),DELPX(7),FI(7),FH(7),
     +X(10),TW(10),TBX(10),DTB(10),XL(10),XD(10),HX(10),XNU(10),CC(10),
     +RT(10),RNU(10),XDRPH(10),XDR(8),XDRH(8),DP(7),DX(7),DPDX(7),DFI(
     +7),DFH(7),XD1(8),DELP1(8),DPD(8)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA EXNO,AMP,OHM/2.0,14.3,8.75/
      DATA ROW,ROWF/1.1666,1.13854/
      DATA FNJE/16.7832/
      DATA DI,WF,HF,NF/0.07,0.003,0.015,5.0/
      DATA PAI/3.141592654/
      DATA TR,BR/30.0,760.0/
      DATA H/1.092,1.447,1.512,1.549,1.549,1.549,1.549,1.512,1.447,
     *1.092/
      DATA X1/0.0,0.26,0.46,0.66,0.86,1.06,1.26,1.46/
      DATA P/19.0,37.5,40.0,44.0,47.0,50.0,53.0,56.0/
      DATA CP,FK/1.00540,0.027050/
      DATA PR/0.706/
      DATA TE,TO/30.804,46.857/
      DATA X/0.0,0.06,0.26,0.46,0.66,0.86,1.06,1.26,1.46,1.52/
      DATA TW/42.860,45.51,55.750,64.409,68.000,69.636,71.273,71.159,
     +66.114,64.136/
      DATA FOTB1/30.5/
      DATA FFI1/0.016/
```

EXPERIMENT NO = 6.0 CURRENT= 14.3 A HEATER RESISTANCE = 8.75 OHM

CROSS-SECTIONAL AREA OF THE TUBE,AX	=	0.00385	SQ.M
CROSS-SECTIONAL AREA OF FIN-TUBE,AXF	=	0.00358	SQ.M
HYDRAULIC DIAMETER,DH	=	0.03579	M
MEAN VELOCITY,V	=	15.49182	M/S
C	=	12.98158	
MASS FLOW RATE,M	=	0.06955	KG/S
MEAN VELOCITY IN FIN-TUBE,VF	=	17.07132	M/S
REYNOLDS NO.BASED ON INSIDE DIA.,REI	=	64594.56	
REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH	=	36396.06	

FRICTION FACTOR CALCULATION:

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.86	1.06	1.26	1.46
P	19.00	37.50	40.00	44.00	47.00	50.00	53.00	56.00
DELH(MM WATER)		0.00	2.50	6.50	9.50	12.50	15.50	18.50
DELP IN KG/SQ.M		0.00	2.49	6.47	9.45	12.44	15.43	18.42
DELPX			12.44	16.18	15.76	15.55	15.43	15.35
FI BASED ON DI			0.013	0.017	0.016	0.016	0.016	0.016
FH BASED ON DH			0.007	0.009	0.008	0.008	0.008	0.008

X	X/DH REH	X/DI REI	FH	FI
0.46	0.353E-03	0.102E-03	0.0065	0.0129
0.66	0.507E-03	0.146E-03	0.0086	0.0167
0.86	0.660E-03	0.190E-03	0.0083	0.0163
1.06	0.814E-03	0.234E-03	0.0082	0.0161
1.26	0.967E-03	0.279E-03	0.0082	0.0160
1.46	0.112E-02	0.323E-03	0.0081	0.0159

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.86	1.06	1.26	1.46
P	19.00	37.50	40.00	44.00	47.00	50.00	53.00	56.00
INCREMENTAL PRESSURE								
DROP, DP(X)		18.50	2.50	4.00	3.00	3.00	3.00	3.00
DX		0.26	0.20	0.20	0.20	0.20	0.20	0.20
DP/DX		71.15	12.50	20.00	15.00	15.00	15.00	15.00
DFI(X)		0.074	0.013	0.021	0.016	0.016	0.016	0.016
DFH(X)		0.038	0.007	0.011	0.008	0.008	0.008	0.008

X/D	0.00	3.71	6.57	9.43	12.29	15.14	18.00	20.86
DELP1(X)	0.00	18.50	21.00	25.00	28.00	31.00	34.00	37.00
DIMENTION LESS PRESSURE DROP								
DP/.5*ROW VF**2	0.00	1.09	1.24	1.48	1.66	1.83	2.01	2.19

THERMAL RESULT:

AIR INLET TEMP = 30.604 C AIR OUTLET TEMP. = 46.857 C
PROPERTIES OF AIR EVALUATED AT TF = 38.330 C
HEAT INPUT TO AIR 1123.664 J/S OR WATT
HEAT INPUT PER UNIT LENGTH QL = 739.252W/M

DIST X	0.000	0.060	0.260	0.460	0.660	0.860	1.060	1.260	1.460	1.520
TW	42.86	45.51	55.75	64.41	68.00	69.64	71.27	71.16	66.11	64.14
TBX	30.80	31.44	33.55	35.66	37.77	39.89	42.00	44.11	46.22	46.86
(TW-TB)X	12.06	14.07	22.20	28.75	30.23	29.75	29.27	27.05	19.89	17.28

EFFECTIVE HEAT TRANSFER AREA AH = 0.3999 SQ.M

X/L	0.00	0.04	0.17	0.30	0.43	0.57	0.70	0.83	0.96	1.00
X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
HX	153.3	131.4	83.3	64.3	61.2	62.1	63.1	68.3	92.9	107.0
NUX	202.8	173.8	110.1	85.1	80.9	82.2	83.5	90.4	122.9	141.5

NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE WE FIND AXIALLY UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE IN THERMAL FULLY DEVELOPED REGIME

(TW-TB)F/ (TW-TBX)	2.53	2.17	1.37	1.06	1.01	1.03	1.04	1.13	1.53	1.77

HEAT TRANSFER COEFF.IN THERMAL
FULLY DEVELOPED REGION, FHE = 60.608 W/SQ.M C

NUSSELT NO. IN THERMAL FULLY DEV. REGION,NUF = 80.165

HEAT TRANSFER COEFFICIENT BASED ON
INSIDE DIA.& NOMIAL AREA,FHI = 110.216 W/SQ. M C

NUSSELT NO. BASED ON INSIDE DIA
AND NOMINAL AREA = 285.112

X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
NUX/FNU	2.53	2.17	1.37	1.06	1.01	1.03	1.04	1.13	1.53	1.77

X	X/DH REH PR	NUX
0.00	0.0000E+00	202.810
0.06	0.6524E-04	173.751
0.26	0.2527E-03	110.138
0.46	0.5002E-03	85.055
0.66	0.7176E-03	80.894
0.86	0.9351E-03	82.189
1.06	0.1153E-02	83.523
1.26	0.1370E-02	90.398
1.46	0.1587E-02	122.926
1.52	0.1653E-02	141.505

CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS,RED= 0.963E+05
HEAT TRANSFER COEFF.FOR SMOOTH
TUBE AT CONSTANT PUMPING POWER,HO = 75.085 W/SQ. M C

CRITERIA R3 = FHI/HO = 1.468

DATA FOR EXPT NO 3

