SC

# HEAT EXCHANGERS

# WITH

# FINNED CLOSED TWO-PHASE THERMOSYPHONS

by

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## SUMMARY

The performance characteristics of a heat exchanger using finned two-phase thermosyphon tubes were studied both analytically and experimentally.

An analytical model was used to determine the effects of parameters such as number of fins, Reynolds number of the flows, the ratio of hot to cold lengths, the ratio of hot to cold Reynolds numbers, change in Prandtl number of stream fluids and number of transfer units of the exchanger. The values obtained from the analytical studies were found to be comparable with similar studies performed previously. The analytical study introduced a model to approximate heat transfer from a finned surface and it was found to be comparable with the correlation cited in literature.

Experiments were performed to verify the analytical findings. The parameters varied were the ratio of hot to cold Reynolds numbers, and the number of fins. Although the basic trends exhibited were quite similar, the quantitative correlation between experiment and analysis was poor. The experimental results compared poorly with those cited in literature.

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# NOMENCLATURE

| | |
|---|---|
| $A$ | Area;Constant |
| $B$ | Constant |
| $B_1$ | Constant depending on arrangement |
| $C$ | Constant |
| $C_p$ | Specific Heat |
| $C_{sf}$ | Constant depending on working fluid and tube material |
| $C_z$ | Constant to account for row variation |
| $C_1$ , $C_2$ | Constants |
| $d$ | Diameter |
| $g$ | Gravitational acceleration |
| $h$ | Heat transfer film coefficient , height |
| $h_{fg}$ | Latent heat of vapourization |
| $J$ | Colburn J-Factor , |
| $k$ | Thermal conductivity |
| $K$ | Ratio of specific heat at constant pressure to that at constant volume |
| $l$ | Length |
| $l'$ | Constant |
| $L^+$ | Ratio of hot to cold length |
| $m$ | Constant depending on arrangement |
| $\dot{m}$ | Mass flow rate |
| $N$ | Constant |
| NTU | Number of transfer units |

| | |
|---|---|
| Nu | Nusselt number , hd/k |
| P | Pressure, Perimeter |
| Pr | Prandtl number, $\mu$Cp/k |
| q | Heat flux |
| Q | Rate of heat transfer |
| r | Radius |
| R | Universal gas constant, Thermal resistance |
| Re | Reynolds number, Vd/$\nu$ |
| Re* | Ratio of hot to cold stream fluids |
| s | Spacing |
| St | Stanton number, h/$\rho$CpV |
| T | Temperature |
| u | Constant |
| U | Overall coefficient of heat transfer |
| V | Velocity |
| w | Fin width |
| y | A variable |

Greek Letter Symbols

| | |
|---|---|
| $\alpha$ | Spacing of fins |
| $\gamma$ | Specific weight |
| $\delta$ | Film thickness |
| $\delta_o$ | Half the fin height |
| $\Delta$ | Difference |
| $\eta$ | Efficiency of the finned tube |
| $\theta$ | Difference in temperature |

| | |
|---|---|
| $\mu$ | Dynamic viscosity |
| $\nu$ | Kinematic viscosity |
| $\xi$ | Fraction of area |
| $\rho$ | Density |
| $\sigma$ | Surface tension |
| $\Omega$ | Constant |

## Subscripts

| | |
|---|---|
| air | Air |
| avg | Average |
| b | Base |
| c | Convective, cold |
| calc | Calculated |
| e | Extreme |
| f | Fin |
| fp | Flat plate |
| h | Hot |
| hyd | Hydraulic |
| i | Inside |
| j | j element |
| l | Lateral |
| m | Log mean |
| n | n element |
| o | Outside |
| out | Outlet |

| | |
|---|---|
| p | Element |
| r | Radially |
| s | Saturation |
| son | Sonic |
| st | Static pressure manometer fluid |
| t | Total, transverse |
| tp | Total pressure manometer fluid |
| v | Vapour |
| x | At any point |

# CHAPTER 1

## Introduction

Heat exchangers are devices which transfer heat energy from one medium to another and can be an integral part of most conventional energy conversion systems. Improvements in heat exchanger design therefore mean an increase in the effectiveness of the whole energy conversion process.

Energy in the form of heat is wasted when the source heat energy is deemed as low grade and therefore inefficient to transfer, or when the source heat energy is inaccessible for physical reasons using conventional heat exchange approaches. A heat exchanger using two phase closed thermosyphons or heat pipes can efficiently be used to transfer heat between fluid streams having a low difference in temperature, such as with low grade heat. Also due to their particular performance characteristics, the thermosyphon heat exchanger can be used where other more conventional heat exchangers become inappropriate.

They can be applied in situations where the hot and cold flows cannot risk cross contamination(i.e.

heat transfer between sodium and water), or in situations where the self pumping mechanism becomes vitally important ( i.e. applications in satellites). They are also useful in situations where hot side wall temperature is important for condensation reasons ( i.e. condensation of acids in steam generators). For the same temperature difference between the flows, the outside metal temperature is higher than in conventional heat exchangers and therefore hot side gases could be exhausted at somewhat lower temperatures without condensation. Thermosyphon heat exchangers are very reliable because they have no moving parts and since every element is separate the exchanger acts like many conventional ones in parallel. The disadvantages of this type of a system are that they would be somewhat more expensive than a rotating regenerative or tubular air preheater, each element must be treated as a pressure part, and since the cold side outer metal temperature is somewhat cooler than conventional exchangers icing could become a problem more readily (i.e. air heater application in steam generators.)

There is quite a need to optimize their performance and study their characteristics closely so as to use thermosyphon heat exchangers beneficially where they are applicable.

The two phase closed thermosyphon or heat pipe is a sealed enclosure containing a working fluid. Its functioning is actuated by a temperature difference across the device. On the hot side or evaporator the working fluid will absorb heat from the hot stream and vapourize causing it to travel to the cold side or condenser section. In the condenser section the working fluid condenses, thus releasing its latent heat to the cold stream. The working fluid will then return to the evaporator section. The mechanism by which it does so determines whether it is to be classified as a heat pipe or a thermosyphon. In the two phase closed thermosyphon the action of an external force (i.e. gravity or centrifugal force) will return the condensate to the evaporator. The heat pipe returns its working fluid by capillary action and therefore contains an internal wick.

There are three general topics pertaining to this subject under which published works may be categorized.

A) Review articles

B) Performance studies of heat pipe or thermosyphon elements.

C) Performance studies of heat exchangers using thermosyphons or heat pipes.

A) Review articles.

A study of the review articles will be made first and then several important, omitted, or recent articles will be examined more closely.

The first extensive review of thermosyphon technology available prior to 1973 was done by D.Japiske (1). In this review all types of thermosyphons are discussed, typically the open thermosyphon, the closed single phase thermosyphon, the closed loop thermosyphon and the two phase closed thermosyphon. He delves extensively into hydrodynamic aspects of the thermosyphon and then gives a review of performance aspects of such a system. Special attention is given to filling volume studies. There is then a discussion of applications such as the cooling of cryogenic equipment and thermopiles.

A more general review was done in 1975 by Finlay (2). He gives selection criteria for a working fluid and states that its operating vapour pressure should be between 0.1 and 10 bar. He discusses regulation schemes by the introduction of a non-condensible gas into the condenser from a reservoir on the end. This gas will interfere with the condensing of the working fluid and thus enable control of the heat transfer process. There is then a discussion of applications to the fields of electronics, engines, production processes, and rotating systems (i.e. cooling of turbine blades).

Similar reviews were done in 1978 by Pragar and Sun (3) and Joy (4). Reay (5) included cost effectiveness studies in his review in 1979.

B. Performance studies of heat pipe or thermosyphon elements.

Performance studies of bare two phase closed thermosyphons were presented in 1972 by Lee and Mital (6). This experimental study was carried out to determine the effect of the amount of working fluid, the operating pressure, the heat flux, $L^{+}$, $Re^{*}$ and the type of working fluid. They concluded that there is a maximum heat flow rate through the thermosyphon, which is imposed by limited flow rates through the system.

In 1976 Bezrodnyi and Beloivan (7) studied maximum heat fluxes experimentally by varying their geometric, physical and regime parameters. These limitations are the dry out limit which occurs when there is only a small amount of working fluid in the pipe, the burn out limit where high liquid fill and high radial heat flux into the evaporator causes the onset of film boiling, and the liquid-vapour interaction, entrainment, or flooding limit which is caused by interference between the rising vapour and the returning condensate.

Groll et al. (8) in 1979 also studied performance limitations of both thermosyphons and heat pipes. Working fluids and structural materials were tested for compatability and load cycling tests on several heat pipes and thermosyphons were performed.

Also in 1979 a similar study was conducted by Imura et al. (9) on the effects of working fluid, length, heat flux, operating pressure, ratio of heated to cooled length and fill quantity which is essentially the same as Lee and Mital's work (6). They found an optimum quantity to be between 10 and 20% of entire volume. A correlating formula for heat transfer on both the evaporator and condenser sides was found. They noticed that the film coefficient on the evaporator side has a tendency to become smaller for a large fill volume and large $L^+$ ratio and if the fill volume is over 25% there is a high degree of liquid-vapour interaction.

In 1981 Clements and Lee (10) examined the effects of a diameter change, while keeping the ratio of height to diameter constant. It was noted that the overall coefficient of heat transfer will decrease with increasing diameter.

Also, in 1981 Shiraishi et al. (11) studied both experimentally and analytically the effect of the amount of working fluid, the operating temperature, the heat flux and the type of working fluid. It is interesting to note that for the purpose of transmission through the system, the evaporator side was divided into two sections, one with the film and the other with the pool.

C. Performance studies of heat exchangers using thermosyphons or heat pipes.

A two-phase thermosyphon or heat pipe exchanger study was published by Feldman and Lu (12) in 1977. As with most studies of this nature, the condenser and evaporator resistances were taken as constant instead of functions of temperature. In their work they used a grooved condenser which has the advantage of entraining the condensate in the grooves and thus reducing the resistance across the film. They conclude that heat exchanger performance can be optimized by using a maximum heat exchanger length, staggered finned tubes, tubes of largest possible diameter, reasonably small fin height, maximum fins per inch and fin thickness that pressure drop will tolerate.

In 1978, Lee and Bedrossian (13) studied experimentally and analytically heat exchangers using bare thermosyphons or heat pipes. They studied the effects of varying the $L^+$ ratio, the $Re^*$ ratio and the effects of changing the Prandtl number of the streams.

Prager and Sun (3) in 1978 studied the use of a finned tube heat exchanger where the internal resistances were again taken as constant. This study uses the NTU method and gives particular attention to hot and cold side fouling.

Sun and Prager (14) also in 1978 published a paper geared towards manufacture, and marketing of heat pipe exchangers. It discusses applications to commercial laundry, food processing, plasticizer curing oven, comfort heating, power generation, refrigeration, energy storage and coal gasification.

The objectives of the present research were as follows:

1) To create a model and a computer program which could simulate and predict the performance of finned two-phase thermosyphon heat exchangers with maximum flexibility, without assuming that the internal resistances were constant.

2) To analytically examine the effects of such parameters as the number of fins, the Reynolds numbers of the two flows, the ratio $L^{+}$, the ratio $Re^{*}$, the change in Prandtl number of the stream fluids and the NTU of the heat exchanger.
3) To set up an experimental test section which would show the effects of the ratio $Re^{*}$ and the number of fins on the performance of the heat exchanger.
4) To compare analytical and experimental results.

# CHAPTER 2

# ANALYSIS

In this chapter the method for the solution of this problem is given both analytically and numerically.

## 2.1 Analytical Model

In the analysis of the system a conductance model was used to estimate the heat transfer in two-phase closed thermosyphons. In this approach for each process involved in the transmission of heat there will be a coefficient of heat transfer calculated. From these individual coefficients an overall coefficient of heat transfer can be predicted.

For the finned tubes being studied the total thermal resistance between the temperature of the hot flow and the temperature of the cold flow can be given by

$$R_t = R_{12} + R_3 + R_5 + R_{67} \tag{2.1}$$

In our model it is assumed that no heat is axially conducted. This is a reasonable assumption because the cross section of the tube is very small in relation to the surface area of the tube.

The heat transferred across one tube is defined as

$$q_p = \left[\frac{1}{R_t}\left(T_h - T_c\right)\right]_p \tag{2.2}$$

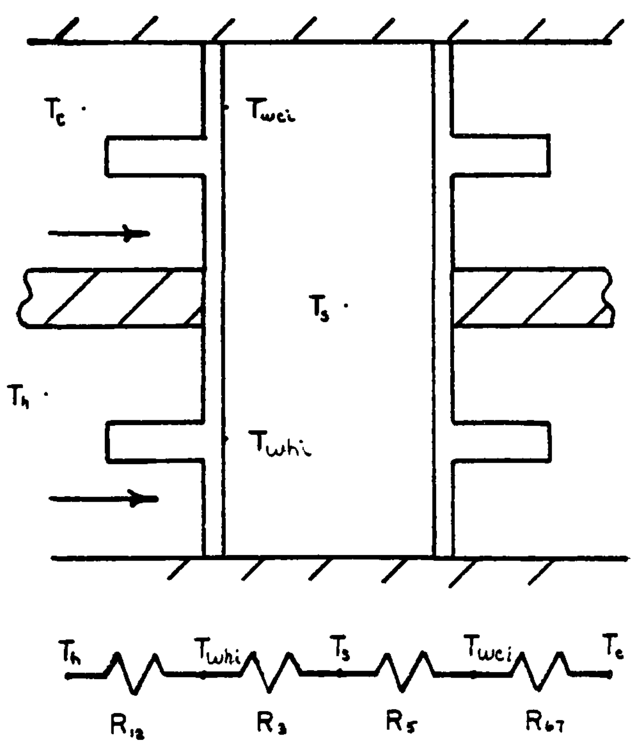
This $q_p$ changes from row to row due to changes in the properties of the hot and cold streams. Therefore the heat transfer for the entire system can be defined as

$$Q_t = \sum_{p=1}^{n} Q_p = A_{to} U_{to} (\Delta T)_m \tag{2.3}$$

or

$$Q_t = \sum_{p=1}^{n} Q_p = A_{ti} U_{ti} (\Delta T)_m \tag{2.4}$$

Fig·2·1




The resistances $R_{12}$ and $R_{67}$ represent the external resistance on hot and cold sides respectively.

$R_3$ is the resistance from evaporation and $R_5$ is the resistance due to the condensation. The Resistance $R_4$ is the resistance which occurs when the flow inside the thermosyphon reaches its choking condition at a Mach value close to one. The sonic velocity is

$$V_{son} = \sqrt{K g R T} \tag{2.5}$$

which at a temperature of 150°F and a K value of 1.33, is around 1500 ft/sec.

The actual velocity attained for a particular case is obtained from

$$Q = \dot{m} h_{fg} \tag{2.6}$$

and

$$V = \dot{m} / \rho A \tag{2.7}$$

It is usually in the range of 10 ft/sec., therefore, since the actual velocity is less than 1% of the sonic velocity, $R_4$ need not be considered.

The thermal resistances $R_{12}$ and $R_{67}$ are the resistances for the outside of a finned tube including conduction through the walls,

$$R_{12} = 1/h_{12} A_{hi} \tag{2.8}$$

and

$$R_{67} = 1/h_{67} A_{ci} \tag{2.9}$$

where $h_{12}$ and $h_{67}$ are the film coefficients of the outside based on the inside.

The film coefficients $h_{12}$ and $h_{67}$ are based on a weighted average between the fin and the base portions which are derived as follows. The efficiency of the finned tube is defined as: (Appendix A)

$$\eta = \frac{2}{u_b\{1-(u_e/u_b)^2\}}\left\{\frac{I_1(u_b)-\beta_1 K_1(u_b)}{I_0(u_b)-\beta_1 K_0(u_b)}\right\} \qquad (2.10)$$

where

$$\beta_1 = I_1(u_e)/K_1(u_e) \qquad (2.11)$$

$$u_b = \frac{(r_e - r_b)\sqrt{2h/kw}}{(r_e/r_b - 1)} \qquad (2.12)$$

and

$$u_e = u_b(r_e/r_b) \qquad (2.13)$$

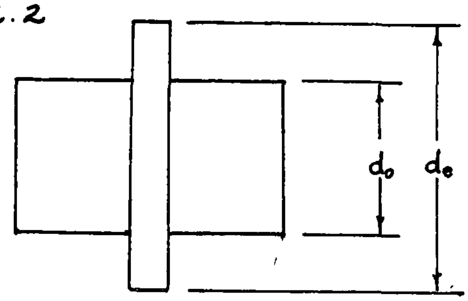
An approximate value of $h_f$ was found by considering the annular fin to be a flat plate. The equivalent length of the fin was found by letting the area of half the fin be equal to the area of the flat plate. The widths of the fin and the flat plate were then considered to be equal

$$A_f = A_{fp} \qquad (2.14)$$

$$\frac{\pi/4\,(d_e^2 - d_o^2)}{2} = (d_e - d_o)\ell_e \qquad (2.15)$$

therefore $$l_e = \frac{\pi(d_e^2 - d_o^2)}{8(d_e - d_o)} \tag{2.16}$$

Fig. 2.2




Then to find the value of $h_f$ one can use the equation for turbulent boundary layer flow across a flat plate, because the fluid stream approaching the element is so disturbed that the turbulent boundary developed from the beginning of the section. The equation is defined in terms of the Stanton, Prandtl, and Reynolds numbers(15).

At any point the Stanton number is

$$St_x \, Pr^{2/3} = 0.0288 \, Re_x^{-0.2} \tag{2.17}$$

To find the average Stanton number, integrate over the length of the plate and then divide by the length of the plate

$$St = \frac{0.0288 \int_0^{l_e} Re_x^{-0.2} \, dx}{l_e \, Pr^{2/3}} \tag{2.18}$$

$$St = \frac{h_f}{\rho C_p V} = 0.036 \left(\frac{\rho V \ell_e}{\mu}\right)^{.02} Pr^{.2/3} \qquad (2.19)$$

$$h_f = 0.036 \rho C_p V \left(\frac{\rho V \ell_e}{\mu}\right)^{.0.2} Pr^{.2/3} \qquad (2.20)$$

To account for variation from row to row, one must multiply by an appropriate $C_z$(16) value.

The film coefficient for the base may be calculated by using the equation for forced convection from a bare tube which is defined in Hilperts equation(16)

$$Nu = h_b d_b / k = B \cdot C_z \cdot Re^m \cdot Pr^{1/3} \qquad (2.21)$$

where B and m are coefficients which depend upon the arrangement, be it inline or staggered. They also depend upon the longitudinal and transverse spacing within that particular arrangement. They are known as Grimisons values and are based on data gathered by Huge, and by Pierson (16) ( Appendix E )

$C_z$ is a coefficient used to correct the film coefficient for the row number. This film coefficient increases for the later rows because of the increased turbulence of the air flow ( Appendix E )

The outside film coefficient is a weighted average of the film coefficients of the fin and the base. It is defined as

$$h_o = \frac{\eta A_f h_f + A_b h_b}{A_f + A_b} \qquad (2.22)$$

Using different sizes of fins it is easier to use a film coefficient of the outside, based on the inside.

$$h_{oi} = \left(\frac{A_f + A_b}{A_i}\right) h_o \qquad (2.23)$$

The film coefficient obtained in this manner was compared with those of Legkiy, Pavlenko, Makorov and Zheludov(17), and Elmahdy and Biggs(18). (See Appendix C) The results showed that numbers obtained from the flat plate approximation were adequate, therefore the rest of the analysis was carried out using this approximation solely.

The resistance $R_3$ is found from the equation

$$R_3 = 1/h_3 A_{hi} \qquad (2.24)$$

where $h_3$ is determined from Rohsenows equation(19) for natural convection boiling.

The wall temperature in the bottom of the thermosyphon is at a temperature above the saturation temperature of the liquid at the pressure $P_s$ and therefore boiling occurs. The heat transferred across the surface of the water depends on the difference in temperature between the surface of the wall and the saturation temperature. One assumption made regarding the boiling heat transfer is that the optimum volume of working fluid is in the thermosyphon and therefore a pool of working fluid does not exist in the bottom of the evaporator. With the optimum filling the boiling will be of the nucleate form because the temperature difference across the film is relatively small.

Rohsenows relation for natural convection nucleate boiling which was developed from empirical data is

$$h_s = \frac{\mu h_{fg}}{\Delta T} \cdot \sqrt{\frac{\gamma - \gamma_v}{\sigma}} \left[ \frac{C_p \Delta T}{h_{fg}} \cdot \frac{1}{C_{sf} \cdot Pr^{1.7}} \right]^{1/0.33} \qquad (2.25)$$

The resistance $R_5$ is found from the equation

$$R_\delta = 1/h_\delta A_{ci} \qquad (2.26)$$

where $h_5$ is determined from Martinellis expression (20) for film condensation heat transfer.

$$Nu = \frac{h_\delta d_i}{k} = 1/\beta \left[ 1/4 - Y + 3/4\, Y^4 - Y^4 \ln Y \right] \qquad (2.27)$$

where (2.28)

$$Y = r_\delta / r_i$$

and (2.29)

$$\beta = Y^4 \ln Y \left( \frac{\ln Y}{2} - \frac{1}{2} \right) + \frac{Y^4}{8} + Y^2 \left( \frac{\ln Y}{2} - \frac{1}{4} \right) + \frac{1}{8}$$

$$\beta = \frac{2 k \nu (T_s - T_{wci})}{g (\gamma - \gamma_v) h_{fg} r_i^4} \qquad (2.30)$$

Martinellis expression is based on the assumptions of a uniform heat flux, the physical properties are independent of temperature, flow is incompressible and completely laminar.