```
C
C
      REAL DI,DH,AX,AXF,V,VF,M,REI,REH,BB,FK
      DIMENSION H(10),X1(8),P(8),DELH(8),DELP(8),DELPX(7),FI(7),FH(7),
     +X(10),TW(10),TBX(10),DTB(10),XL(10),XD(10),HX(10),XNU(10),CC(10),
     +RT(10),RNU(10),XDRPH(10),XDR(8),XDRH(8),DP(7),DX(7),DPDX(7),DFI(
     +7),DFH(7),XD1(8),DELP1(8),DPD(8)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA EXNO,AMP,OHM/3.0,14.3,8.75/
      DATA ROW,ROWF/1.1623,1.12853/
      DATA FNUE/17.0711/
      DATA DI,WF,HF,NF/0.07,0.003,0.015,6.0/
      DATA PAI/3.141592654/
      DATA TR,BR/31.2,760.0/
      DATA H/0.965,1.219,1.283,1.295,1.321,1.321.1.321,1.295,1.219,
     "0.955/
      DATA X1/0.0,0.25,0.46,0.66,0.85,1.05,1.26,1.46/
      DATA P/15.0,25.0,27.0,31.0,34.0,37.0,40.0,42.0/
      DATA CP,FK/1.00659,0.027270/
      DATA PR/0.705/
      DATA TE,TO/31.540,49.600/
      DATA X/0.0,0.06,0.26,0.45,0.66,0.85,1.05,1.26,1.45,1.52/
      DATA TW/49.980,52.90,64.140,71.520,72.730,76.810,79.000,78.890,
     +72.470,70.090/
      DATA FDTB1/34.5/
      DATA FFI1/0.019/
```

EXPERIMENT NO = 3.0 CURRENT= 14.3 A HEATER RESISTANCE = 8.75 OHM

CROSS-SECTIONAL AREA OF THE TUBE,AX	= 0.00385	SQ.M
CROSS-SECTIONAL AREA OF FIN-TUBE,AXF	= 0.00358	SQ.M
HYDRAULIC DIAMETER,DH	= 0.03579	M
MEAN VELOCITY,V	= 14.34675	M/S
C	= 13.00755	
MASS FLOW RATE,M	= 0.06417	KG/S
MEAN VELOCITY IN FIN-TUBE,VF	= 15.89095	M/S
REYNOLDS NO.BASED ON INSIDE DIA.,REI	= 58823.73	
REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH	= 33318.05	

FRICTION FACTOR CALCULATION:

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.85	1.05	1.25	1.46
P	15.00	25.00	27.00	31.00	34.00	37.00	40.00	42.00
DELH(MM WATER)		0.00	2.00	6.00	9.00	12.00	15.00	17.00
DELP IN KG/SQ.M		0.00	1.99	5.97	8.96	11.95	14.93	16.92
DELPX			9.95	14.93	14.93	14.93	14.93	14.10
FI BASED ON DI			0.012	0.018	0.018	0.018	0.018	0.017
FH BASED ON DH			0.006	0.009	0.009	0.009	0.009	0.009

X	X/DH REH	X/DI REI	FH	FI
0.46	0.386E-03	0.112E-03	0.0061	0.0120
0.66	0.553E-03	0.160E-03	0.0092	0.0180
0.86	0.721E-03	0.209E-03	0.0092	0.0180
1.06	0.889E-03	0.257E-03	0.0092	0.0180
1.26	0.106E-02	0.306E-03	0.0092	0.0180
1.46	0.122E-02	0.355E-03	0.0087	0.0170

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.86	1.06	1.25	1.46
P	15.00	25.00	27.00	31.00	34.00	37.00	40.00	42.00
INCREMENTAL PRESSURE								
DROP, DP(X)		10.00	2.00	4.00	3.00	3.00	3.00	2.00
DX		0.26	0.20	0.20	0.20	0.20	0.20	0.20
DP/DX		38.46	10.00	20.00	15.00	15.00	15.00	10.00
DFI(X)		0.046	0.012	0.024	0.018	0.018	0.018	0.012
DFH(X)		0.024	0.006	0.012	0.009	0.009	0.009	0.006

X/D	0.00	3.71	6.57	9.43	12.29	15.14	18.00	20.86
DELPI(X)	0.00	10.00	12.00	16.00	19.00	22.00	25.00	27.00
DIMENTION LESS PRESSURE DROP								
DP/.5 RON VF 2	0.00	0.69	0.83	1.10	1.31	1.51	1.72	1.86

THERMAL RESULT:

AIR INLET TEMP = 31.640 C AIR OUTLET TEMP. = 49.600 C
PROPERTIES OF AIR EVALUATED AT TF = 40.620 C
HEAT INPUT TO AIR 1160.156 J/S OR WATT
HEAT INPUT PER UNIT LENGTH QL = 763.260W/M