Another resistance term is sometimes included in the resistance model caused by vapour pressure drop. This was excluded from the present analysis because the pressure drop is negligible.

## 2.2 Numerical Solution

All calculations were done using an IBM 360/365 and IBM 370 computers in conjunction with 'Fortran 1V level G' and 'Watfiv' compilers. Some work was also done using 'York APL'.

The conduction model for a finned thermosyphon tube is given in the subroutine called 'FONDUC'. The sequence will be described here in step form.

i) The value of $T_{whi}$ is estimated and since R is known, a value of Q can be found from the equation

$$Q = \frac{T_h - T_{whi}}{R_{12}} \tag{2.31}$$

The value of Q remains constant for a given thermosyphon.

ii) Evaluating $R_3$ poses a problem because $R_3$ is a function of $T_3$ (as well as other temperature dependent properties) not been quantified. Therefore $T_s$ must be estimated and then $R_3$ can be found. The heat flux is then found from the equation

$$Q_3 = \frac{T_{whi} - T_s}{R_3} \tag{2.32}$$

$T_s$ must be re-estmiated until $Q_3$ is equal to Q.

iii) Evaluating $R_5$ is also a problem because $R_5$ is a function of $T_{wci}$ (among other variables) which has not yet been defined. Therefore $T_{wci}$ must be estimated and then $R_5$ may be found. The heat flux is then found from the equation

$$Q_5 = \frac{T_5 - T_{wci}}{R_5} \tag{2.33}$$

$T_{wci}$ must be re-estimated until $Q_5$ is equal to Q.

iv) The value of $(T_c)_{calc}$ may then be calculated.

$$(T_c)_{calc} = T_{wci} - Q\,R_{67} \tag{2.34}$$

If the value of $(T_c)_{calc}$ is not equal to $T_c$ within $\pm 10^{-2}$ to $10^{-3}$ then the process is repeated, starting from re-estimating $T_{whi}$ .

The heat transfer in a parallel flow heat exchanger is simulated in the subroutine 'PARLEL'.

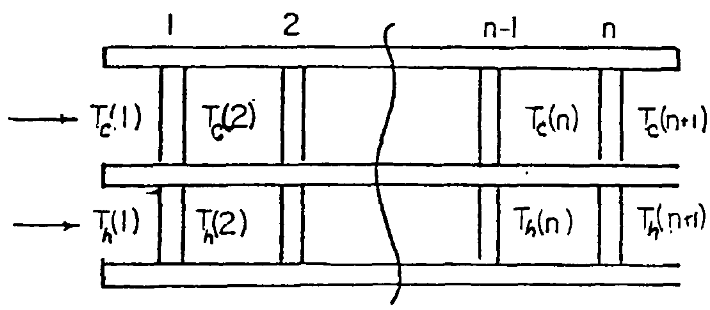



Fig.2.3

Consider the preceding diagram in which there are two air flows travelling through the system in the same direction. The rows are numbered so that a specific row will be referred to as the $j^{th}$ row. There are a total of $n$ rows.

The values are known at the inlet ($T_h(1)$, $T_c(1)$), so that the values leaving a particular row may be determined from the equation

$$Q = \dot{m}\, C_p\, \Delta T \qquad (2.35)$$

The heat transfer in a counterflow exchanger is modeled in subroutine 'CONFLO'.

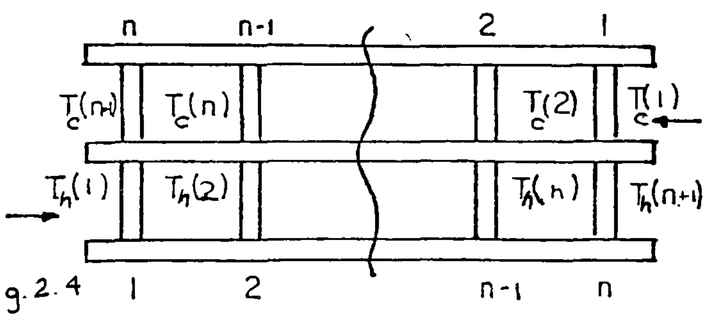



Fig. 2.4

The nomenclature used in this problem has been changed because the flows are now opposed. The rows on the cold side are referred to by 'KAY', whereas on the hot side 'J' is used.

The value of $T_c(n+1)$ is estimated, then the temperatures along the exchanger are calculated. If the value of $T_c(1)$ is not equal to the prescribed one, the value of $T_c(N+1)$ will have to be re-estimated, and then the process will repeat.

The flow charts for the program are given in Appendix D, the actual program used is given in Appendix E, the nomenclature for the program is given in Appendix F and input cards required for the program are given in Appendix G.

# CHAPTER 3

## EXPERIMENTAL FACILITIES & PROCEDURES

The test section and thermosyphons were designed and manufactured at the workshop of the department of Mechanical Engineering at the University of Ottawa. The rest of the test facility was the same as that of Bedrossian (21) except for the rotary switch, potentiometer and ice point which were replaced with a multi-channel digital thermocouple reader with an electronic ice point. The equipment was set up and calibrated, then debugged and run while processing the results simultaneously. The apparatus, setup, calibration and experimental procedure will be discussed in this section.

### 3.1 Apparatus

A schematic diagram of the experimental apparatus is given in Figure 3.1. Air is induced through both the hot and cold side by fans located at the outlets.

The air flow measurements are made using calibrated orifice plates located at the inlets of both the hot and the cold sides. The method of calibration is given in Appendix H. The air on the hot side was heated by a resistance type heater.

The bulk inlet and outlet temperatures on each side of the heat exchanger were taken using the average of five thermocouple readings. To ensure complete mixing, baffles were located upstream of each of the thermocouple locations. The thermocouples were of the copper-constantan type and were attached to a multi-channel digital thermocouple reader with a built-in ice point. To ensure validity of the readings the device was calibrated as described in Appendix H.

Air pressure differences were measured across both the hot and cold sides of the test section.

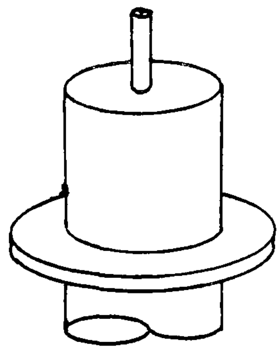
The test section shown in Figure 3.2 and Figure 3.3 contains a total of thirteen thermosyphons in a staggered arrangement (Three rows of three and two rows of two as shown in Figure 3.4).

The thermosyphons are shown in Figures 3.5 to 3.8. Although only thirteen were used, extras were made so that testing would not be delayed because of the defective thermosyphons.

The thermosyphons were machined from

SAE brass cylindrical cast rods (Figure 3.9), although commercial ones are available similar to those shown in Figures 3.10 and 3.11. Once the machining was completed one end was closed with a plug and a silver solder weld. The other end was closed except for a tube which extended from the plug as shown in Figure 3.12

Fig. 3.12



Through this side the thermosyphon was evacuated to about $1 \times 10^{-5}$ torr by means of a vacuum pump. Distilled water was then put into the tube and the tube was immediately sealed. They were sealed by heated tongs being applied at the extending tube, into which a silver solder strip had been previously inserted. The functioning of individual tubes was verified by testing the heat transfer characteristics of the thermosyphon. If they were not satisfactory the tube was dismantled and the process was repeated. The surfaces of the tubes were then sandblasted to remove oxides and other contaminants which were formed during the machining, welding and filling procedures. Sand-blasting would increase the film coefficient higher than that analytically calculated.

The test section was made from super asbestos sheets and is depicted in Figures 3.2, 3.3., and 3.4. To accomodate the finned tubes the divider was made of several pieces which fitted in around the the tubes. The joints were sealed with silicon sealer which had to be freshly applied every time the test section was dismantled.

### 3.2 Experimental Procedures

The purpose of obtaining experimental data was to help substantiate the analytical results. The experiments can be placed into two categories.

The first category of experiments consisted of varying the Reynolds number ratio ($Re^*$) of the two flows keeping the number of fins constant at sixteen per side. For each value of $Re^*$, five Reynolds numbers were used. The ratio $Re^*$ was then changed and the process was repeated. The $Re^*$ values evaluated in this manner were 0.5, 0.75, 1.0, 1.33 and 2.0.

In the second category of experiments the ratio $Re^*$ was kept constant at a nominal value of one and once again the five different Reynolds numbers were used. The chief variable in these experiments was the number of fins which was varied from 16, 12, 8, 4, and 2 fins per side.

The experimental procedure used was as follows:

1) The fans would be turned on first and then the heater would be started. The fans had to be operating prior to starting the heater to avoid heater burnout.
2) The doors to the laboratory would be opened so that the added circulation would avoid a change in the inlet temperature. This also avoided a transient condition from occuring during the course of the experiment.
3) The flows would be temporarily adjusted and the heater would be set to a specified point.
4) The system would be allowed to reach a steady state condition. This would take 30 minutes for the first reading and 20 minutes for each additional reading.
5) Final adjustments would be made to the flow rates by dampering the fans.
6) The pressure readings would then be taken from the multimanometer. A total of ten pressure readings were taken. Two pressures would be read at the inlet and the outlet of both the cold section and the hot section. The other two pressures were measured just downstream of the orifice plates and were used to calculate the

flow rates through the hot and cold sections.

7) A total of twenty thermocouples were read. Five were at the inlet and outlet of both the hot and cold ducts. They were arranged to give an entrance and exit temperature for both ducts.

8) The atmospheric pressure was read from a manometer located in laboratory B207 of Colonel By Hall.

The raw experimental data is given in Appendix I. A specification of the experimental equipment is presented in Appendix J. The experimental data was processed using a fortran program given in Appendix K.

# CHAPTER 4

## RESULTS AND DISCUSSION

### 4.1 Results from Analysis

In the analysis the type of heat exchanger considered was a ten row, air to air, parallel flow type with a staggered arrangement of finned tubes as defined in Appendix L.

There were three main studies done to investigate the operating characteristics of this finned two-phase thermosyphon heat exchanger.

The first is a study of the effect of changing the number of fins for various Reynolds numbers. The total heat transfer is given at various Reynolds numbers as a function of the number of fins (Figure 4.1). The overall coefficient of heat transfer based on the inside area is presented in a similar fashion in Figure 4.2. The overall coefficient of heat transfer and total heat transfer both increase with increasing numbers of fins and increased Reynolds numbers. The overall coefficient of heat transfer will increase because it is based on an inside area which is constant even though additional surface has been added to the outside.

The overall coefficient of heat transfer will increase with increasing Reynolds number because the greater velocity of the stream fluid will lower the outer resistance as seen from Hilperts equation. At lower Reynolds numbers the overall coefficient of heat transfer becomes almost constant for any number of fins. This infers that at lower Reynolds numbers the outside film coefficient is very insignificant to the overall heat transfer coefficient and thus the inside resistances have a stronger influence at lower Reynolds number. The magnitudes and trends are comparable to those of Lee and Bedrossian(13), remembering that in that particular study the overall coefficient of heat transfer was based on the area of either the hot or the cold side, and this study is based on the total area. No direct quantitative comparison will be given because the systems studied were not entirely the same.

In the second study the effects of changing the ratios $L^+$ and $Re^*$ with a constant number of 24 fins was examined. The total heat transfer and the overall coefficient of heat transfer based on the inside area are given in Figures 4.3 and 4.4, respectively. Consistent with Lee and Bedrossians(13) work the optimum $L^+$ is one which means that for optimum operation the condenser and the evaporator sections should be the same sizes. It is thought that by selectively finning one of the sides this value of optimum $L^+$ will change. These figures also show that increasing the flow through the hot side will increase the heat transfer rate while a simultaneous increase of $L^+$ and $Re^*$ will cause the heat transfer rate to drop. It is worthwhile to note that a simultaneous decrease of $Re^*$ and $L^+$ will cause the heat transfer rate to drop at a faster rate. It should be noted that in Lee and Bedrossians(13) study the Reynolds number of the hot flow rather than that of the cold flow was kept constant.

The third study was done to examine the effects of changes in the Prandtl number using water or oil as the stream fluids as compared with air (using a non-dimensional Prandtl number in the analysis directly was not possible). The overall coefficient of heat transfer based on the inside area was plotted versus the number of fins at various Reynolds numbers. This was done for both water and oil in Figures 4.5 and 4.6, respectively. The amount of total heat transfer was then compared using the oil , water and air as the stream fluids, in Figure 4.7. As expected the heat transfer increases due to the greater Prandtl number and while external resistances are the controlling factors for air, the internal resistances become more important at moderate Prandtl number (ie. water ).

The number of transfer units(NTU) were calculated for the results from the analysis and are shown in Figure 4.c. This shows that the NTU will increase for a greater number of fins on the tubes. This means that for the same flowrate past the tube there is a higher heat flux through the tube with more fins due to the increased surface area. The NTU is higher for a smaller Reynolds because the overall coefficient of heat transfer does not increase as quickly as th mass flow rate.

The major shortcomings of the theoretical analysis are that an optimum fill volume was assumed. The implcations of this are that there is no allowance made for burnout resulting from overfilling or dryout caused from underfilling. Martinellis condensation expression is based on assumptions of uniform heat flux and completely laminar flow. Both of these assumptions are not entirely true because there is a small temperature gradient along the walls of the thermosyphons caused by returning vapour thus creating an un-uniform heat flux. There is also an incredible percolating action within causing an extreme turbulence. In the evaporator section Rohsenows expression for natural convection boiling is based on an ideal situation where water is nucleating on the sides of a container and then boiling. In the actual case the falling film is receiving heat and then boiling in the pool at the bottom of the section. The analysis does not allow for the two separate sections in the evaporator, one containing the returning condensate and the other cotaining the boiling pool. There has been no allowance for axial conduction along the tube which in this particular study may be significant due to the thick walls of the tubes. There is also the problem of heat transfered through the other components in the system besides the tubes such as the walls and the divider which has been assumed to be negligible.

## 4.2 Results from Experiments

As previously stated there were two main main studies done experimentally. The conditions for these tests established from the experimental setup which is given in Chapter 3.

In the first study the Reynolds number ratio $Re^*$ was varied while keeping the number of fins constant at sixteen per side and readings were taken for various Reynolds numbers. As predicted by the present analysis the overall coefficient of heat transfer based on the outside area increases at any given Reynolds number of the cold flow as shown in Figure 4.9. This is because the increase of $Re^*$ will increase the Reynolds number on the hot side and therefore decrease the outer resistance, thus increasing the overall coefficient of heat transfer.

In the second study the number of fins per side was varied while keeping the Reynolds number ratio ($Re^*$) constant for various Reynolds numbers. The results are shown for 2, 4 and 16 fins per side in Figures 4.10, 4.11 and 4.12, respectively. The heat balance of the hot flow was consistently higher than that of the cold flow giving a higher coefficient of heat transfer. The reason for this was rationalized to be leakage because it was noted that the pressure drop across the cold side was exactly equal to the pressure drop across the hot side when it should have been less. It was also noted that improvements were made to these figures by attempts to seal the test section more adequately during the course of the experiment. The problem seemed to be in the design of the test section which in order to accomodate the finned tubes had to be made of many separate pieces fitting together around the tubes. It was necessary to dismantle the test section continuously during the course of the experiment in order to machine fins from the tubes. This aspect further increased the likeliness of leakage occuring. An analytical comparison was made later using input established from the experimental conditions and is given in Figures 4.10, 4.11 and 4.12. It is consistently lower than the experimental results but exhibits a similar upward trend best shown in Figure 4.13 where it is non-dimensionalized with respect to a similar result with no fins. Figure 4.14 shows just experimental values and indicates in much the same fashion as Figure 4.2 that the overall coefficient of heat transfer based on the internal area increases with the number of fins at a certain Reynolds number.

Figure 4.15 shows pressure drop across the test section as a function of Reynolds number for various numbers of fins. It can be seen that although marginally, pressure drop does increase as more fins are added. An average value of hot and cold side pressure drop was used because these values were similar in magnitude. This indicates that leakage was occuring between the hot and the cold sides because the pressure drop for the hot side should be greater than that of the cold side for the same Reynolds numbers.

# CHAPTER 5

## CONCLUSIONS AND RECOMMENDATIONS

The following conclusions can be drawn from the present study.

1. The overall heat transfer can be increased by using finned surfaces as opposed to bare surfaces. This effect is more pronounced at higher Reynolds numbers.

2. The overall coefficient of heat transfer increases as more fins are added for the range of fins investigated. This effect is more pronounced at higher Reynolds numbers.

3. The optimum ratio $L^+$ is 1.0 for the situations studied. This might be changed by selective finning of either the hot or the cold side.

4. Increases in the Prandtl number of the stream fluid will increase heat transfer to a certain point after which it will tend to level out with further increase.

5. The NTU can be increased with increasing number of fins and is higher for lower Reynolds numbers.

6. Although marginally for the range tested the pressure drop does increase with the number of fins added.

7. Heat transfer from a finned surface can be reasonably predicted by using the flat plate model.

8. Analytical findings correlate well with those cited in literature.

9. Experimental results show the correct trends however their magnitudes correlate poorly to either the analysis or results of this nature cited in previous literature. The cause of this discrepancy is partially due to leakage which could not be corrected during the course of the experiment. Other reasons for the discrepancy would be short-comings in the analysis which may be improved in the future.

Thermosyphon heat exchangers can be utilized most effectively when a specific situation dictates so. For this reason research could be oriented towards places where thermosyphons have a distinct advantage over conventional heat exchange approaches. There are topics such as selective finning which could be related to changes in the type of fluid on either the hot or the cold sides which could further increase the flexibility of the thermosyphon heat exchanger.

To improve the validity of this type of experiment the following recommendations could be followed. To locate leakage measurement of flow could be performed both entering and exiting the test section. Errors in temperature measurement could be made less significant by increasing the size of the test section. This would increase the temperature drops on both hot and cold sides thus facilitating reasonable temperature difference measurement.

FIGURES

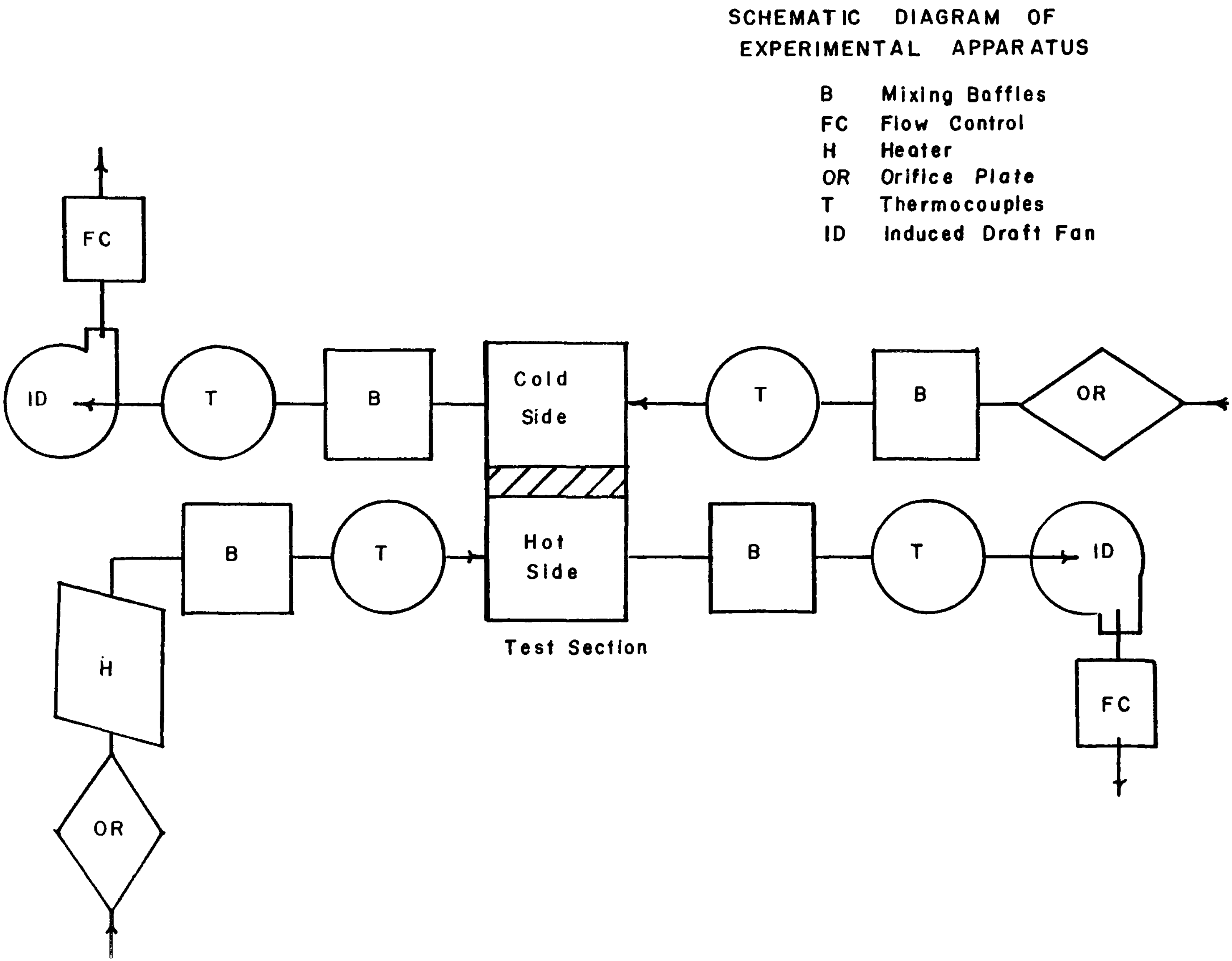
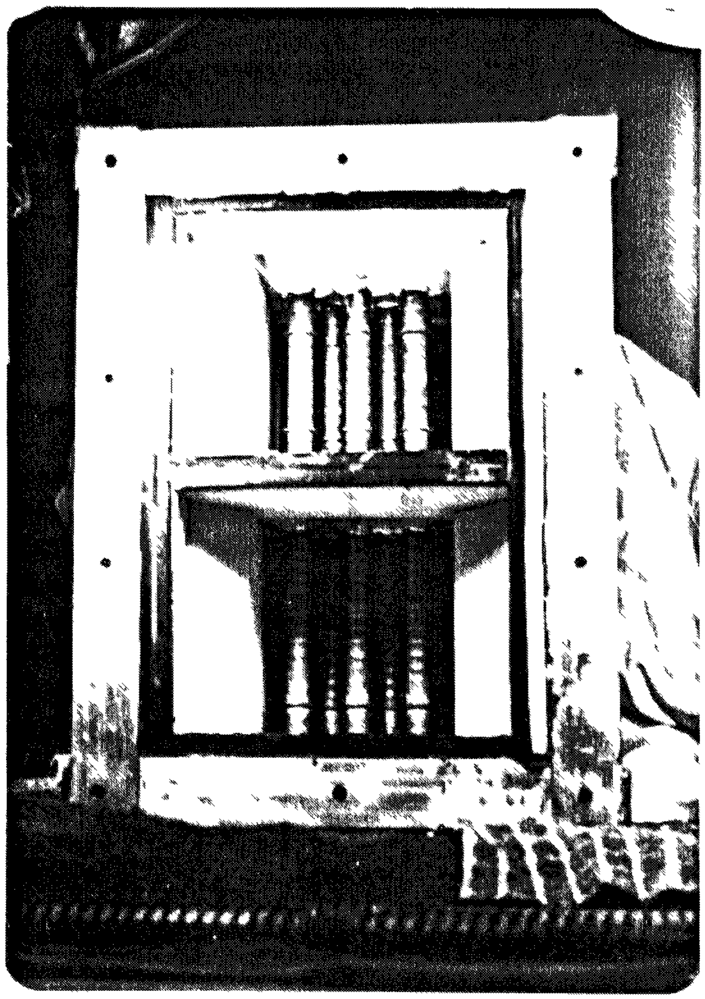

SCHEMATIC DIAGRAM OF
EXPERIMENTAL APPARATUS
B Mixing Baffles
FC Flow Control
H Heater
OR Orifice Plate
T Thermocouples
ID Induced Draft Fan
FC
ID
T
B
Cold
Side
T
B
OR
B
T
Hot
Side
B
T
ID
H
Test Section
FC
OR

Fig.3.1



Photograph of Test Section

Fig.3.2

TEST SECTION ARRANGEMENT

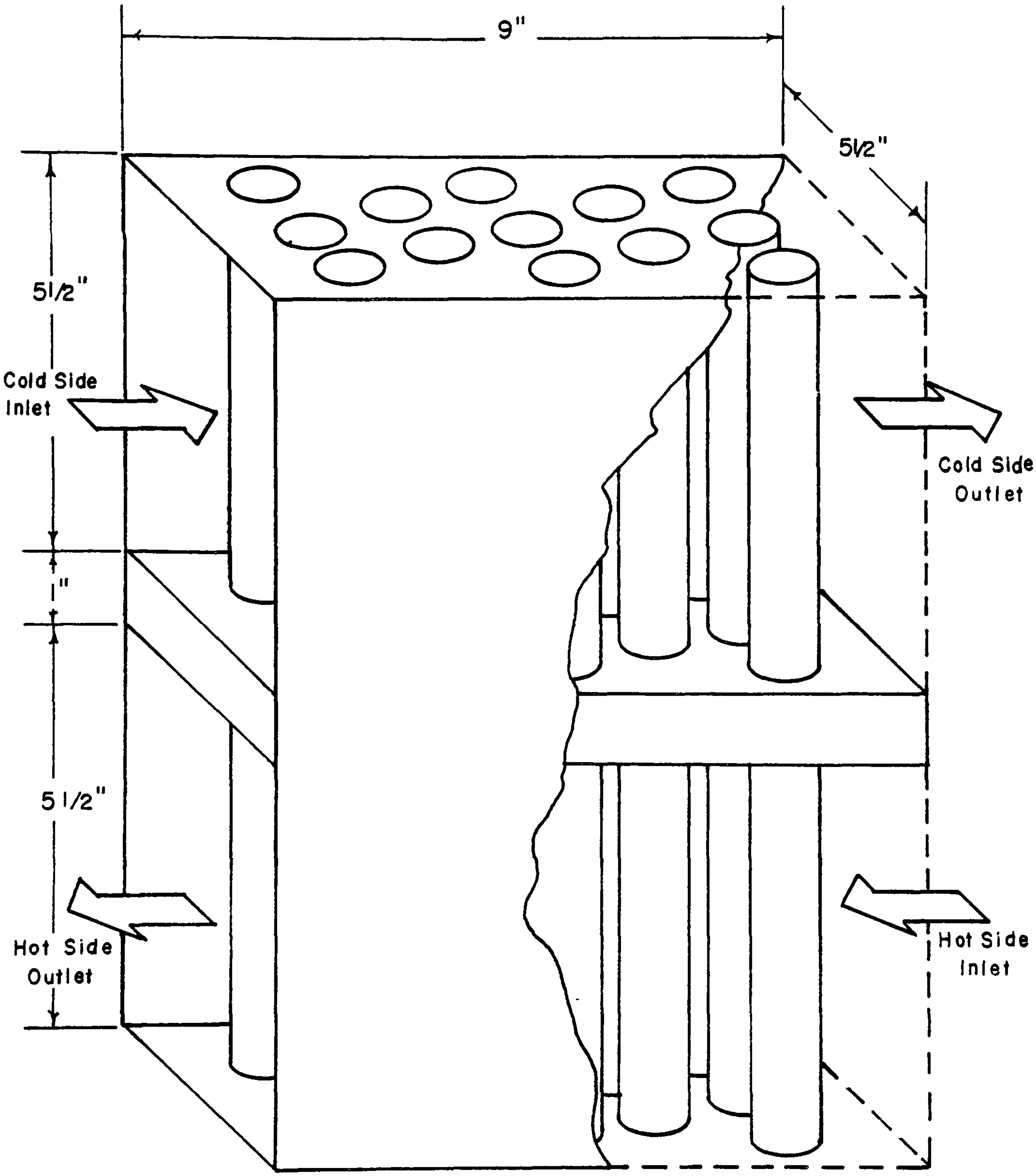
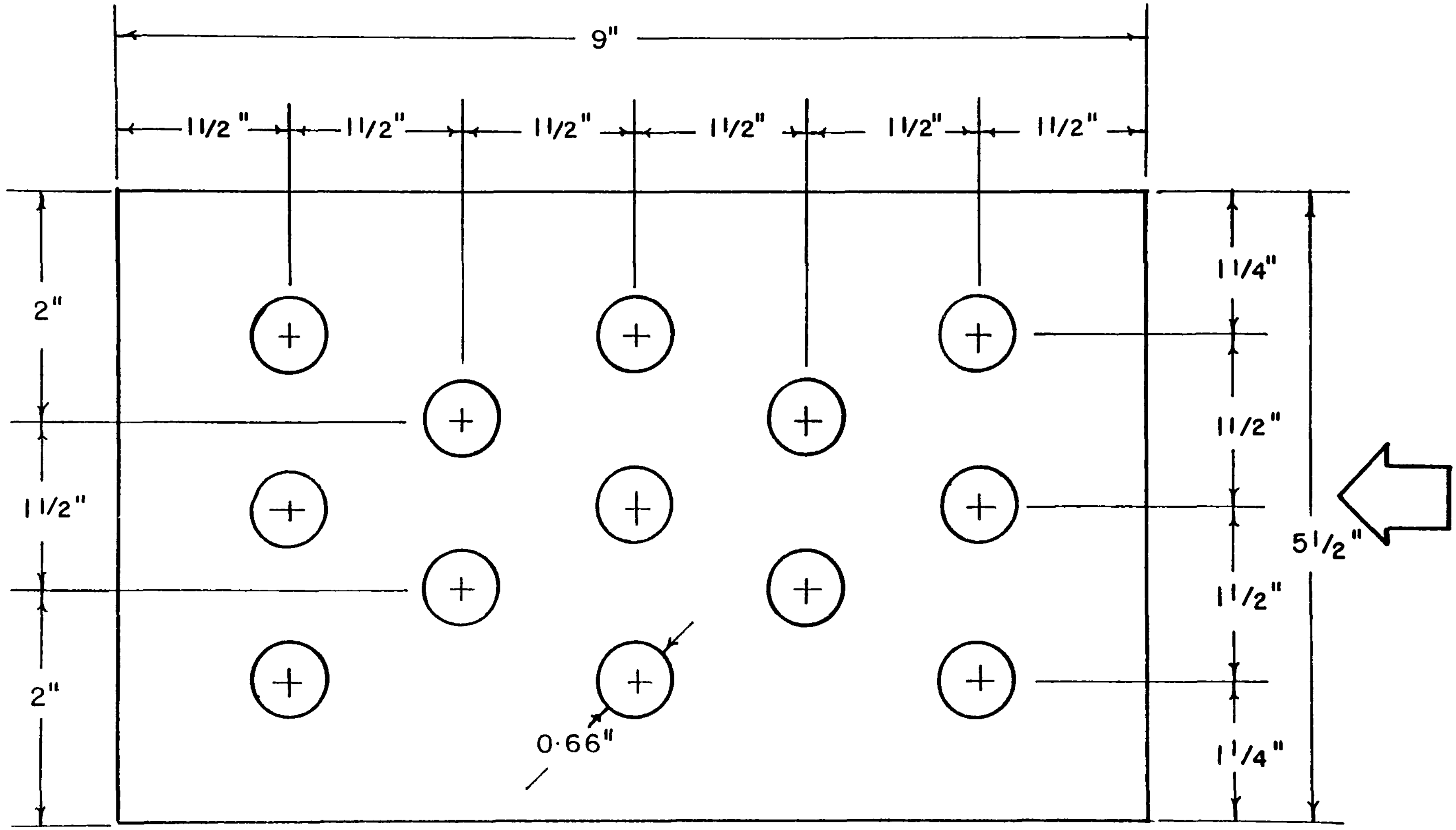



Fig.3.3




FINNED TUBE ARRANGEMENT

Fig.3.4

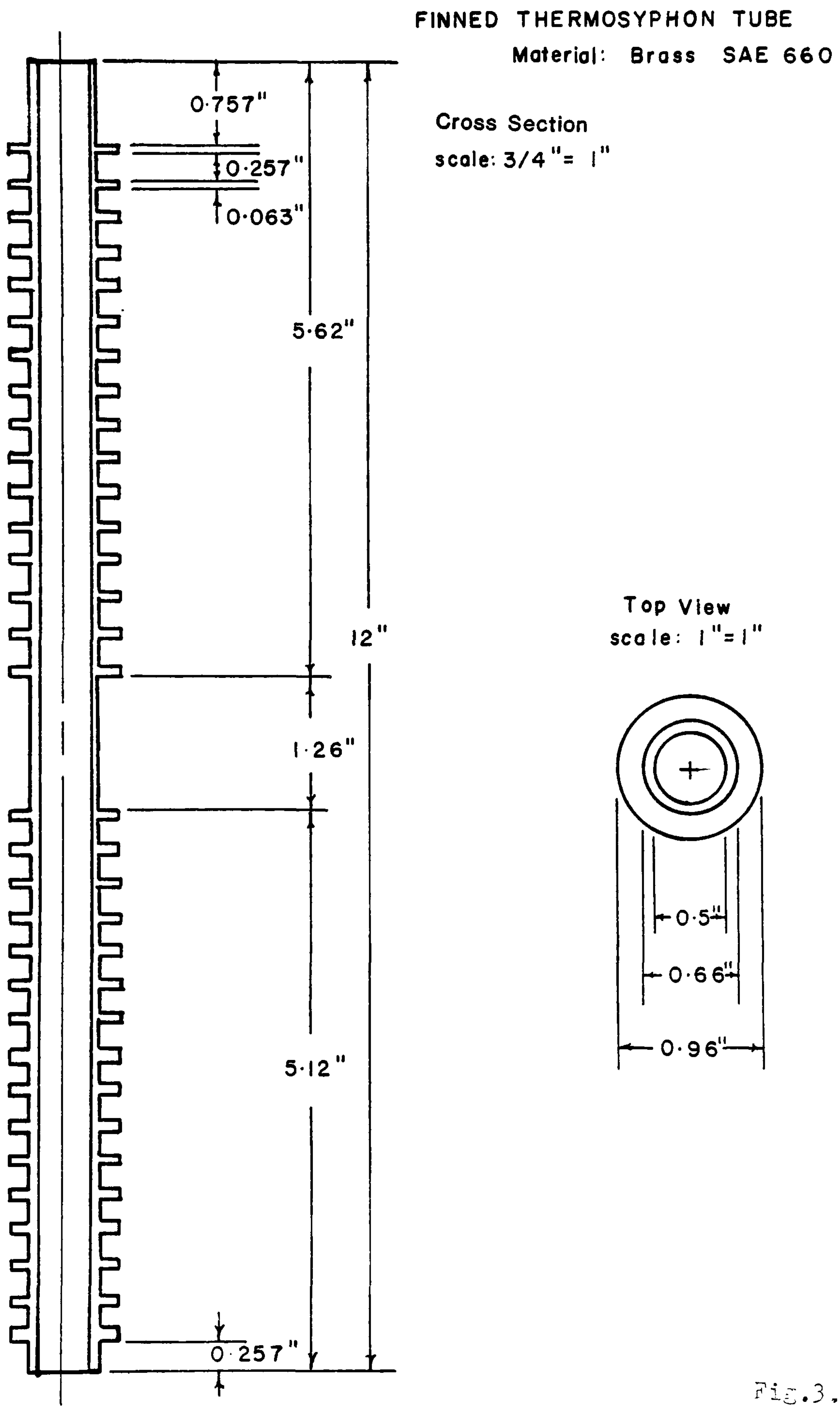
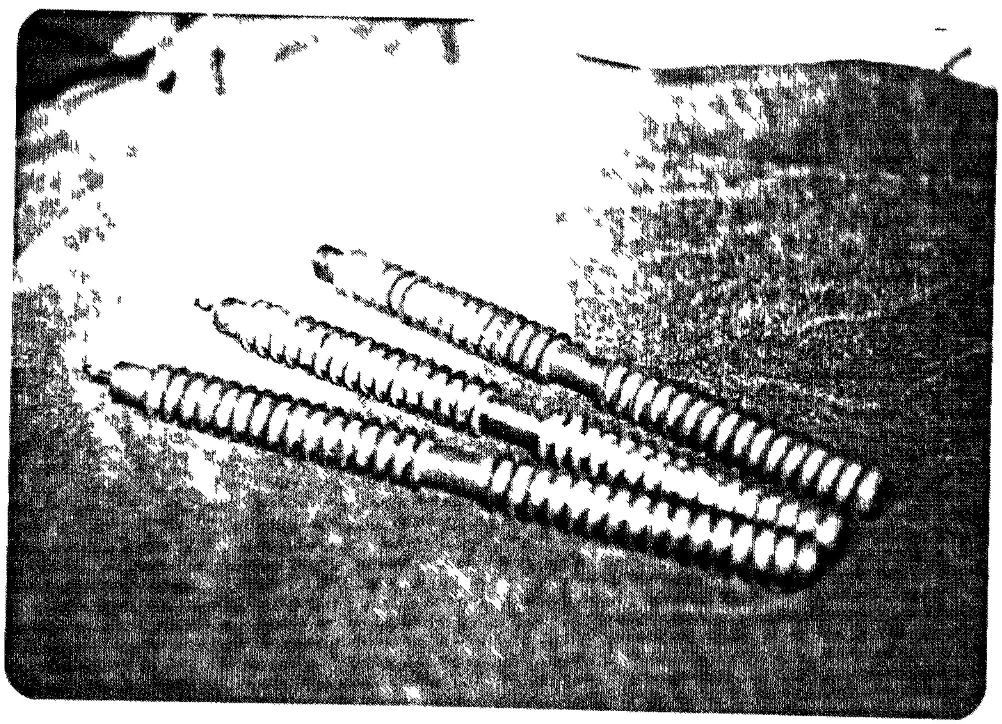
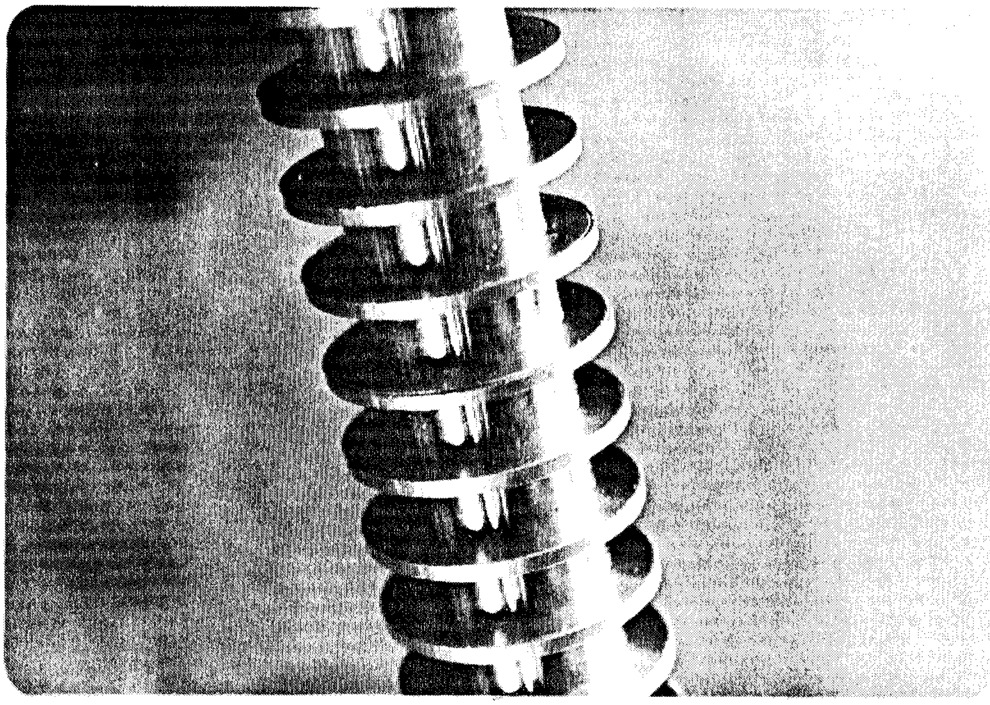

FINNED THERMOSYPHON TUBE
Material: Brass SAE 660
Cross Section
scale: 3/4"= 1"
0.757"
0.257"
0.063"
5.62"
12"
1.26"
5.12"
0.257"
Top View
scale: 1"=1"
0.5"
0.66"
0.96"

Fig.3.5



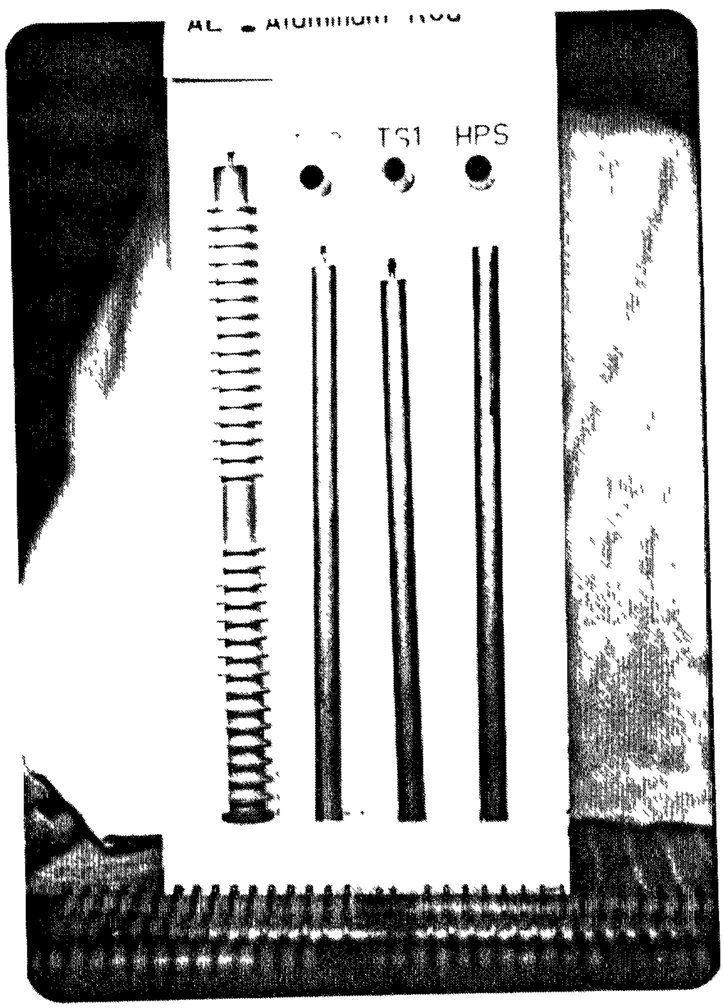
Three Finned Thermosyphon Tubes

Fig.3.6



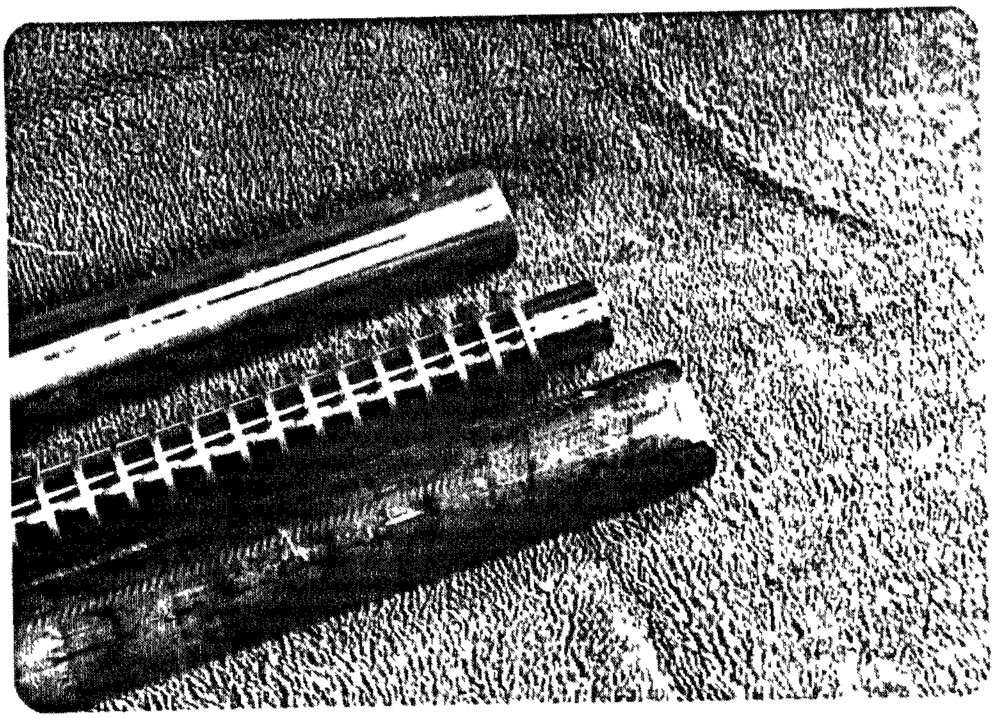
Closeup of Machined Finned Surface

Fig.3.7




From Left to Right: Finned Thermosyphon Tubes
Two Bare Thermosyphons
Heat Pipe

Fig.3.