```
DIST X      0.000 0.060 0.250 0.450 0.650 0.860 1.050 1.250 1.460 1.520
TW          49.98 52.90 64.14 71.52 72.73 76.81 79.00 78.89 72.47 70.09
TBX         31.64 32.35 34.71 37.08 39.44 41.80 44.15 46.53 48.89 49.60
(TW-TB)X    18.34 20.55 29.43 34.44 33.29 35.01 34.84 32.36 23.58 20.49
EFFECTIVE HEAT TRANSFER AREA AH =    0.3999  SQ.M
X/L         0.00  0.04  0.17  0.30  0.43  0.57  0.70  0.83  0.96  1.00
X/D         0.00  0.86  3.71  6.57  9.43 12.29 15.14 18.00 20.86 21.71
HX         104.1  92.9  64.9  55.4  57.3  54.5  54.8  59.0  80.9  93.1
NUX        136.6 121.9  85.1  72.7  75.2  71.6  71.9  77.4 106.2 122.3

NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE    WE FIND AXIALLY
 UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE    IN THERMAL FULLY DEVELOPED
REGIME

(TW-TB)F/
(TW-TBX)    1.88  1.68  1.17  1.00  1.04  0.99  0.99  1.07  1.46  1.68


HEAT TRANSFER COEFF.IN THERMAL
FULLY DEVELOPED   REGION, FHE =        55.321  W/SQ.M C

NUSSELT NO. IN THERMAL FULLY DEV. REGION,NUF =  72.510

HEAT TRANSFER COEFFICIENT BASED ON
INSIDE DIA.& NOMIAL AREA,FHI      =    100.602 W/SQ. M C


NUSSELT NO. BASED ON INSIDE DIA
AND NOMINAL       AREA            =    258.237


X/D          0.00  0.86  3.71  6.57  9.43 12.29 15.14 18.00 20.86 21.71
NUX/FNU      1.88  1.68  1.17  1.00  1.04  0.99  0.99  1.07  1.46  1.68

  X        X/DH REH PR     NUX
 0.00     0.0000E+00     136.589
 0.06     0.7137E-04     121.893
 0.25     0.3093E-03      85.125
 0.46     0.5471E-03      72.726
 0.65     0.7850E-03      75.245
 0.86     0.1023E-02      71.555
 1.05     0.1261E-02      71.911
 1.26     0.1499E-02      77.407
 1.46     0.1737E-02     106.240
 1.52     0.1808E-02     122.257


CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS,REO=    0.908E+05
HEAT TRANSFER COEFF.FOR SMOOTH
TUBE AT CONSTANT PUMPING POWER,HO =    72.128 W/SQ. M C

CRITERIA R3 = FHI/HO =     1.395
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DATA FOR EXPT NO 4

```
C
C
C
      REAL DI,DH,AX,AXF,V,VF,M,REI,REH,BB,FK
      DIMENSION H(10),X1(8),P(8),DELH(8),DELP(8),DELPX(7),FI(7),FH(7),
     +X(10),TW(10),TBX(10),DTB(10),XL(10),XD(10),HX(10),XNU(10),CC(10),
     +RT(10),RNU(10),XDRPH(10),XDR(8),XDRH(8),DP(7),DX(7),DPDX(7),DFI(
     +7),DFH(7),XD1(8),DELP1(8),DPD(8)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA EXNO,AMP,OHM/4.0,14.3,8.75/
      DATA ROW,ROWF/1.1523,1.11987/
      DATA FNJE/17.3159/
      DATA DI,WF,HF,NF/0.07,0.003,0.015,6.0/
      DATA PAI/3.141592654/
      DATA TR,BR/31.2,760.0/
      DATA H/0.635,0.813,0.876,0.889,0.914,0.914,0.889,0.876,0.813,
     "0.635/
      DATA X1/0.0,0.26,0.46,0.66,0.86,1.06,1.26,1.46/
      DATA P/10.5,19.5,20.0,23.5,26.0,26.0,30.0,32.0/
      DATA CP,FK/1.00575,0.027460/
      DATA PR/0.704/
      DATA TE,TO/31.950,54.120/
      DATA X/0.0,0.06,0.26,0.46,0.66,0.86,1.06,1.26,1.46,1.52/
      DATA TW/53.600,56.57,69.930,78.630,79.930,84.510,87.760,87.810,
     +79.800,77.110/
      DATA FDT31/40.0/
      DATA FFI1/0.019/
```