Various Stages of Machining:

Cast Rod
Machined Rod
Finned Tube

Fig.3.[illegible]

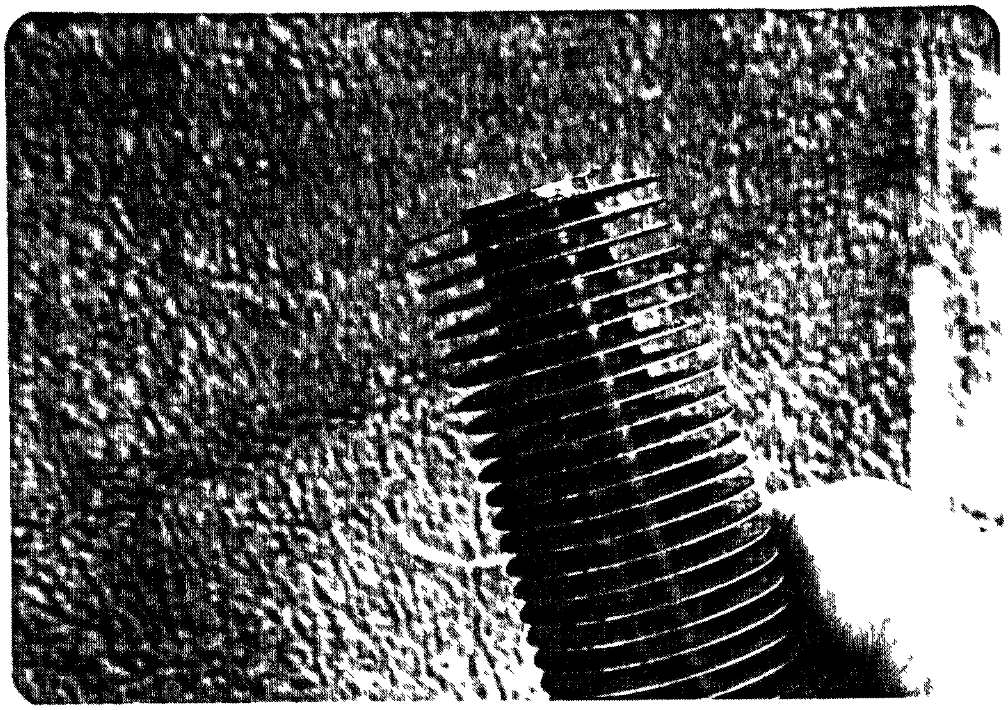


Fig.3.10

Commercially Available Helically Finned Tube

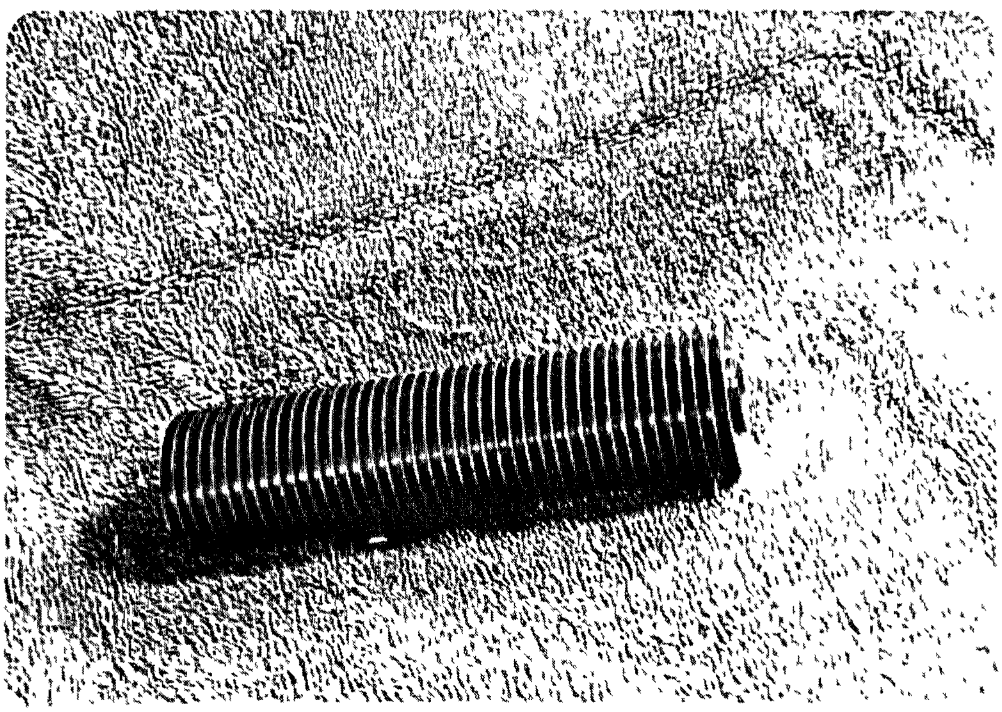


Fig.3.11

Commercially Available Helically Finned Tube

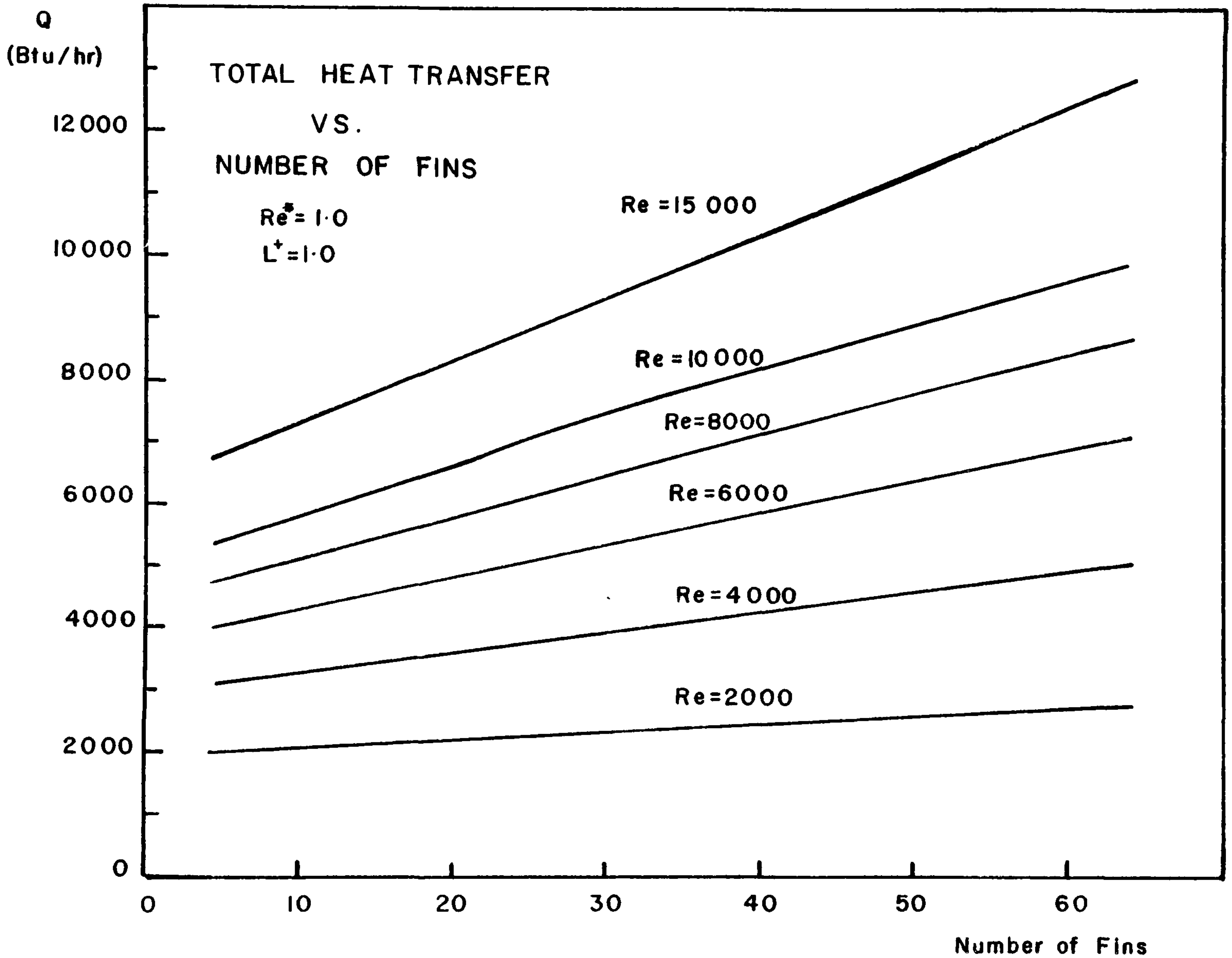

Q
(Btu/hr)
TOTAL HEAT TRANSFER
VS.
NUMBER OF FINS
Re* = 1·0
L+ = 1·0
Re = 15 000
Re = 10 000
Re = 8000
Re = 6000
Re = 4000
Re = 2000
12000
10000
8000
6000
4000
2000
0
0
10
20
30
40
50
60
Number of Fins

Fig.4.1

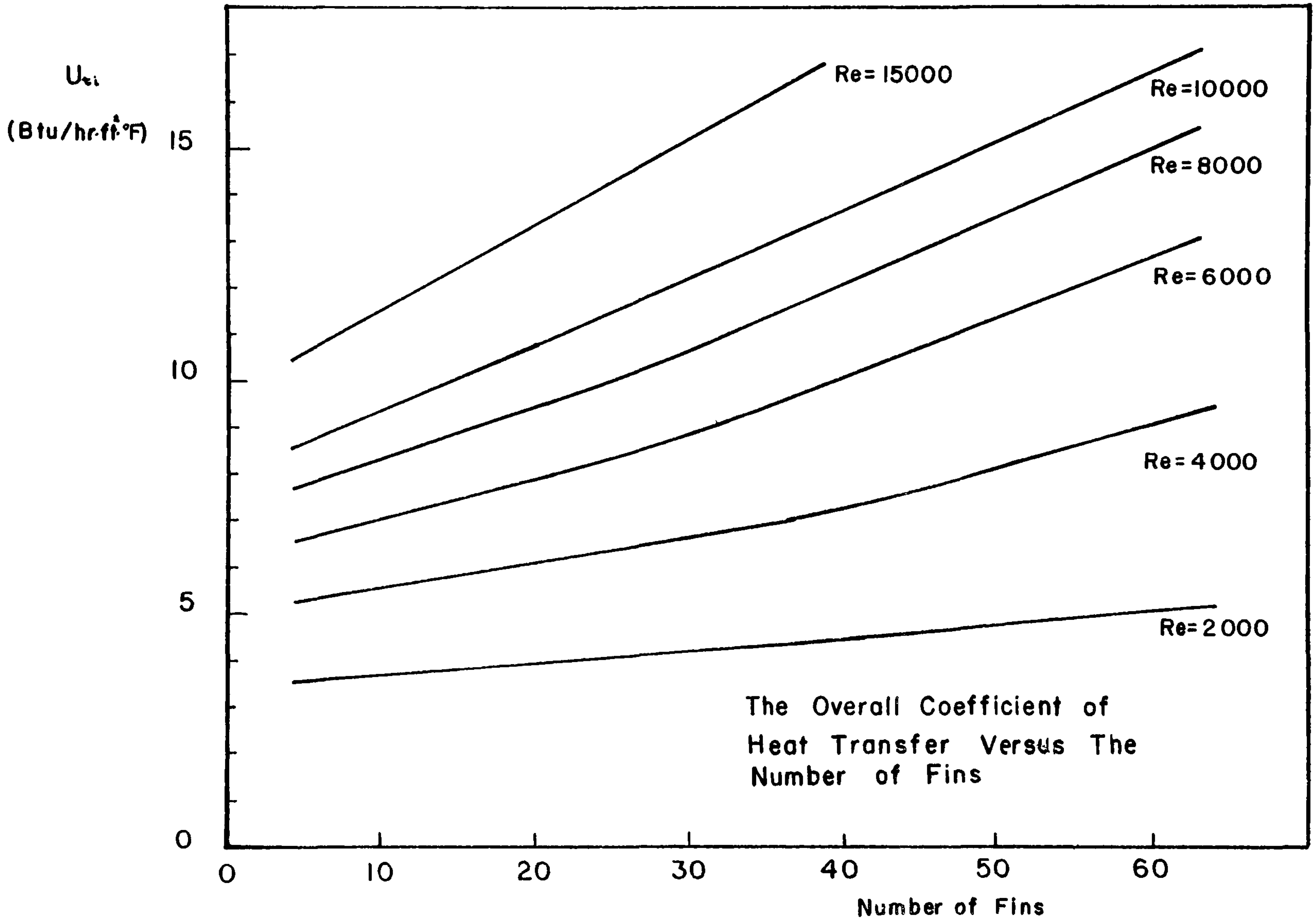
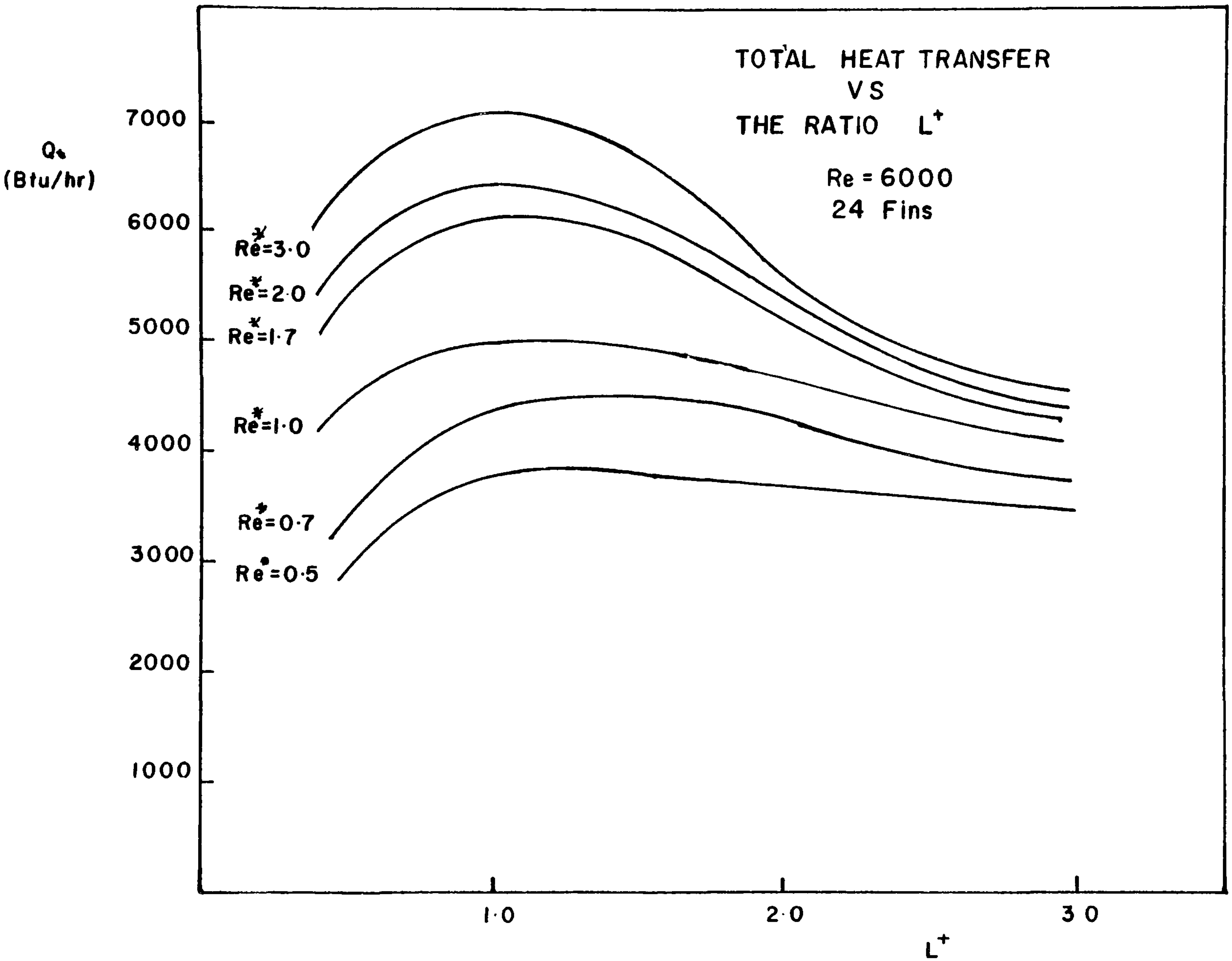

U
(Btu/hr·ft²·°F)
15
10
5
0
0
10
20
30
40
50
60
Number of Fins
Re= 15000
Re=10000
Re=8000
Re=6000
Re=4000
Re=2000
The Overall Coefficient of
Heat Transfer Versus The
Number of Fins

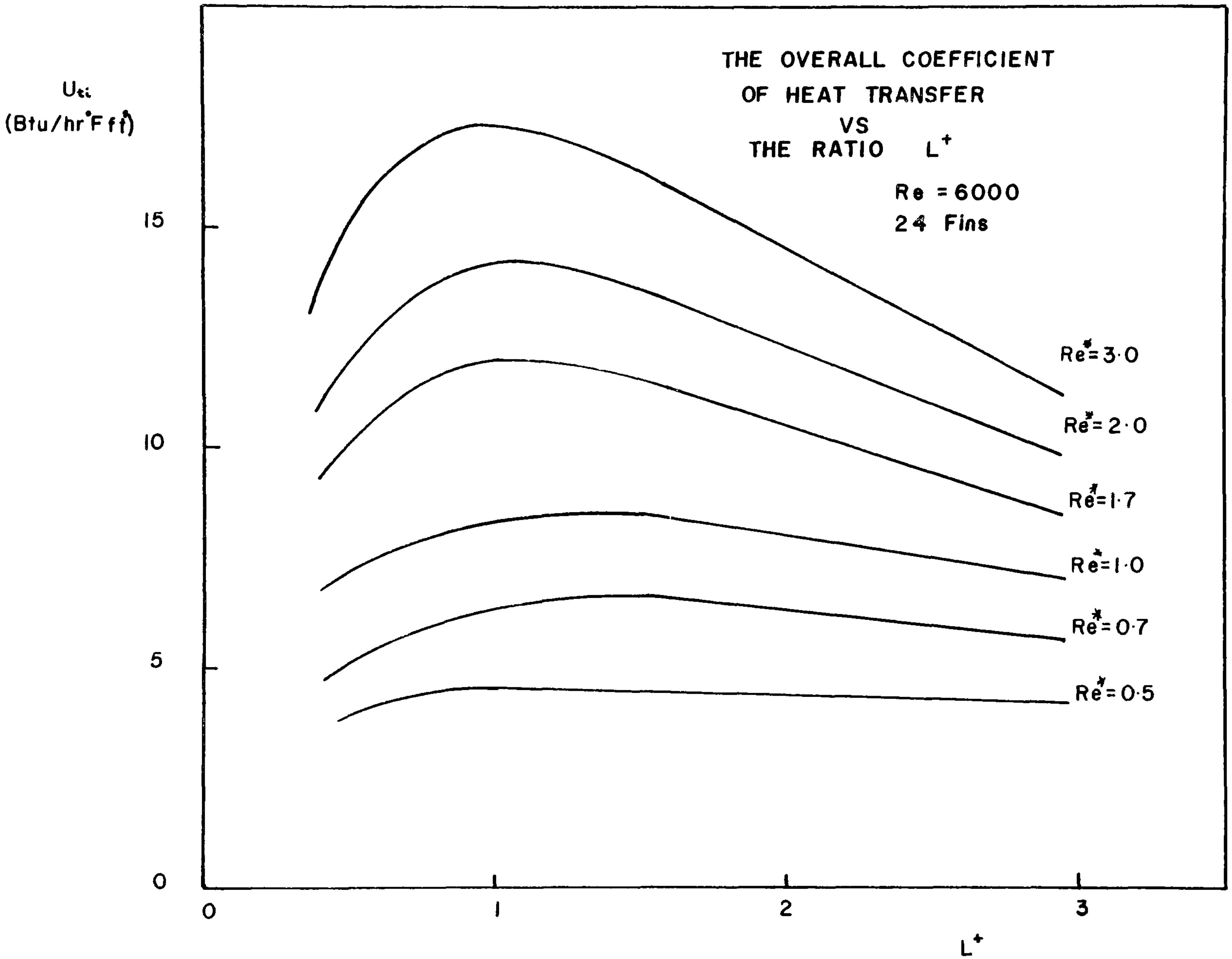
Fig.4.2



TOTAL HEAT TRANSFER
VS
THE RATIO L+
Re = 6000
24 Fins
Q
(Btu/hr)
7000
6000
5000
4000
3000
2000
1000
Re*=3·0
Re*=2·0
Re*=1·7
Re*=1·0
Re*=0·7
Re*=0·5
1·0
2·0
3 0
L+