EXPERIMENT NO = 4.0 CURRENT= 14.3 A HEATER RESISTANCE = 8.75 OHM

CROSS-SECTIONAL AREA OF THE TUBE,AX	=	0.00385	SQ.M
CROSS-SECTIONAL AREA OF FIN-TUBE,AXF	=	0.00358	SQ.M
HYDRAULIC DIAMETER,DH	=	0.03579	M
MEAN VELOCITY,V	=	11.79354	M/S
C	=	13.00756	
MASS FLOW RATE,M	=	0.05275	KG/S
MEAN VELOCITY IN FIN-TUBE,VF	=	13.16405	M/S
REYNOLDS NO.BASED ON INSIDE DIA.,REI	=	47676.11	
REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH	=	27210.45	

FRICTION FACTOR CALCULATION:

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.85	1.06	1.26	1.46
P	10.50	19.50	20.00	23.50	26.00	28.00	30.00	32.00
DELH(MM WATER)		0.00	0.50	4.00	6.50	8.50	10.50	12.50
DELP IN KG/SQ.M		0.00	0.50	3.98	6.47	8.46	10.45	12.44
DELPX			2.49	9.95	10.78	10.58	10.45	10.37
FI BASED ON DI			0.004	0.018	0.019	0.019	0.018	0.018
FH BASED ON DH			0.002	0.009	0.010	0.010	0.009	0.009

X	X/DH REH	X/DI REI	FH	FI
0.46	0.472E-03	0.138E-03	0.0023	0.0044
0.66	0.673E-03	0.198E-03	0.0090	0.0176
0.86	0.883E-03	0.258E-03	0.0098	0.0191
1.06	0.109E-02	0.318E-03	0.0096	0.0187
1.26	0.129E-02	0.378E-03	0.0095	0.0185
1.46	0.150E-02	0.437E-03	0.0094	0.0183

AXIAL DIST,X	0.00	0.26	0.46	0.65	0.86	1.06	1.25	1.46
P	10.50	19.50	20.00	23.50	26.00	28.00	30.00	32.00
INCREMENTAL PRESSURE								
DROP, DP(X)		9.00	0.50	3.50	2.50	2.00	2.00	2.00
DX		0.26	0.20	0.20	0.20	0.20	0.20	0.20
DP/DX		34.62	2.50	17.50	12.50	10.00	10.00	10.00
DFI(X)		0.061	0.004	0.031	0.022	0.018	0.018	0.018
DFH(X)		0.031	0.002	0.015	0.011	0.009	0.009	0.009

X/D	0.00	3.71	6.57	9.43	12.29	15.14	18.00	20.86
DELPI(X)	0.00	9.00	9.50	13.00	15.50	17.50	19.50	21.50
DIMENTION LESS PRESSURE DROP								
DP/.5*RHO*VF**2	0.00	0.91	0.96	1.31	1.57	1.77	1.97	2.17

THERMAL RESULT:

AIR INLET TEMP = 31.950 C AIR OUTLET TEMP. = 54.120 C
PROPERTIES OF AIR EVALUATED AT TF = 43.035 C
HEAT INPUT TO AIR 1177.452 J/S OR WATT
HEAT INPUT PER UNIT LENGTH QL = 774.639W/M

DIST X	0.000	0.060	0.260	0.460	0.660	0.860	1.060	1.260	1.460	1.520
TW	53.60	56.57	69.98	78.63	79.93	84.51	87.76	87.81	79.80	77.11
TBX	31.95	32.83	35.74	38.66	41.58	44.49	47.41	50.33	53.24	54.12
(TW-TB)X	21.65	23.74	34.24	39.97	38.35	40.02	40.35	37.48	26.56	22.99

EFFECTIVE HEAT TRANSFER AREA AH = 0.3999 SQ.M

X/L	0.00	0.04	0.17	0.30	0.43	0.57	0.70	0.83	0.96	1.00
X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
HX	89.5	81.6	56.6	48.5	50.5	48.4	48.0	51.7	72.9	84.3
NUX	116.6	106.3	73.7	63.2	65.8	63.1	62.6	67.4	95.1	109.8

NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE WE FIND AXIALLY UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE IN THERMAL FULLY DEVELOPED REGIME

(TW-TB)F/

(TW-TBX)	1.85	1.68	1.17	1.00	1.04	1.00	0.99	1.07	1.51	1.74

HEAT TRANSFER COEFF.IN THERMAL FULLY DEVELOPED REGION, FHE = 48.426 W/SQ.M C

NUSSELT NO. IN THERMAL FULLY DEV. REGION,NUF = 63.120

HEAT TRANSFER COEFFICIENT BASED ON INSIDE DIA.& NOMIAL AREA,FHI = 88.063 W/SQ. M C

NUSSELT NO. BASED ON INSIDE DIA AND NOMINAL AREA = 224.486

X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
NUX/FNU	1.85	1.68	1.17	1.00	1.04	1.00	0.99	1.07	1.51	1.74