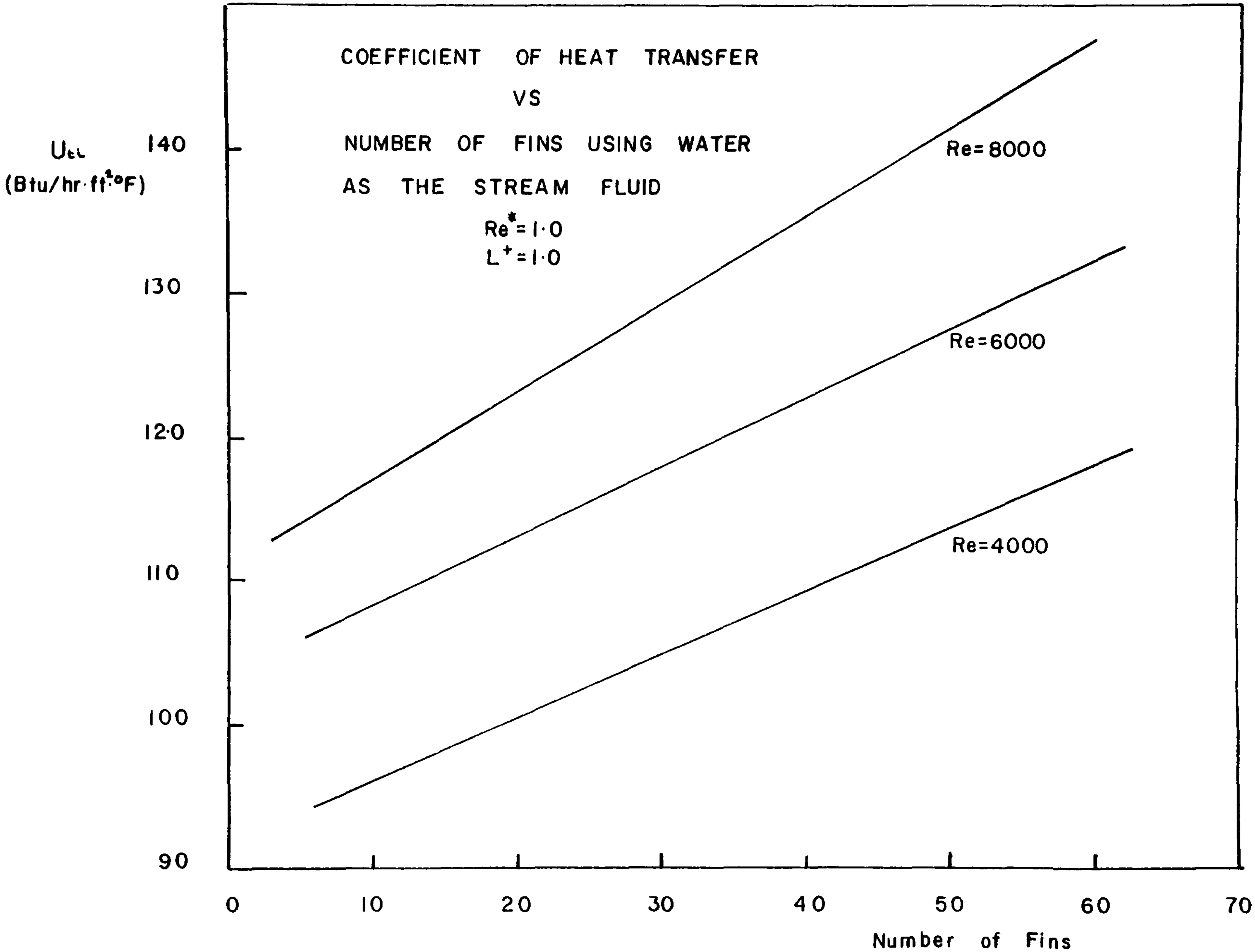



Fig.4.3



THE OVERALL COEFFICIENT
OF HEAT TRANSFER
VS
THE RATIO L+
Re = 6000
24 Fins
Uti
(Btu/hr°F ft²)
15
10
5
0
0
1
2
3
L+
Re* = 3·0
Re* = 2·0
Re* = 1·7
Re* = 1·0
Re* = 0·7
Re* = 0·5


Fig.4.4



COEFFICIENT OF HEAT TRANSFER
VS
NUMBER OF FINS USING WATER
AS THE STREAM FLUID
Re* = 1·0
L+ = 1·0
Re=8000
Re=6000
Re=4000
Utt
(Btu/hr·ft²·°F)
140
130
120
110
100
90
0
10
20
30
40
50
60
70
Number of Fins


Fig.4.5

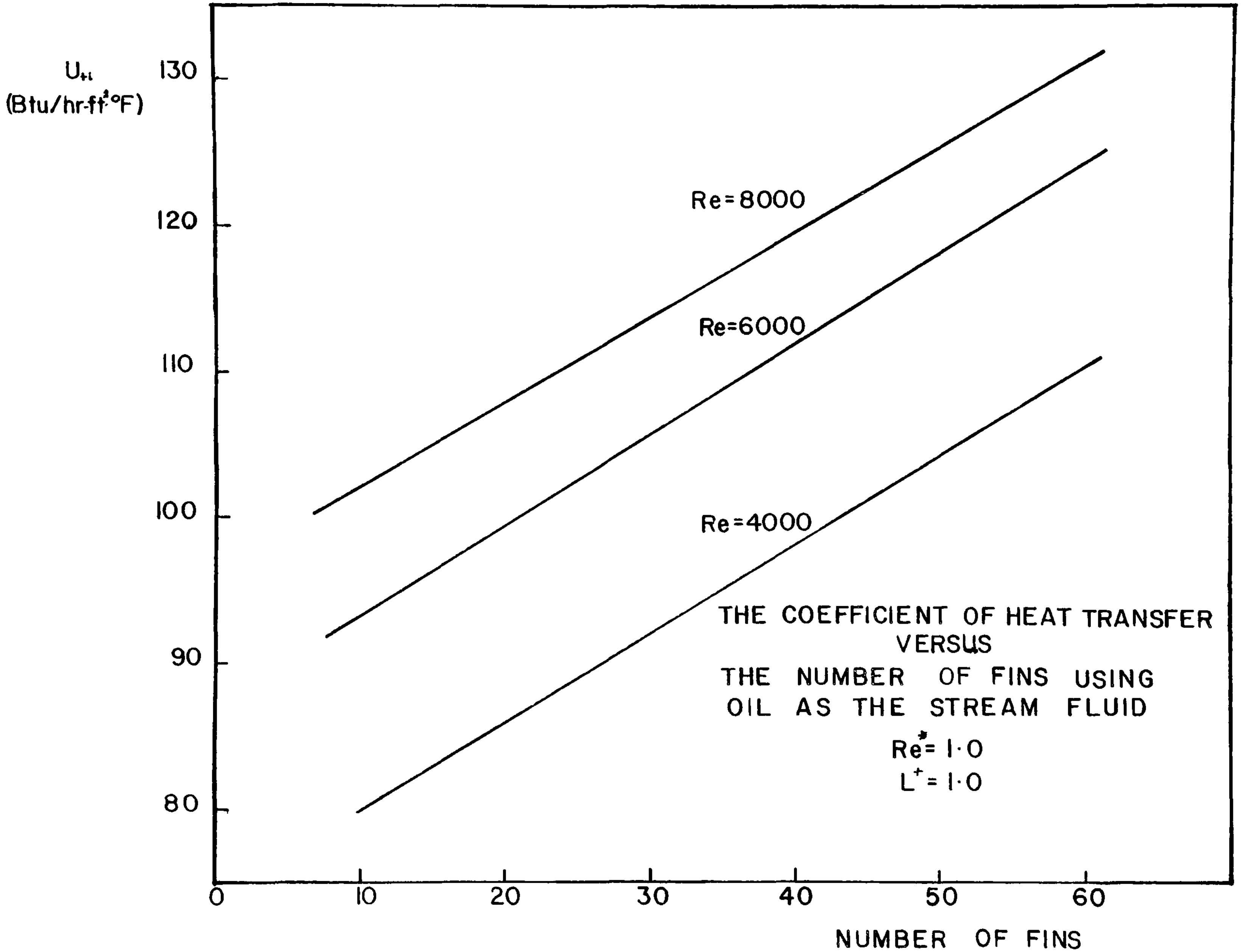
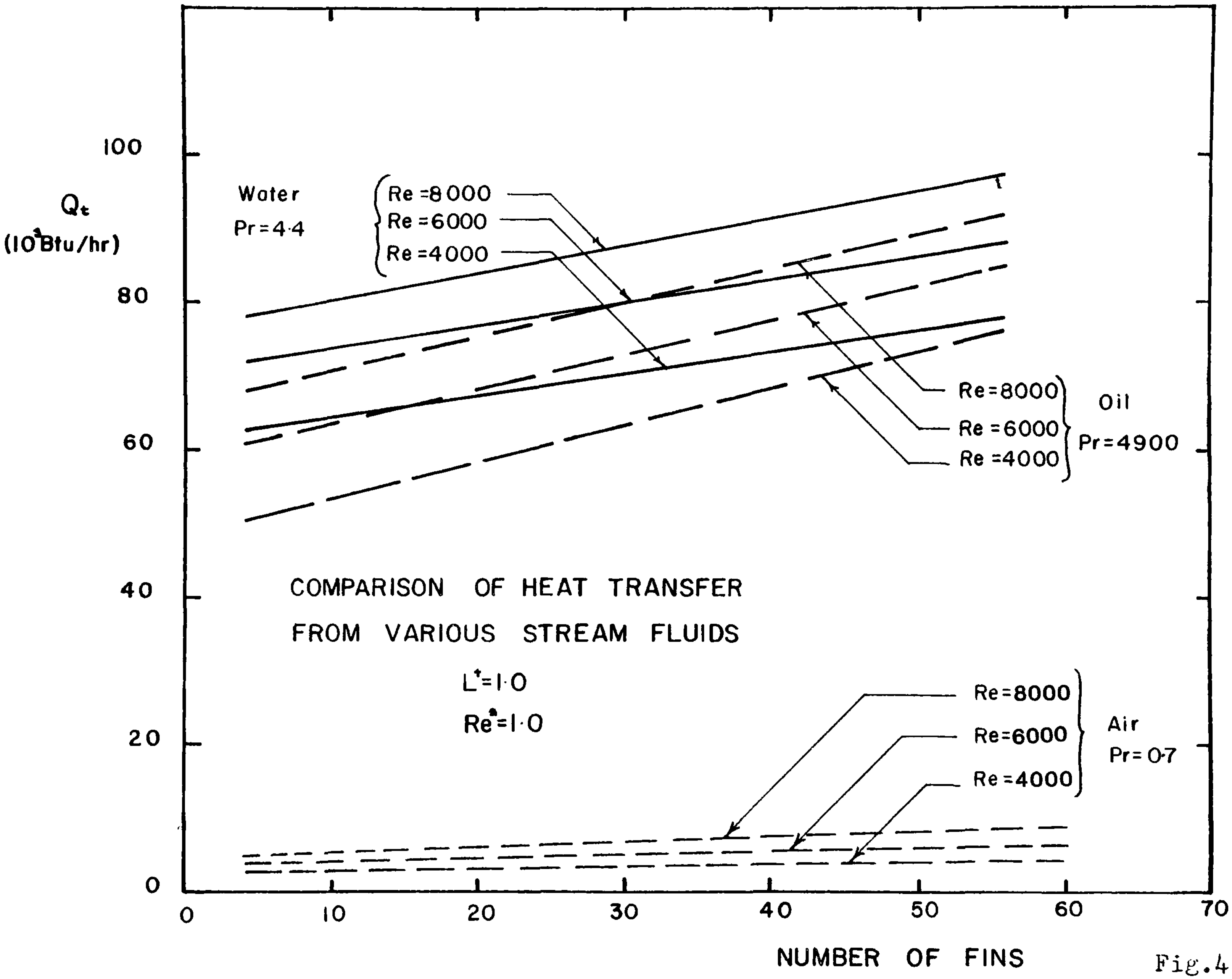

U
(Btu/hr-ft²°F)
130
120
110
100
90
80
0
10
20
30
40
50
60
NUMBER OF FINS
Re=8000
Re=6000
Re=4000
THE COEFFICIENT OF HEAT TRANSFER
VERSUS
THE NUMBER OF FINS USING
OIL AS THE STREAM FLUID
Re*= 1·0
L+= 1·0

Fig.4.6


Water
Pr=4·4
Re=8000
Re=6000
Re=4000
$Q_t$
($10^3$Btu/hr)
100
80
60
40
20
0
Re=8000
Re=6000
Re=4000
Oil
Pr=4900
COMPARISON OF HEAT TRANSFER
FROM VARIOUS STREAM FLUIDS
$L^+$=1·0
$Re^*$=1·0
Re=8000
Re=6000
Re=4000
Air
Pr=0·7
0
10
20
30
40
50
60
70
NUMBER OF FINS
Fig.4.7

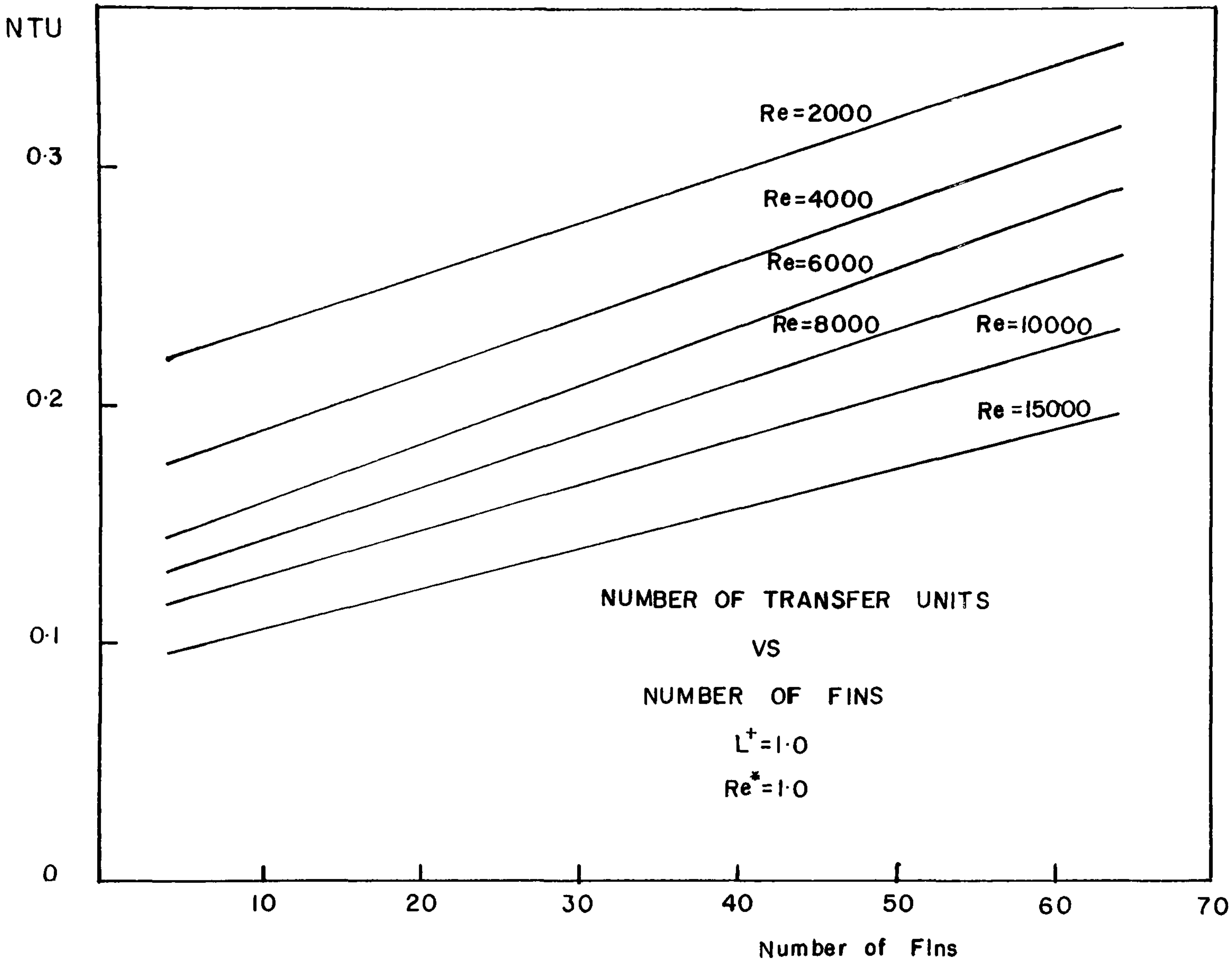
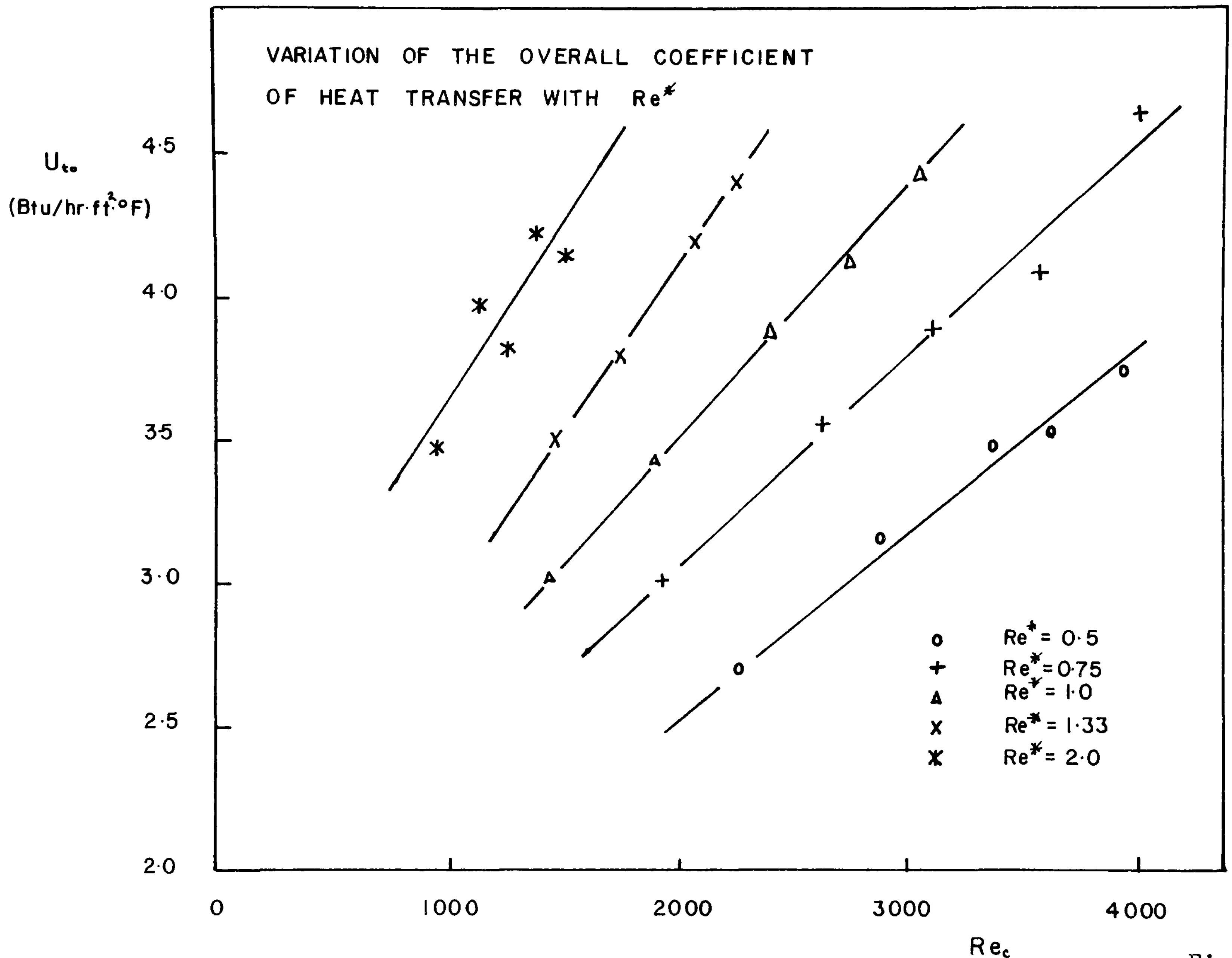
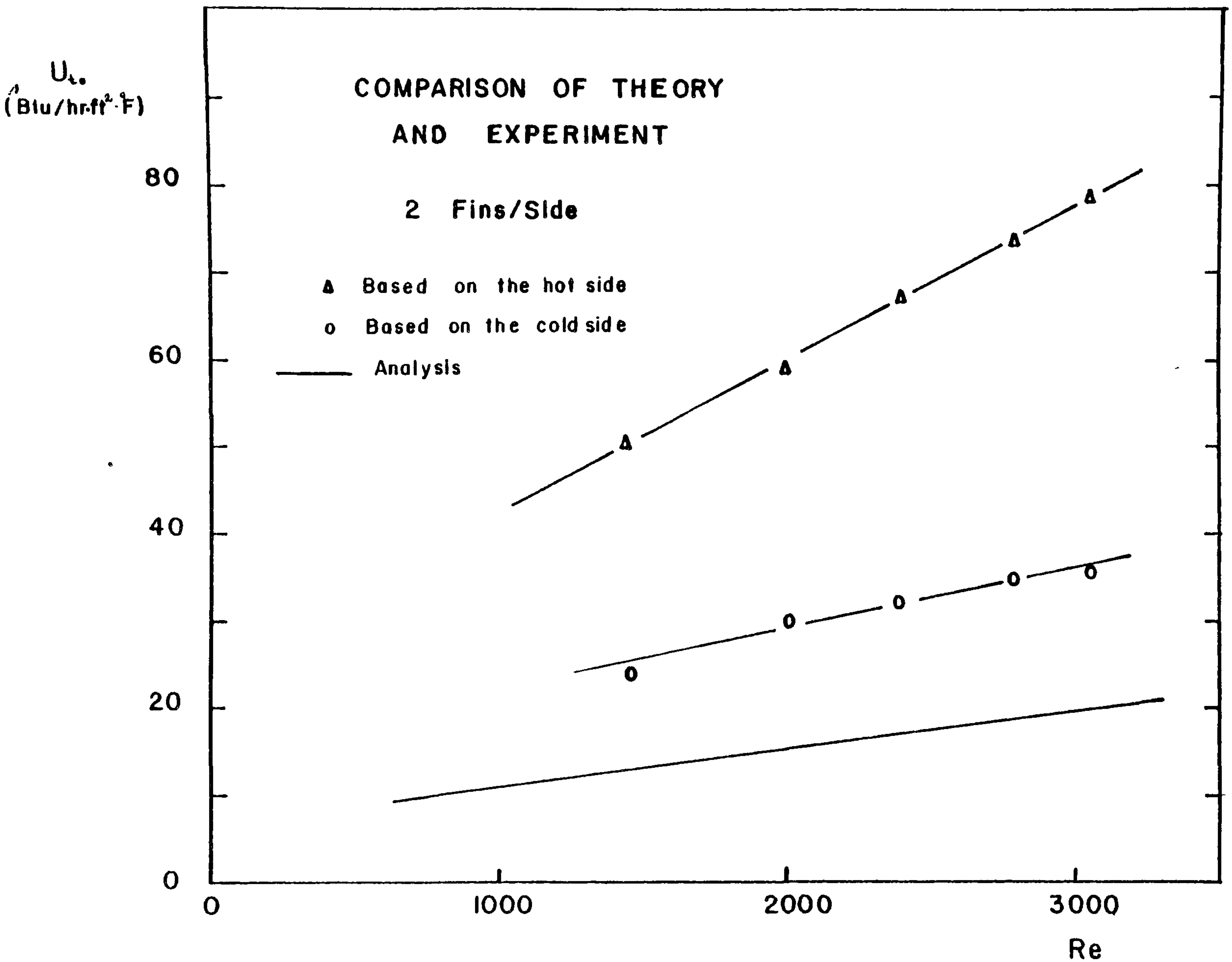

NTU
0·3
0·2
0·1
0
Re=2000
Re=4000
Re=6000
Re=8000
Re=10000
Re =15000
NUMBER OF TRANSFER UNITS
VS
NUMBER OF FINS
$L^{+}=1{\cdot}0$
$Re^{*}=1{\cdot}0$
10
20
30
40
50
60
70
Number of Fins

Fig.4.8



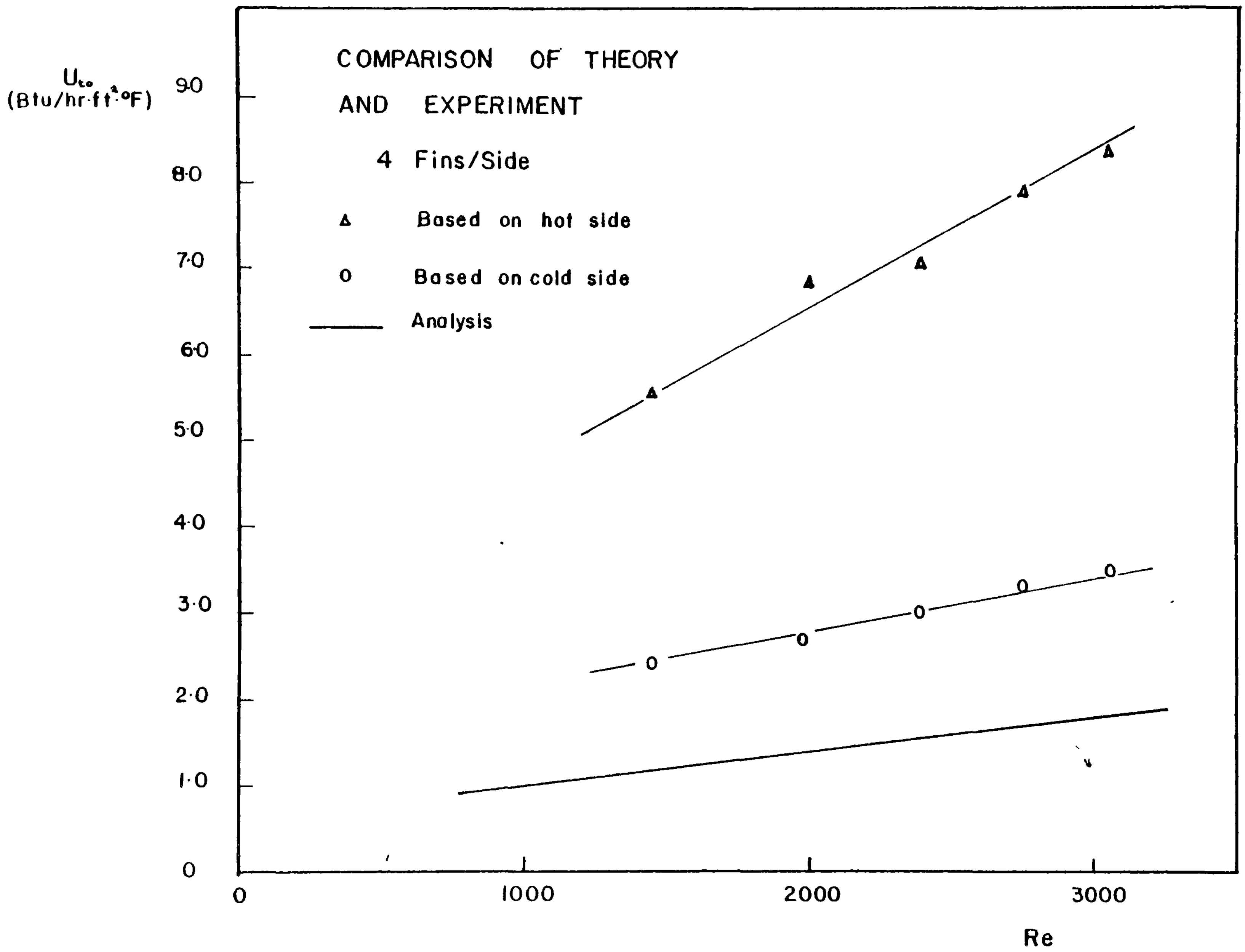
VARIATION OF THE OVERALL COEFFICIENT
OF HEAT TRANSFER WITH Re*
U_to
(Btu/hr·ft²·°F)
4·5
4·0
3·5
3·0
2·5
2·0
0
1000
2000
3000
4000
Re_c
o Re* = 0·5
+ Re* = 0·75
Δ Re* = 1·0
x Re* = 1·33
✱ Re* = 2·0


Fig.4.9



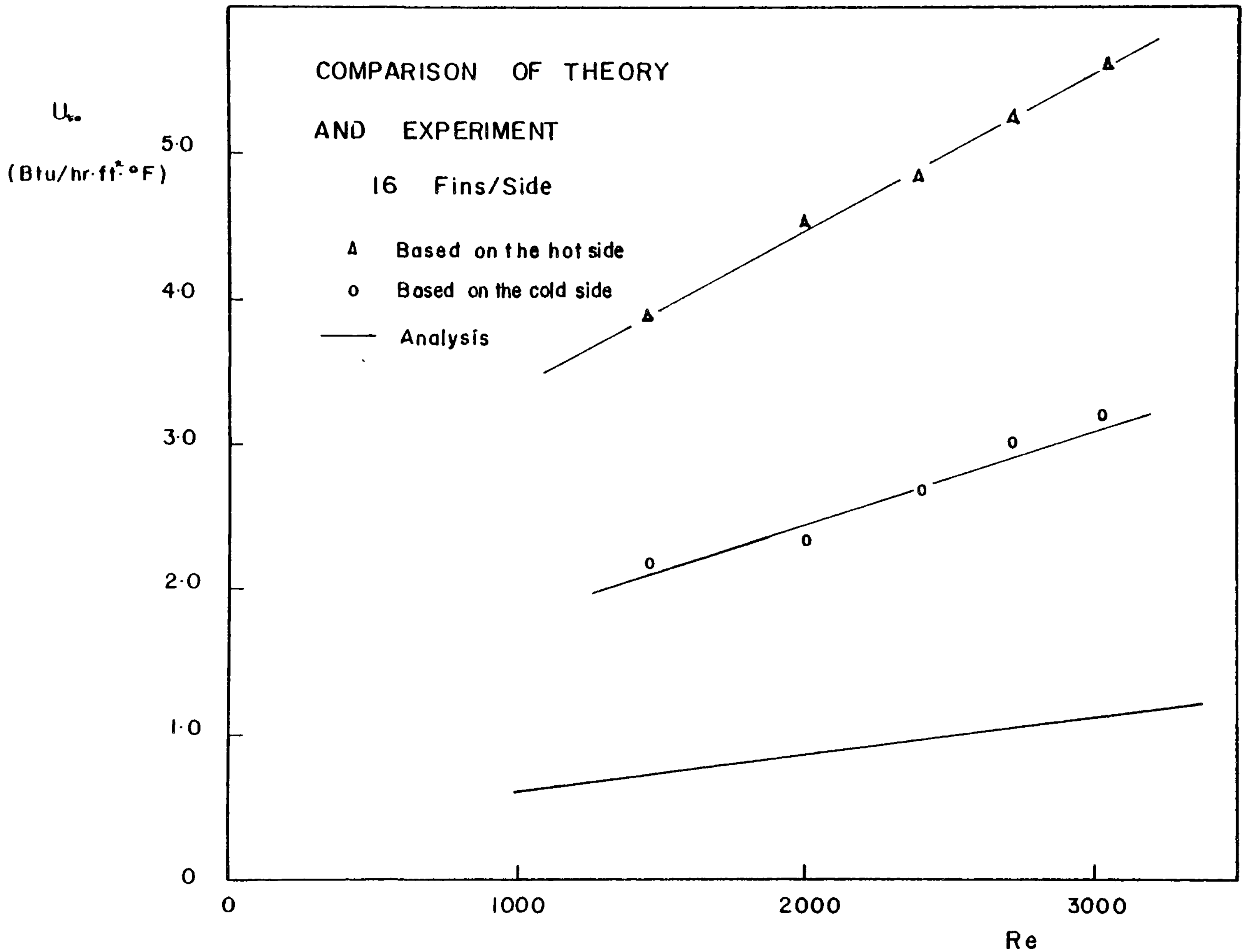
$U_{t}$ (Btu/hr·ft²·°F)
COMPARISON OF THEORY
AND EXPERIMENT
2 Fins/Side
Δ Based on the hot side
o Based on the cold side
Analysis
80
60
40
20
0
0
1000
2000
3000
Re


Fig.4.10



$U_{to}$ (Btu/hr·ft²·°F)
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0
COMPARISON OF THEORY AND EXPERIMENT
4 Fins/Side
Δ Based on hot side
o Based on cold side
Analysis
0
1000
2000
3000
Re


Fig.4.11



COMPARISON OF THEORY
AND EXPERIMENT
16 Fins/Side
Based on the hot side
Based on the cold side
Analysis
$U_{t\circ}$
(Btu/hr·ft²·°F)
5·0
4·0
3·0
2·0
1·0
0
0
1000
2000
3000
Re


Fig.4.12

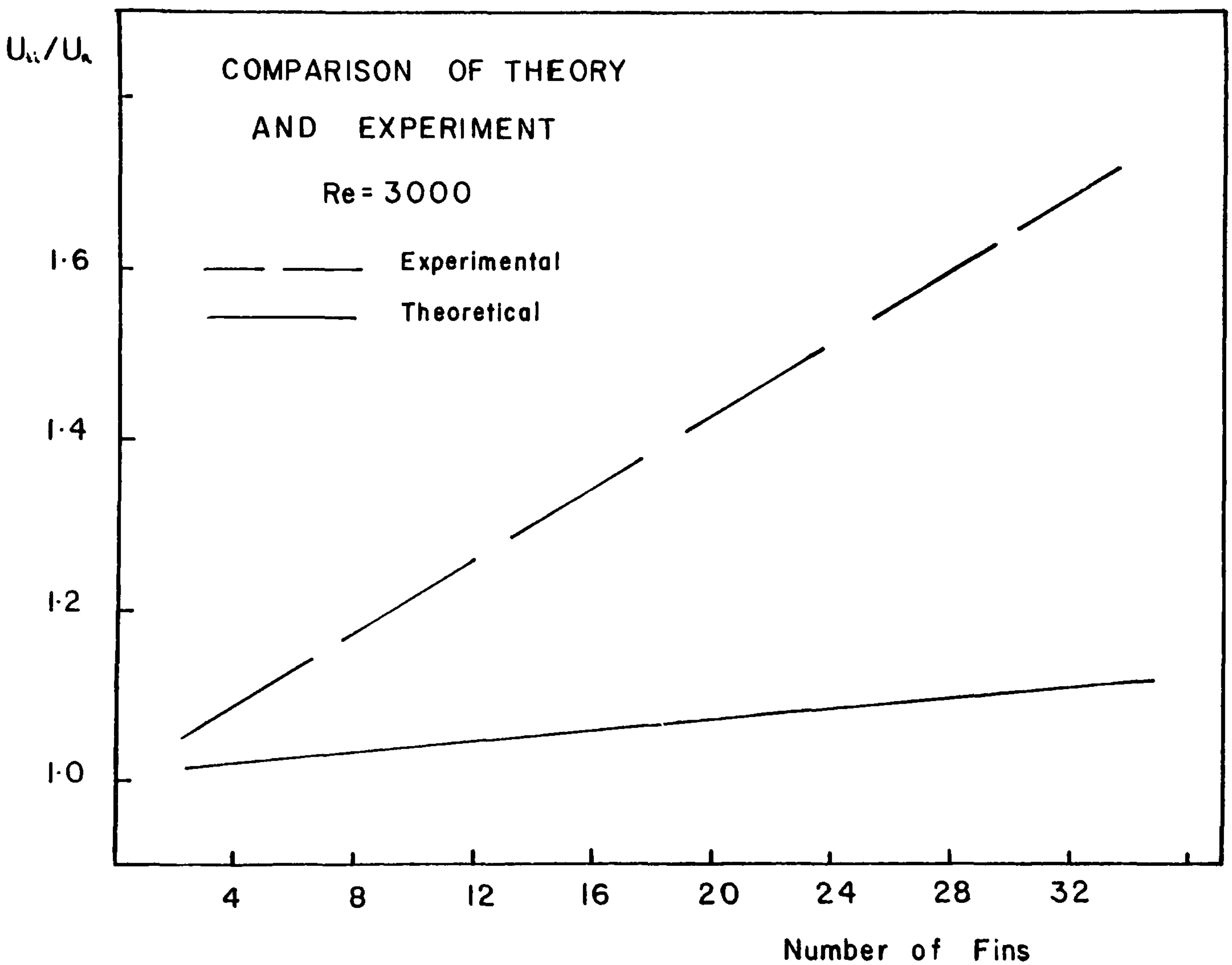

COMPARISON OF THEORY
AND EXPERIMENT
Re = 3000
Experimental
Theoretical
1·6
1·4
1·2
1·0
4
8
12
16
20
24
28
32
Number of Fins

Fig.4.13

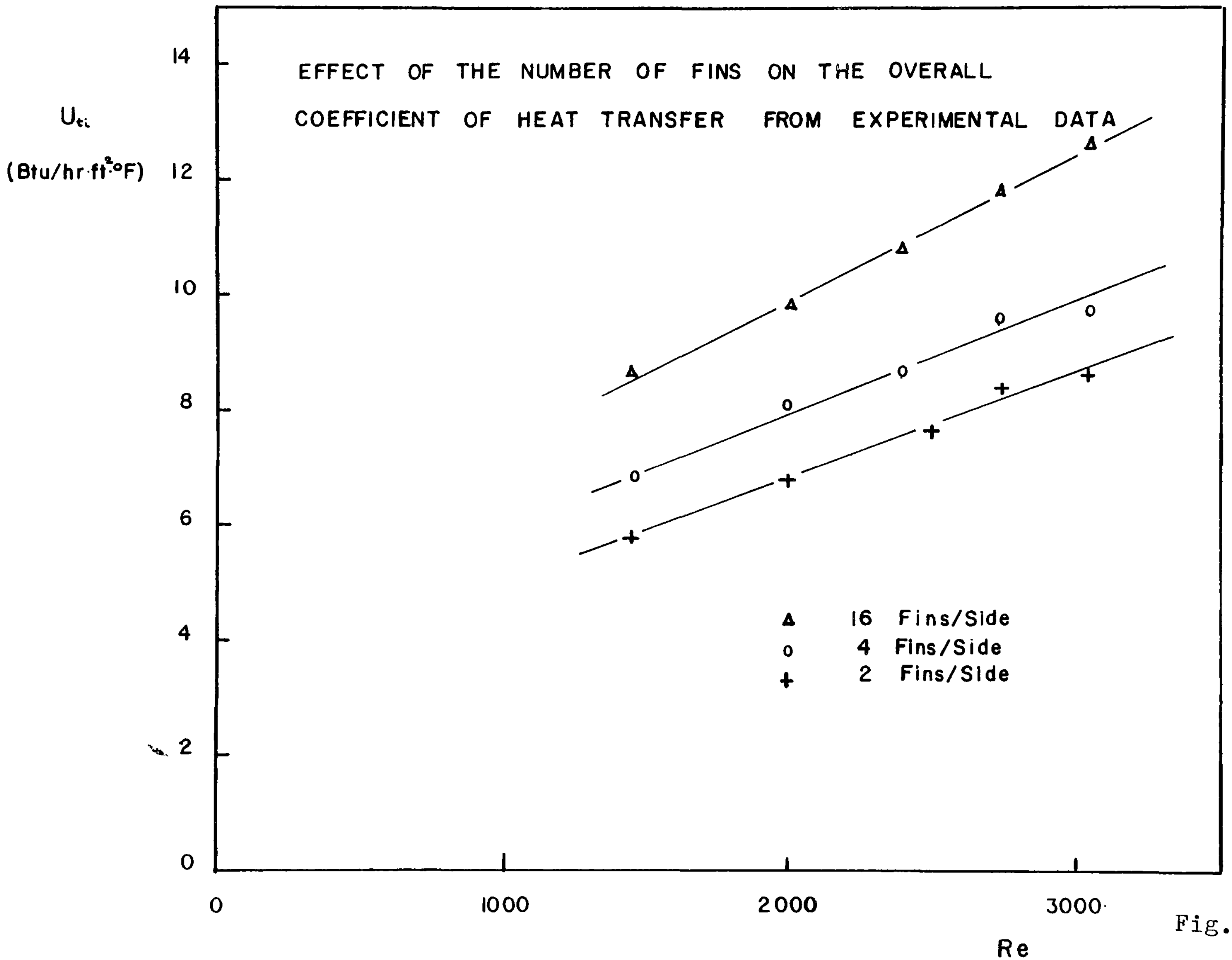

EFFECT OF THE NUMBER OF FINS ON THE OVERALL COEFFICIENT OF HEAT TRANSFER FROM EXPERIMENTAL DATA
$U_{ti}$ (Btu/hr·ft²·°F)
0
2
4
6
8
10
12
14
0
1000
2000
3000
Re
16 Fins/Side
4 Fins/Side
2 Fins/Side