X	X/DH REH PR	NUX
0.00	0.0000E+00	116.619
0.06	0.8751E-04	106.330
0.26	0.3792E-03	73.743
0.46	0.6709E-03	63.166
0.66	0.9626E-03	65.829
0.86	0.1254E-02	63.094
1.06	0.1546E-02	62.573
1.26	0.1838E-02	67.360
1.46	0.2129E-02	95.077
1.52	0.2217E-02	109.821

CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS,REO= 0.736E+05
HEAT TRANSFER COEFF.FOR SMOOTH
TUBE AT CONSTANT PUMPING POWER,HO = 61.381 W/SQ. M C

CRITERIA R3 = FHI/HO = 1.435

DATA FOR EXPT NO 5

```
C
C
C
      REAL DI,DH,AX,AXF,V,VF,M,REI,REH,BB,FK
      DIMENSION H(10),X1(8),P(8),DELH(8),DELP(8),DELPX(7),FI(7),FH(7),
     +X(10),TW(10),TBX(10),DTB(10),XL(10),XD(10),HX(10),XNU(10),CC(10),
     +RT(10),RNU(10),XDRPH(10),XDR(8),XDRH(8),DP(7),DX(7),DPDX(7),DFI(
     +7),DFH(7),XD1(8),DELPI(8),DPD(8)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA EXNO,AMP,OHM/5.0,14.3,8.75/
      DATA ROW,ROWF/1.1638,1.11386/
      DATA FNUE/17.4909/
      DATA DI,WF,HF,NF/0.07,0.003,0.015,6.0/
      DATA PAI/3.141592654/
      DATA TR,BR/30.8,760.0/
      DATA H/0.381,0.559,0.609,0.622,0.635,0.635,0.622,0.609,0.559,
     "0.381/
      DATA X1/0.0,0.26,0.46,0.66,0.86,1.06,1.26,1.46/
      DATA P/08.5,14.5,15.5,17.5,19.5,21.0,22.0,23.0/
      DATA CP,FK/1.00587,0.027586/
      DATA PR/0.704/
      DATA TE,TO/32.220,57.300/
      DATA X/0.0,0.06,0.26,0.46,0.66,0.86,1.06,1.26,1.46,1.52/
      DATA TW/54.120,57.65,70.960,83.870,88.300,90.570,93.040,92.950,
     +85.480,83.180/
      DATA FDTB1/44.0/
      DATA FFI1/0.020/
```

EXPERIMENT NO = 5.0 CURRENT= 14.3 A HEATER RESISTANCE = 8.75 OHM

CROSS-SECTIONAL AREA OF THE TUBE,AX	=	0.00385	SQ.M
CROSS-SECTIONAL AREA OF FIN-TUBE,AXF	=	0.00358	SQ.M
HYDRAULIC DIAMETER,DH	=	0.03579	M
MEAN VELOCITY,V	=	9.69943	M/S
C	=	12.99900	
MASS FLOW RATE,M	=	0.04344	KG/S
MEAN VELOCITY IN FIN-TUBE,VF	=	10.89895	M/S
REYNOLDS NO.BASED ON INSIDE DIA.,REI	=	38817.88	
REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH	=	22303.03	

FRICTION FACTOR CALCULATION:

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.86	1.06	1.26	1.46
P	8.50	14.50	15.50	17.50	19.50	21.00	22.00	23.00
DELH(MM WATER)		0.00	1.00	3.00	5.00	6.50	7.50	8.50
DELP IN KG/SQ.M		0.00	1.00	2.99	4.98	6.47	7.47	8.46
DELPX			4.98	7.47	8.30	8.09	7.47	7.05
FI BASED ON DI			0.013	0.019	0.022	0.021	0.019	0.018
FH BASED ON DH			0.007	0.010	0.011	0.011	0.010	0.009

X	X/DH REH	X/DI REI	FH	FI
0.46	0.576E-03	0.159E-03	0.0066	0.0129
0.66	0.827E-03	0.243E-03	0.0099	0.0194
0.86	0.108E-02	0.316E-03	0.0110	0.0215
1.06	0.133E-02	0.390E-03	0.0107	0.0210
1.26	0.158E-02	0.464E-03	0.0099	0.0194
1.46	0.183E-02	0.537E-03	0.0094	0.0183

AXIAL DIST,X	0.00	0.26	0.46	0.66	0.86	1.06	1.26	1.46
P	8.50	14.50	15.50	17.50	19.50	21.00	22.00	23.00
INCREMENTAL PRESSURE								
DROP, DP(X)		6.00	1.00	2.00	2.00	1.50	1.00	1.00
DX		0.26	0.20	0.20	0.20	0.20	0.20	0.20
DP/DX		23.08	5.00	10.00	10.00	7.50	5.00	5.00
DFI(X)		0.060	0.013	0.026	0.026	0.019	0.013	0.013
DFH(X)		0.031	0.007	0.013	0.013	0.010	0.007	0.007

X/D	0.00	3.71	6.57	9.43	12.29	15.14	18.00	20.86
DELP1(X)	0.00	6.00	7.00	9.00	11.00	12.50	13.50	14.50
DIMENTION LESS PRESSURE DROP								
DP/.5*RDW*VF**2	0.00	0.39	1.04	1.33	1.63	1.85	2.00	2.15