Fig.4.14

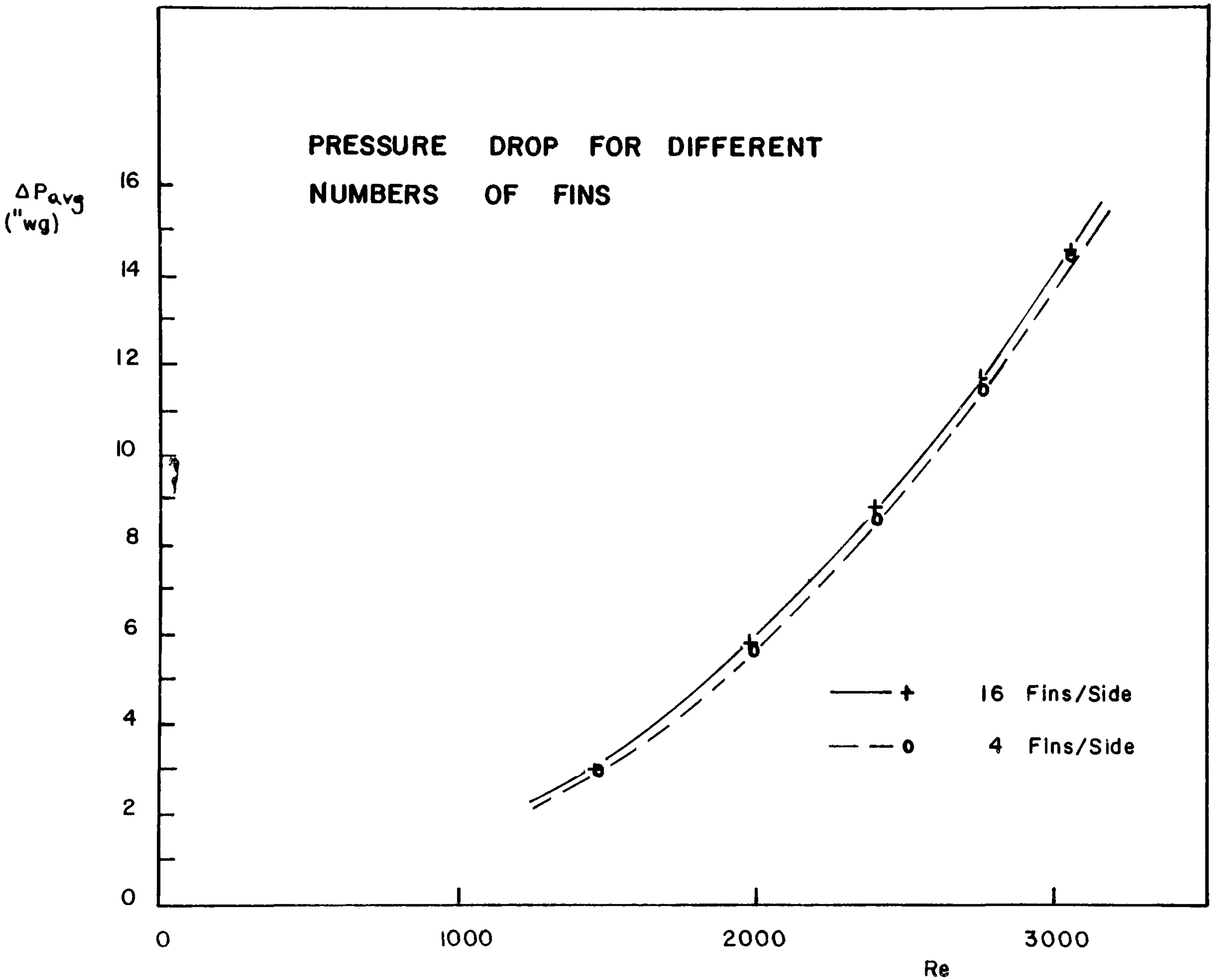

PRESSURE DROP FOR DIFFERENT
NUMBERS OF FINS
ΔPavg
("wg)
16
14
12
10
8
6
4
2
0
0
1000
2000
3000
Re
16 Fins/Side
4 Fins/Side

Fig.4.15

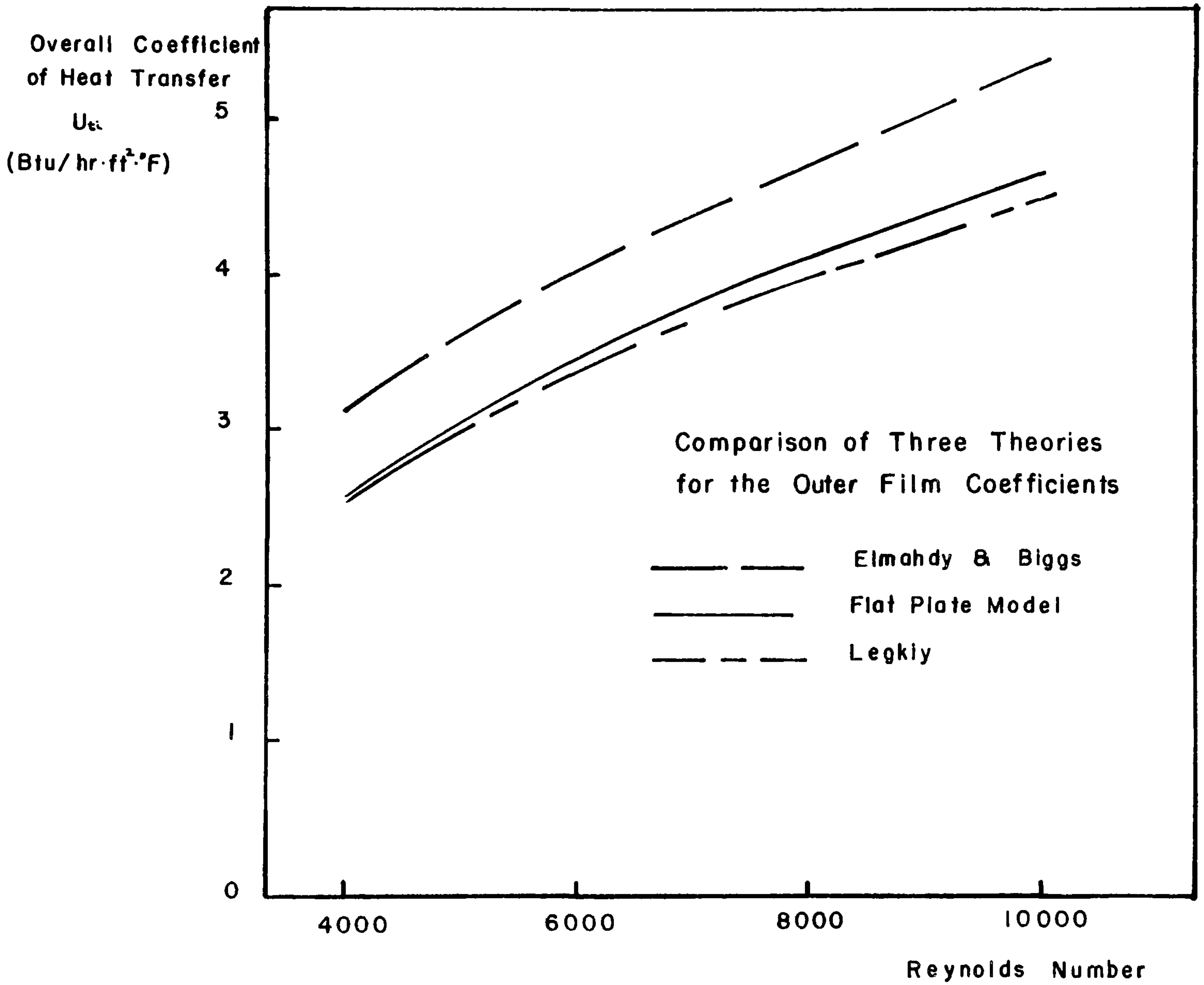

Overall Coefficient
of Heat Transfer
U
(Btu/hr·ft²·°F)
5
4
3
2
1
0
4000
6000
8000
10000
Reynolds Number
Comparison of Three Theories
for the Outer Film Coefficients
Elmahdy & Biggs
Flat Plate Model
Legkiy

Fig.C.1

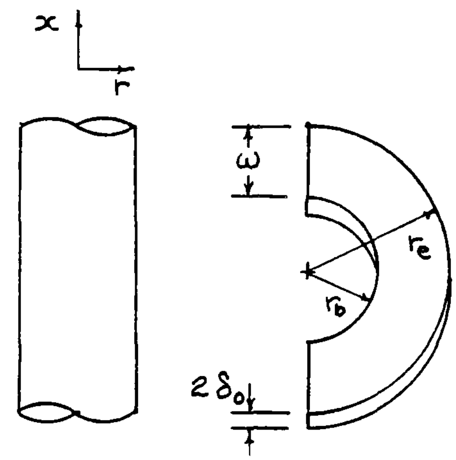
# APPENDIX A

## The Derivation of the Fin Efficiency of an Annular Fin of Rectangular Profile

# APPENDIX A

## The Derivation of the Fin Efficiency of an Annular Fin of Rectangular Profile (22)

Fig. A.1




Assumptions:

1) Heat transfer rate may be taken to depend upon the radial direction totally.
2) The temperature at the base of the fin is constant.

   ( $T_o$ = constant)
3) The outside film coefficient is constant.
4) The temperature gradient in the x direction is zero

Let $$\theta = \Delta T = T - T_\infty \qquad (A.1)$$

where $T_\infty$ is the temperature of the surrounding fluid.

Upon differentiation with respect to the radius one obtains

$$d\theta/dr = d(\Delta T)/dr = dT/dr \tag{A.2}$$

The Fourier equation of heat conduction states that

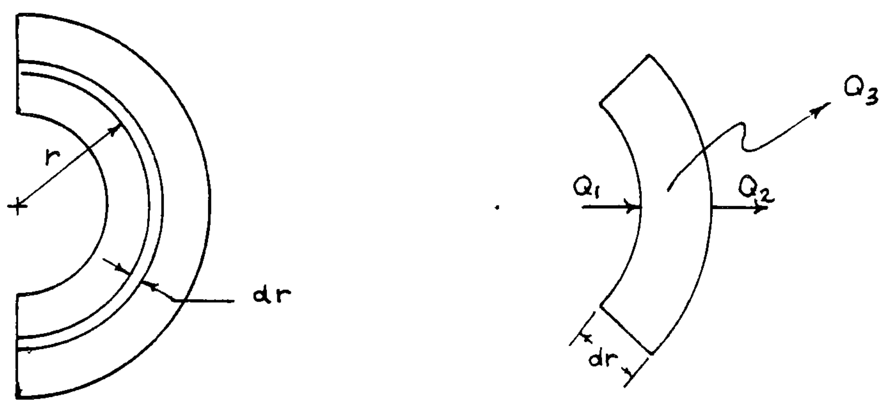
$$Q = -kA\,dT/dr \tag{A.3}$$

Therefore

$$Q = -kA\,d\theta/dr \tag{A.4}$$

Consider the radial heat flow through a differential element

Fig. A.2




Conduction into the element is equal to convection from the element and conduction out of the element.

$$Q_1 = Q_2 + Q_3 \tag{A.5}$$

$$q_r A = q_r A + \frac{d}{dr}(q_r A)\,dr + q_c P\,dr \tag{A.6}$$

where

$$q_r = -k\frac{d\theta}{dr} \quad ; \quad q_c = h_f\theta \tag{A.7 \& A.8}$$

Therefore $$0 = \frac{d}{dr}\left(k\frac{d\theta}{dr}\ 2\pi r(2\delta_o)\right) dr + 4\pi r\,dr\,h_f\theta \tag{A.9}$$

$$0 = -k\delta_o\left(\frac{r d^2\theta}{dr^2} + \frac{d\theta}{dr}\right) + h_f\theta r\,dr \tag{A.10}$$

$$0 = \delta_o\frac{d^2\theta}{dr^2} + \delta_o\frac{d\theta}{dr} - \frac{h_f}{k}\theta \tag{A.11}$$

$$0 = r^2\frac{d^2\theta}{dr^2} + r\frac{d\theta}{dr} - \frac{h_f}{k\delta}r^2\theta \tag{A.12}$$

By letting $$N = (h_f/k\delta_o)^{1/2} \tag{A.13}$$

$$0 = r^2\frac{d^2\theta}{dr^2} + r\frac{d\theta}{dr} - N^2r^2\theta \tag{A.14}$$

This is a Bessel equation of the second kind. It can be compared to the Douglass equation.

$$0 = x^2\frac{d^2y}{dx^2} + \left[(1-2A)x - 2Bx^2\right]\frac{dy}{dx} + \left[C^2D^2x^{2C} + B^2x^2 - B(1-2A)x + A^2 - C^2n^2\right]y \tag{A.15}$$

whose solution is

$$y = x^A e^{Bx}\left\{C_1 J_n(Dx^C) + C_2 Y_n(Dx^C)\right\} \tag{A.16}$$

where
$A = 0$
$B = 0$
$C = 1$
$D = \sqrt{-N^2} = Ni$
$n = 0$

Therefore the solution is

$$\theta = C_1 J_0(iNr) + C_2 Y_0(iNr) \tag{A.17}$$

at $\quad r = r_b \qquad \theta = \theta_b$

and at $\quad r = r_e \qquad \frac{d\theta}{dr} = 0$

Differentiating equation A.17 we get

$$\frac{d\theta}{dr} = C_1 N J_1(iNr) - C_2 N Y_1(iNr) \tag{A.18}$$

or

$$d\theta = C_1 N I_1(Nr) - C_2 N K_1(Nr) \tag{A.19}$$

Inserting the first boundary condition into the equation A.17 we get

$$\theta_b = C_1 I_0(Nr_b) + C_2 K_0(Nr_b) \tag{A.20}$$

then putting the second boundary condition into equation A.19 we get

$$0 = C_1 I_1(Nr_e) - C_2 K_1(Nr_e) \tag{A.21}$$

Therefore

$$C_1 = C_2 \left[ \frac{K_1(Nr_e)}{I_1(Nr_e)} \right] \tag{A.22}$$

Putting this result into equation A.20 we get

$$\theta_b = C_2 \left[ \frac{K_1(Nr_e)}{I_1(Nr_e)} \right] I_0(Nr_b) + C_2 K_0(Nr_b) \tag{A.23}$$

Solving for C

$$C_2 = \frac{\theta_b I_1(Nr_e)}{K_1(Nr_e) I_0(Nr_b) + I_1(Nr_e) K_0(Nr_b)} \qquad (A.24)$$

Putting this result into equation A.23 we obtain

$$C_1 = \frac{\theta_b K_1(Nr_e)}{K_1(Nr_e) I_0(Nr_b) + I_1(Nr_e) K_0(Nr_b)} \qquad (A.25)$$

We put the values of $C_1$ and $C_2$ into equation A.17 which can be rewritten as

$$\theta = C_1 I_0(Nr) + C_2 K_0(Nr) \qquad (A.26)$$

and get

$$\frac{\theta}{\theta_b} = \frac{K_1(Nr_e) I_0(Nr) + I_1(Nr_e) K_0(Nr)}{K_1(Nr_e) I_0(Nr_b) + I_1(Nr_e) K_0(Nr_b)} \qquad (A.27)$$

The heat dissipated from the surfaces of the fin are found from the convection equation which is

$$\frac{dQ}{dr} = 4\pi h_f \theta r \qquad (A.28)$$

$$dQ = 4\pi h_f \theta r\, dr \qquad (A.29)$$

$$Q = 4\pi h_f \int_{r_b}^{r_e} \theta r\, dr \qquad (A.30)$$

It has been shown in equation A.14 that

$$r^2 \frac{d^2\theta}{dr^2} + r\frac{d\theta}{dr} - N^2 r^2 \theta = 0 \qquad (A.31)$$

or
$$N^2 r^2 \theta = r^2 \frac{d^2\theta}{dr^2} + r\frac{d\theta}{dr} \tag{A.32}$$

since
$$\frac{d}{dr}\left(r\frac{d\theta}{dr}\right) = r^2\frac{d^2\theta}{dr^2} + \frac{d\theta}{dr} \tag{A.33}$$

then
$$\frac{d}{dr}\left(r\frac{d\theta}{dr}\right) = N^2 r\theta \tag{A.34}$$

thus, one can write
$$\int_{r_b}^{r_e} \theta r dr = \frac{1}{N^2}\int_{r_b}^{r_e} \frac{d}{dr}\left(r\frac{d\theta}{dr}\right)dr \tag{A.35}$$

$$= \frac{1}{N^2}\left(r\frac{d\theta}{dr}\right)\Big|_{r_b}^{r_e} \tag{A.36}$$

$$= \frac{1}{N^2}\left\{r_e\left(\frac{d\theta}{dr}\right)_{r_e} - r_b\left(\frac{d\theta}{dr}\right)_{r_b}\right\} \tag{A.37}$$

since at

then
$$\int_{r_b}^{r_e} \theta r dr = \frac{-r_b}{N^2}\left(\frac{d\theta}{dr}\right)_{r_b} \tag{A.38}$$

to evaluate $d\theta/dr$ at $r_b$ we differentiate equation A.27 and get

$$\left(\frac{d\theta}{dr}\right)_{r_e} = \theta_b\left[\frac{NK_1(Nr_e)I_1(Nr_b) - NI_1(Nr_e)K_1(Nr_b)}{K_1(Nr_e)I_0(Nr_b) + I_1(Nr_e)K_0(Nr_b)}\right] \tag{A.39}$$

$$= \theta_b N \Omega \tag{A.40}$$

Therefore
$$\int_{r_b}^{r_e} \theta r dr = \left(\frac{-r_b}{N^2}\right)\left(\frac{d\theta}{dr}\right)_{r_b} \tag{A.41}$$

$$= \frac{-r_b}{N^2}\theta_b N\Omega \tag{A.42}$$

Putting this result into Equation (30)

$$Q = -\frac{4\pi h_f r_b}{N}\,\Theta_b\Omega \tag{A.43}$$

The efficiency of the fin is defined as the ratio of the heat transfer of the fin to that with the fin maintained at

$$\eta = \frac{Q_o}{Q_b} = \frac{4\pi h_f}{h_f A_f}\int_0^{A_f}\frac{\Theta\, dA}{\Theta_b} = \frac{4\pi}{\Theta_b A_f}\int_0^{A_f}\Theta\, dA \tag{A.44}$$

$$= \frac{4\pi}{\Theta_b A_f}\left(-\frac{r_b\Theta_b\Omega}{N}\right) \tag{A.45}$$

$$= -\frac{4\pi r_b\Omega}{A_f N} \tag{A.46}$$

Since

$$A \simeq 2\pi(r_e^2 - r_b^2) \tag{A.47}$$

then

$$\eta = -\frac{2 r_b\Omega}{N}\left\{\frac{1}{r_e^2 - r_b^2}\right\} \tag{A.48}$$

$$= -2\Omega\,\frac{u_b}{(u_e^2 - u_b^2)} \tag{A.49}$$

$$= \frac{2\,\Omega}{u_b\left(1 - \frac{u_e^2}{u_b^2}\right)} \tag{A.50}$$

# APPENDIX B

## Coefficients from Hilperts Equation

# APPENDIX B

## Coefficients from Hilperts Equation

| $s_t/d$ = | 1.25 | | 1.50 | | 2.00 | | 3.00 | |
|---|---|---|---|---|---|---|---|---|
| $s_l/d$ | B | m | B | m | B | m | B | m |
| **Staggered:** | | | | | | | | |
| 0.600 | ..... | ..... | ..... | ..... | ..... | ..... | 0.213 | 0.636 |
| 0.900 | ..... | ..... | ..... | ..... | 0.446 | 0.571 | 0.401 | 0.581 |
| 1.000 | ..... | ..... | 0.497 | 0.558 | ..... | ..... | ..... | ..... |
| 1.125 | ..... | ..... | ..... | ..... | 0.478 | 0.565 | 0.518 | 0.560 |
| 1.250 | 0.518 | 0.556 | 0.505 | 0.554 | 0.519 | 0.556 | 0.522 | 0.562 |
| 1.500 | 0.451 | 0.568 | 0.460 | 0.562 | 0.452 | 0.5[illegible] | 0.4[illegible] | 0.568 |
| 2.000 | 0.404 | 0.572 | 0.416 | 0.568 | 0.482 | 0.556 | 0.449 | 0.570 |
| 3.000 | 0.310 | 0.592 | 0.356 | 0.580 | 0.440 | 0.562 | 0.421 | 0.574 |
| **In Line :** | | | | | | | | |
| 1.250 | 0.348 | 0.592 | 0.275 | 0.608 | 0.100 | 0.704 | 0.063 | 0.752 |
| 1.500 | 0.367 | 0.566 | 0.250 | 0.620 | 0.101 | 0.702 | 0.068 | 0.744 |
| 2.000 | 0.418 | 0.570 | 0.299 | 0.602 | 0.229 | 0.632 | 0.198 | 0.648 |
| 3.000 | 0.290 | 0.601 | 0.357 | 0.584 | 0.374 | 0.581 | 0.286 | 0.608 |

| Row | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|
| $C_z$ for staggered tubes....... | 0.68 | 0.75 | 0.83 | 0.89 | 0.92 | 0.95 | 0.97 | 0.98 | 0.99 | 1.0 |
| $C_z$ for inline tubes....... | 0.64 | 0.80 | 0.87 | 0.90 | 0.92 | 0.94 | 0.96 | 0.98 | 0.99 | 1.0 |

# APPENDIX C

## A Comparison of Three Theories Involving Convection from an Annular Finned Surface

# APPENDIX C

## A Comparison of Three Theories Involving Convection from an Annular Finned Surface

To ensure the adequacy of the flat plate approximation used by the author, it was compared to two other theories for heat transfer from finned surfaces.

Elmahdy and Biggs ( 18 ) developed an expression for the Colburn J-factor as a function of the Reynolds number based on a hydraulic diameter. It was assumed that the J-factor was a function of the Reynolds number and of the form

$$J = C_1 Re^{C_2} \tag{C.1}$$

where $C_1$ and $C_2$ are constants dependent upon the physical dimensions of the heat exchanger. A multiple regression technique was used to determine the most influential coil dimensions. The equations for $C_1$ and $C_2$ which were developed are

$$C_1 = 0.159 \left[\frac{\omega}{2\delta_o}\right]^{0.141} \left[\frac{d_{hyd}}{2\delta_o}\right]^{0.065} \tag{C.2}$$

and

$$C_2 = 0.323 \left[\frac{2\delta_o}{\omega}\right]^{0.049} \left[\frac{\alpha}{2\delta_o}\right]^{0.077} \tag{C.3}$$

One can see that the relevant dimensions are the fin thickness, spacing, and height as well as the hydraulic diameter which is also included in the Reynolds number.