THERMAL RESULT:

AIR INLET TEMP = 32.220 C AIR OUTLET TEMP. = 57.300 C
PROPERTIES OF AIR EVALUATED AT TF = 44.760 C
HEAT INPUT TO AIR 1097.012 J/S OR WATT
HEAT INPUT PER UNIT LENGTH QL = 721.718W/M

DIST X	0.000	0.060	0.250	0.460	0.660	0.860	1.050	1.250	1.460	1.520
TW	54.12	57.65	70.96	83.37	88.30	90.67	93.04	92.98	85.48	83.18
TBX	32.22	33.21	36.51	39.81	43.11	46.41	49.71	53.01	56.31	57.30
(TW-TB)X	21.90	24.44	34.45	44.06	45.19	44.26	43.33	39.97	29.17	25.88

EFFECTIVE HEAT TRANSFER AREA AH = 0.3999 SQ.M

X/L	0.00	0.04	0.17	0.30	0.43	0.57	0.70	0.83	0.96	1.00
X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
HX	82.4	73.8	52.4	41.0	39.9	40.8	41.7	45.2	61.9	69.7
NUX	106.9	95.8	68.0	53.1	51.5	52.9	54.0	58.6	80.3	90.5

NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE WE FIND AXIALLY UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE IN THERMAL FULLY DEVELOPED REGIME

(TW-TB)F/ (TW-TBX)	2.01	1.80	1.28	1.00	0.97	0.99	1.02	1.10	1.51	1.70

HEAT TRANSFER COEFF.IN THERMAL
FULLY DEVELOPED REGION, FHE = 41.016 W/SQ.M C

NUSSELT NO. IN THERMAL FULLY DEV. REGION,NUF = 53.217

HEAT TRANSFER COEFFICIENT BASED ON
INSIDE DIA.& NOMIAL AREA,FHI = 74.588 W/SQ. M C

NUSSELT NO. BASED ON INSIDE DIA
AND NOMINAL AREA = 189.268

X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
NUX/FNU	2.01	1.80	1.28	1.00	0.97	0.99	1.02	1.10	1.51	1.70

X	X/DH REH PR	NUX
0.00	0.0000E+00	106.921
0.06	0.1068E-03	95.809
0.25	0.4626E-03	67.970
0.46	0.8185E-03	53.145
0.66	0.1174E-02	51.816
0.86	0.1530E-02	52.905
1.05	0.1886E-02	54.040
1.25	0.2242E-02	58.583
1.45	0.2598E-02	80.273
1.52	0.2705E-02	90.478

CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS,RED= 0.600E+05
HEAT TRANSFER COEFF.FOR SMOOTH
TUBE AT CONSTANT PUMPING POWER,HD = 52.311 W/SQ. M C

CRITERIA R3 = FHI/HD = 1.426

DATA FOR EXPT NO 6

```
[illegible]
[illegible]
[illegible]
      REAL DI,DH,AX,AXF,V,VF,M,REI,REH,BR,FK
      DIMENSION H(10),X1(8),P(8),DELH(8),DELP(8),DELPX(7),FI(7),FH(7),
     +X(10),TW(10),TBX(10),DTB(10),XL(10),XD(10),HX(10),XNU(10),CC(10),
     +RT(10),RNU(10),XDRPH(10),XDR(8),XDRH(8),DP(7),DX(7),DPDX(7),DFI(
     +7),DFH(7),XD1(8),DELP1(8),DPD(8)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA EXNO,AMP,DHM/6.0,14.3,8.75/
      DATA ROW,ROWF/1.1666,1.09570/
      DATA FNUE/17.9990/
      DATA DI,WF,HF,NF/0.07,0.003,0.015,6.0/
      DATA PAI/3.141592654/
      DATA TR,BR/30.0,760.0/
      DATA H/0.203,0.279,0.305,0.305,0.305,0.305,0.305,0.305,0.279,
     "0.203/
      DATA X1/0.0,0.26,0.46,0.66,0.86,1.06,1.26,1.46/
      DATA P/04.0,08.0,08.5,10.0,11.5,12.5,13.5,14.5/
      DATA CP,FK/1.00720,0.027970/
      DATA PR/0.703/
      DATA TE,TO/31.190,58.370/
      DATA X/0.0,0.06,0.26,0.46,0.66,0.86,1.06,1.26,1.46,1.52/
      DATA TW/65.386,69.70,87.770,102.149,108.723,111.771,114.812,
     +113.875,105.106,102.830/
      DATA FOT31/60.0/
      DATA FFI1/0.028/
      AX=PAI*(DI**2.0)/4.0
```

EXPERIMENT NO = 6.0 CURRENT= 14.3 A HEATER RESISTANCE = 8.75 OHM

CROSS-SECTIONAL AREA OF THE TUBE,AX	=	0.00385	SQ.M
CROSS-SECTIONAL AREA OF FIN-TUBE,AXF	=	0.00358	SQ.M
HYDRAULIC DIAMETER,DH	=	0.03579	M
MEAN VELOCITY,V	=	6.84291	M/S
C	=	12.98188	
MASS FLOW RATE,M	=	0.03072	KG/S
MEAN VELOCITY IN FIN-TUBE,VF	=	7.83542	M/S
REYNOLDS NO.BASED ON INSIDE DIA.,REI	=	26612.79	
REYNOLDS NO.BASED ON HYDRAULIC DIA.,REH	=	15581.34	

FRICTION FACTOR CALCULATION:

AXIAL DIST,X	0.00	0.26	0.46	0.65	0.85	1.06	1.25	1.46
P	4.00	8.00	8.50	10.00	11.50	12.50	13.50	14.50
DELH(MM WATER)		0.00	0.50	2.00	3.50	4.50	5.50	6.50
DELP IN KG/SQ.M		0.00	0.50	1.99	3.48	4.48	5.48	6.47
DELPX			2.49	4.98	5.81	5.60	5.48	5.39
FI BASED ON DI			0.013	0.025	0.030	0.029	0.028	0.028
FH BASED ON DH			0.006	0.013	0.015	0.015	0.014	0.014

X	X/DH REH	X/DI REI	FH	FI
0.46	0.325E-03	0.247E-03	0.0065	0.0127
0.66	0.116E-02	0.354E-03	0.0130	0.0254
0.86	0.154E-02	0.462E-03	0.0152	0.0296
1.06	0.190E-02	0.569E-03	0.0146	0.0286
1.25	0.226E-02	0.676E-03	0.0143	0.0279
1.46	0.262E-02	0.784E-03	0.0141	0.0275

AXIAL DIST,X	0.00	0.26	0.46	0.65	0.86	1.06	1.25	1.46
P	4.00	8.00	8.50	10.00	11.50	12.50	13.50	14.50
INCREMENTAL PRESSURE								
DROP, DP(X)		4.00	0.50	1.50	1.50	1.00	1.00	1.00
DX		0.26	0.20	0.20	0.20	0.20	0.20	0.20
DP/DX		15.38	2.50	7.50	7.50	5.00	5.00	5.00
DFI(X)		0.079	0.013	0.038	0.038	0.025	0.026	0.026
DFH(X)		0.040	0.007	0.020	0.020	0.013	0.013	0.013

X/D	0.00	3.71	6.57	9.43	12.29	15.14	18.00	20.86
DELP1(X)	0.00	4.00	4.50	6.00	7.50	8.50	9.50	10.50
DIMENTION LESS PRESSURE DROP								
DP/.5*RHO*VF**2	0.00	1.17	1.31	1.75	2.19	2.48	2.77	3.06

THERMAL RESULT:

AIR INLET TEMP = 31.190 C AIR OUTLET TEMP. = 68.370 C
PROPERTIES OF AIR EVALUATED AT T= = 49.780 C
HEAT INPUT TO AIR 1150.466 J/S OR WATT
HEAT INPUT PER UNIT LENGTH QL = 756.885W/M

DIST X	0.000	0.060	0.250	0.460	0.660	0.860	1.050	1.250	1.460	1.520
TW	65.39	69.70	87.77	102.15	108.72	111.77	114.81	113.87	105.11	102.83
TBX	31.19	32.66	37.55	42.44	47.33	52.23	57.12	62.01	66.90	68.37
(TW-TB)X	34.20	37.04	50.22	59.71	61.39	59.54	57.69	51.86	38.20	34.46

EFFECTIVE HEAT TRANSFER AREA AH = 0.3999 SQ.M

X/L	0.00	0.04	0.17	0.30	0.43	0.57	0.70	0.83	0.96	1.00
X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
HX	55.3	51.1	37.7	31.7	30.8	31.8	32.8	36.5	49.5	54.9
NUX	70.8	65.4	48.2	40.6	39.5	40.7	42.0	46.7	63.4	70.3

NOW WE PLOT (TW-TB)X VS X/L CURVE & FROM THE CURVE WE FIND AXIALLY UNCHANGING WALL-TO-BULK TEMP. DIFFERENCE IN THERMAL FULLY DEVELOPED REGIME

(TW-TB)F/ (TW-TBX)	1.75	1.62	1.19	1.00	0.98	1.01	1.04	1.16	1.57	1.74

HEAT TRANSFER COEFF.IN THERMAL FULLY DEVELOPED REGION, FHE = 31.544 W/SQ.M C

NUSSELT NO. IN THERMAL FULLY DEV. REGION,NUF = 40.355

HEAT TRANSFER COEFFICIENT BASED ON INSIDE DIA.& NOMIAL AREA,FHI = 57.363 W/SQ. M C

NUSSELT NO. BASED ON INSIDE DIA AND NOMINAL AREA = 143.561

X/D	0.00	0.86	3.71	6.57	9.43	12.29	15.14	18.00	20.86	21.71
NUX/FNU	1.75	1.62	1.19	1.00	0.98	1.01	1.04	1.16	1.57	1.74

X	X/DH REH PR	NUX
0.00	0.0000E+00	70.825
0.06	0.1530E-03	65.383
0.26	0.6632E-03	48.225
0.46	0.1173E-02	40.564
0.66	0.1683E-02	39.452
0.86	0.2194E-02	40.674
1.06	0.2704E-02	41.979
1.26	0.3214E-02	46.697
1.46	0.3724E-02	63.396
1.52	0.3877E-02	70.283

CONSTANT PUMPING POWER SMOOTH TUBE REYNOLDS,RED= 0.449E+05
HEAT TRANSFER COEFF.FOR SMOOTH TUBE AT CONSTANT PUMPING POWER,HO = 42.052 W/SQ. M C

CRITERIA R3 = FHI/HO = 1.364

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C     PROGRAM 1
C     PROGRAM TO CONVERT EACH THERMOCOUPLE READING IN MV TO TEMP. IN
C     DEGREE CENTIGRADE.
C     PROGRAM FOR M.SC.ENGG.THESIS OF M.HUQ.
C     ET  MEANS EXPERIMENTAL WALL TEMP. IN DEGREE CENTIGRADE
C     ETMV MEANS EXPERIMENTAL THERMOCOUPLE READING IN MV
C
C
      REAL ETMV,ET
      DIMENSION ETMV(39),T(118),TMV(118),ET(39)
      OPEN(UNIT=1,FILE='INPUT',STATUS='OLD')
      OPEN(UNIT=3,FILE='OUTPUT',STATUS='NEW')
      DATA T/0.,1.,2.,3.,4.,5.,6.,7.,8.,9.,10.,11.,12.,13.,14.,
     115.,16.,17.,18.,19.,20.,21.,22.,23.,24.,25.,26.,27.,28.,
     229.,30.,31.,32.,33.,34.,35.,36.,37.,38.,39.,40.,41.,42.,
     343.,44.,45.,46.,47.,48.,49.,50.,51.,52.,53.,54.,55.,56.,
     457.,58.,59.,60.,61.,62.,63.,64.,65.,66.,67.,68.,69.,70.,
     571.,72.,73.,74.,75.,76.,77.,78.,79.,80.,81.,82.,83.,84.,
     685.,86.,87.,88.,89.,90.,91.,92.,93.,94.,95.,96.,97.,98.,
     799.,100.,101.,102.,103.,104.,105.,106.,107.,108.,109.,110.,
     8111.,112.,113.,114.,115.,116.,117./
      DATA TMV/0.,0.039,0.078,0.117,0.156,0.195,0.234,0.273,
     +0.312,0.351,0.391,0.430,0.470,0.510,0.549,0.589,0.629,0.669,
     +0.709,0.749,0.789,0.830,0.870,0.911,0.951,0.992,1.032,1.073,
     +1.114,1.155,1.196,1.237,1.279,1.320,1.361,1.403,1.444,1.486,
     +1.528,1.569,1.611,1.653,1.695,1.738,1.780,1.822,1.865,1.907,
     +1.950,1.992,2.035,2.078,2.121,2.164,2.207,2.250,2.294,2.337,
     +2.380,2.424,2.467,2.511,2.555,2.599,2.643,2.687,2.731,2.775,
     +2.819,2.864,2.908,2.953,2.997,3.042,3.087,3.131,3.176,3.221,
     +3.266,3.312,3.357,3.402,3.447,3.493,3.538,3.584,3.630,3.376,
     +3.721,3.767,3.813,3.859,3.906,3.952,3.998,4.044,4.091,4.137,
     +4.184,4.231,4.277,4.324,4.371,4.418,4.465,4.512,4.559,4.607,
     +4.654,4.701,4.748,4.796,4.844,4.891,4.939,4.987,5.035,5.083/
      WRITE(3,7)
 7    FORMAT(9X,'MV-TEMPERATURE'/9X,' CONVERTION TABLE'///)
      WRITE(3,10)
 10   FORMAT(9X,'READING NOS',6X,'MV',6X,'TEMP.C'/9X,11('_'),4X,2('_'),
     +6X,6('_'))
      DO 300 I=1,39
      READ(1,15)ETMV(I)
 15   FORMAT(F6.3)
      DO 50 K=2,118
      JJ=K-1
      IF(ETMV(I)-TMV(K)) 33,44,50
C33   WRITE(3,34)ETMV(I),TMV(JJ),TMV(K),T(JJ),T(K)
C34   FORMAT(5X,5F15.5)
 33   ET(I)=((ETMV(I)-TMV(JJ))*(T(K)-T(JJ))/(TMV(K)-TMV(JJ)))+T(JJ)
      GO TO 55
 44   ET(I)=T(K)
      GO TO 55
 50   CONTINUE
 55   WRITE(3,20)I,ETMV(I),ET(I)
 20   FORMAT(13X,I2,7X,F8.3,3X,F8.3)
 300  CONTINUE
      STOP
```

MV-TEMPERATURE
CONVERTION TABLE

READING NOS	MV	TEMP.C
1	1.245	31.190
2	1.674	41.500
3	2.550	61.886
4	2.704	65.386
5	2.644	64.023
6	2.895	69.705
7	2.811	67.818
8	3.641	87.768
9	3.770	89.065
10	4.378	102.149
11	4.322	100.957
12	4.688	108.723
13	4.612	107.105
14	4.833	111.771
15	4.721	109.417
16	4.978	114.812
17	4.953	114.292
18	4.933	113.875
19	4.901	113.209
20	4.517	105.106
21	4.508	104.915
22	4.410	102.830
23	4.317	100.851
24	4.174	97.787
25	3.744	88.500
26	2.830	68.244
27	2.837	68.400
28	2.840	68.467
29	4.035	95.872
30	2.650	64.159
31	3.506	83.299
32	3.257	77.800
33	3.391	81.681
34	3.293	78.597
35	3.661	87.826
36	3.906	92.000
37	3.255	77.756
38	3.349	79.822
39	1.206	30.244

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