Legkiy, Pavlenko, Makarov and Zheludov ( 17) used arrays of miniature heat flux sensors to investigate local heat transfer coefficients on the surface of a tube with constant thickness annular fins cooled by a transverse air flow. The average heat transfer coefficient was defined as

$$h_{avg} = \sum_{1}^{6} \frac{h_j \xi_j \Delta T_j}{\Delta T_j} \tag{C.4}$$

where is the difference in temperature between a point on the surface and the bulk fluid flow temperature, and $\xi$ is the fraction of area covered by the section. These values were compared with the heat transfer coefficient calculated from

$$Nu = 0.18\left[1 + 93 I_1(A/302)\right] Re^{B} \tag{C.5}$$

where

$$A = \frac{0.5(d_e - d_o)\ell'}{d_o s} \tag{C.6}$$

$$B = \frac{12.9}{20.8 + 0.5\left(\frac{d_e - d_o}{d_o - 2\delta_o}\right)} \tag{C.7}$$

and

$$\ell' = \frac{2 d_o}{\pi}\left(\frac{d_e - d_o}{d_o + 2\delta_o} + 1\right) \tag{C.8}$$

and were shown to be in good agreement. The actual heat transfer coefficient equation that was used was originally taken from Legkiy,V.M.' In the collection: Problems of Heat Transfer and Thermodynamics.' which seems to be inaccesible.

These were compared in a study, the results of which are shown in Figure C.1. The present flat plate approximation comes between the other two and therefore is assumed to be adequate.

# APPENDIX D

## Flow Charts

MAIN PROGRAM

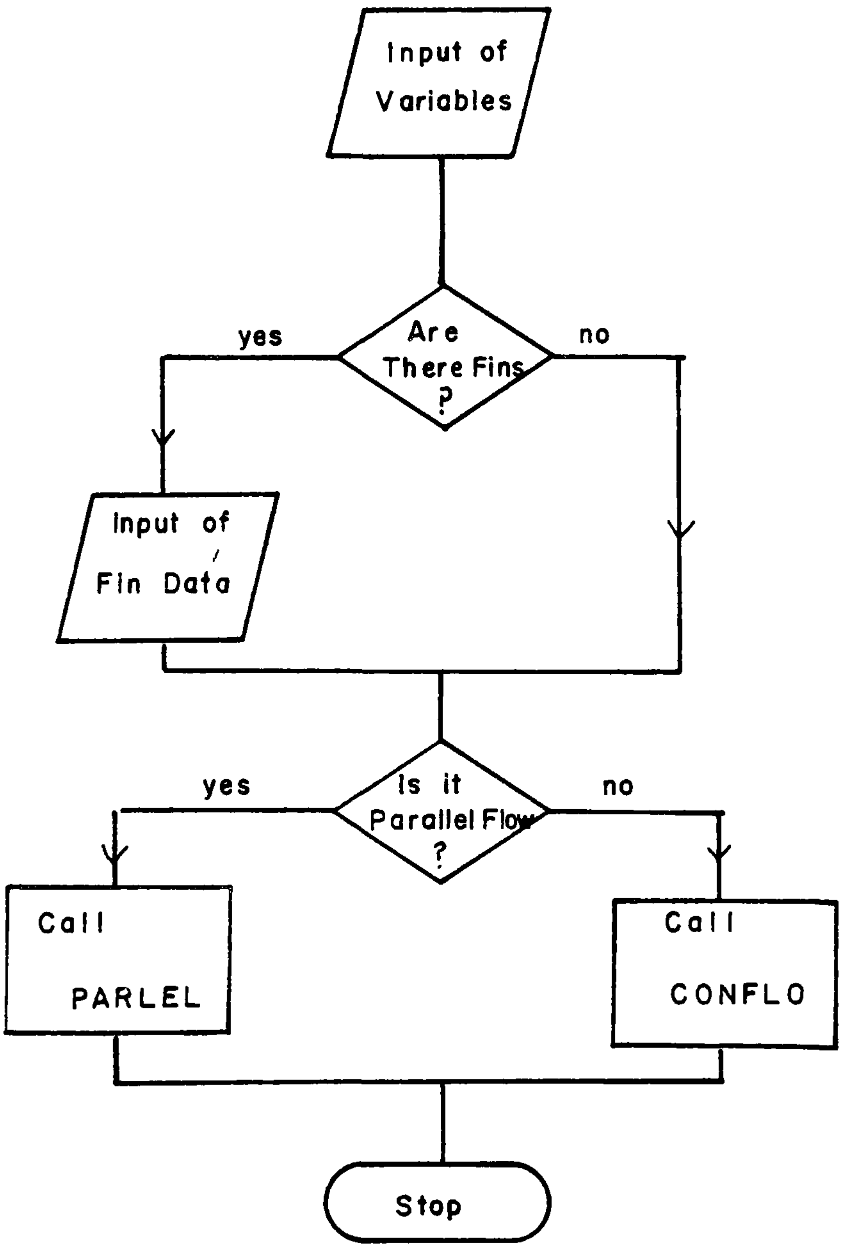
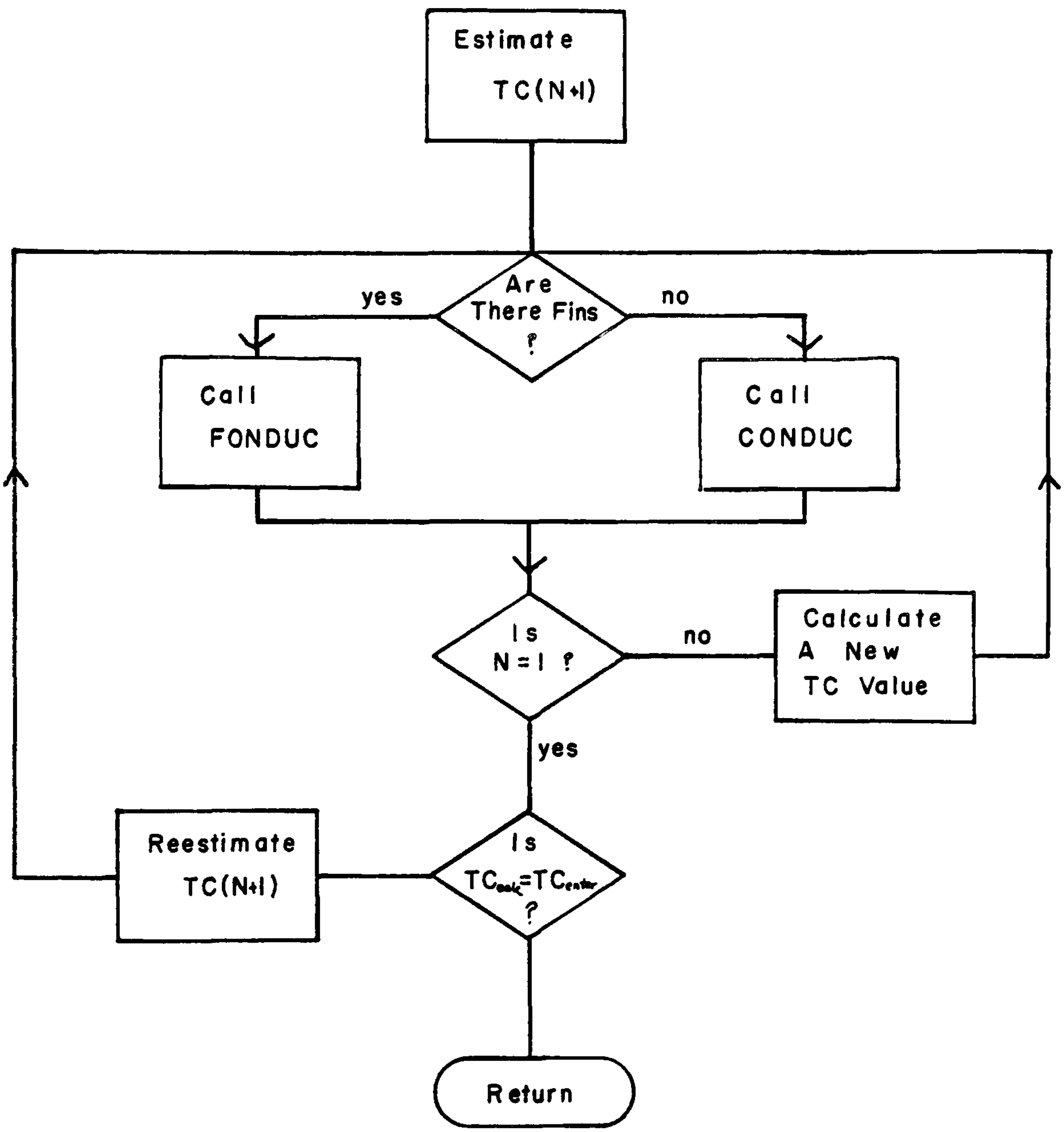



Fig.D.1




SUBROUTINE CONFLO

Fig.D.2

SUBROUTINE PARLEL

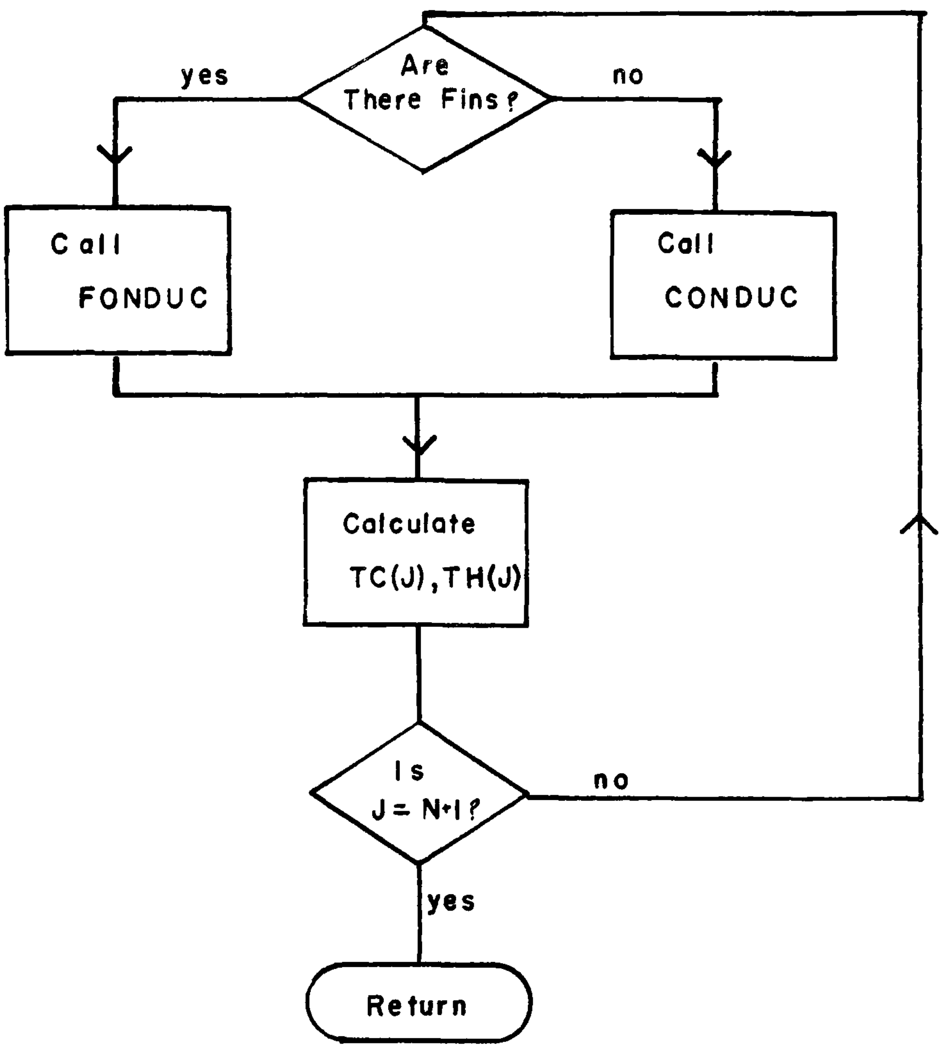
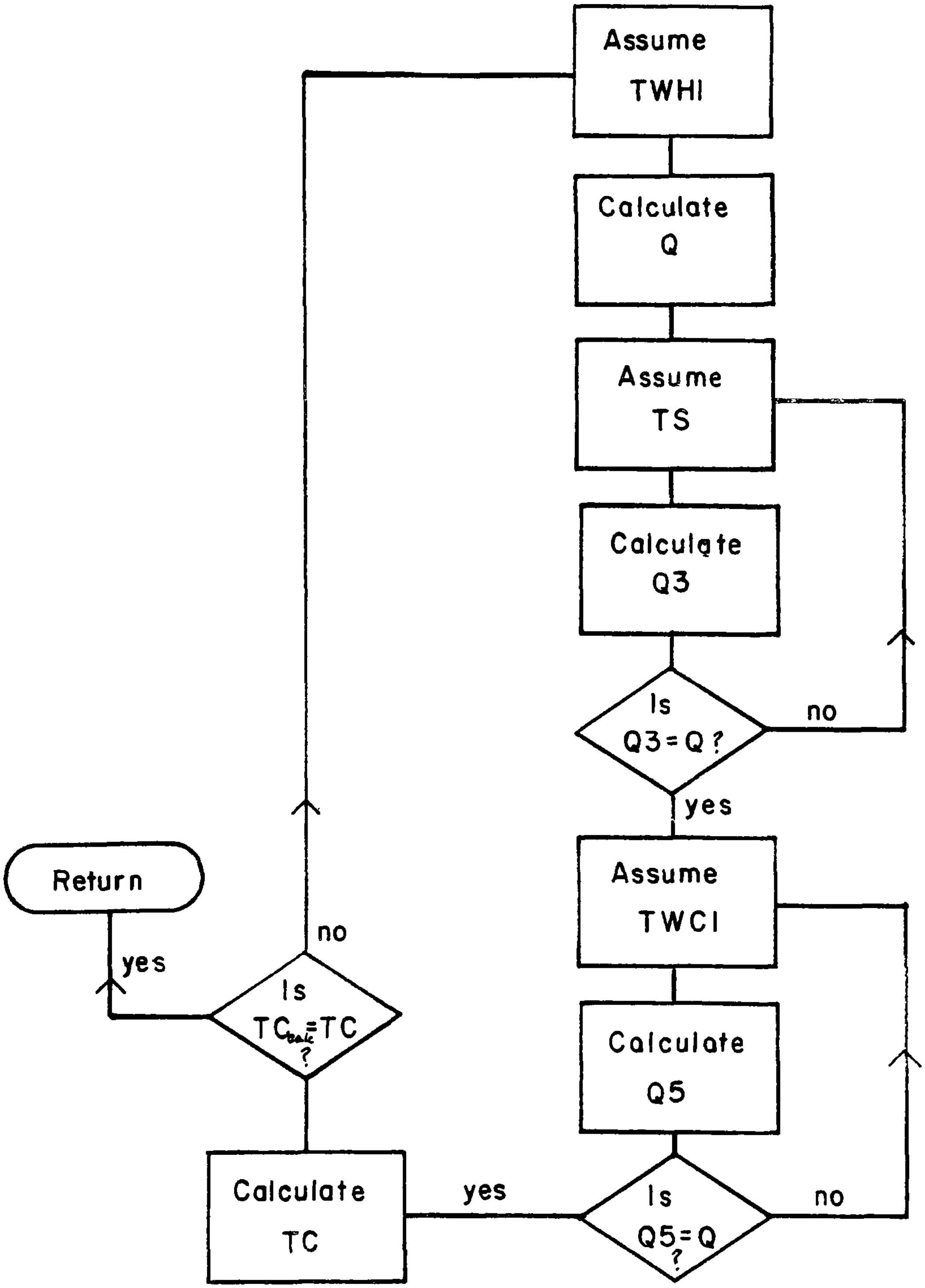



Fig.D.3




SUBROUTINE FONDUC

Fig.D.4

SUBROUTINE

CONDUC

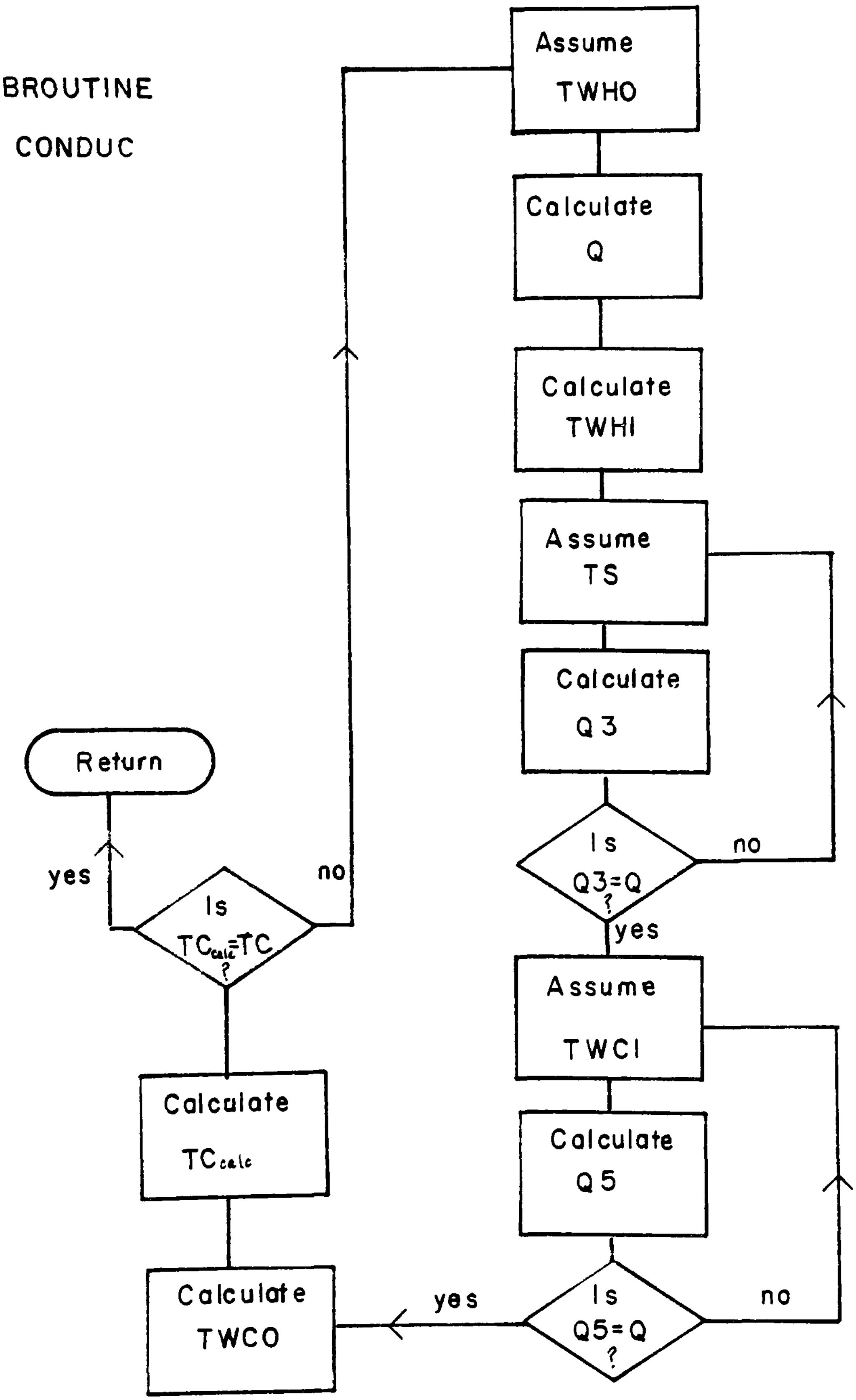



Fig.D.5

# APPENDIX E

## Program used for Analysis

```
//                 JOB          (SH169145,
//   ),'CLEMENTS',MSGLEVEL=(2,0),
//   CLASS=B
//   EXEC WATFIV,TIME=(5,0),REGION=200K
//WATFIV.SYSIN  DD  *
$JOB    SHXXXXXX,PAGES=60,TIME=300
C        PROG1 USES AIR AS THE WORKING FLUID
         REAL K,IDIA
         REAL HT
         REAL LC,LH
         INTEGER FINS
         INTEGER THEORY
         REAL LPLUS
         REAL MC,MH
         REAL LMTD
         DOUBLE PRECISION DELTTS(80)
         DOUBLE PRECISION PERC1(70)
         REAL TS(170),PS(170)
         REAL THOT(20),TCOLD(20),HEAT(20)
         REAL TLC7(25),STAGC7(25)
         REAL THOTE(10,10),TCOLDE(10,10),HEATF(10,10)
         INTEGER TYPE
         REAL ODIA
         JOFBLO=1
         READ(5,301)TYPE
    301  FORMAT(I1)
         READ(5,333)INL
    333  FORMAT(I1)
         READ(5,100)TFC,TFH,REC
    100  FORMAT(F6.2,4X,F6.2,4X,F8.2)
         READ(5,101)M
    101  FORMAT(I2)
         READ(5,103)LPLUS
    103  FORMAT(F4.2)
         READ(5,104)REPLUS
    104  FORMAT(F4.2)
         READ(5,107)WIDTH
    107  FORMAT(F4.1)
         READ(5,108)TALL
    108  FORMAT(F4.1)
         READ(5,110)K
    110  FORMAT(F5.1)
         READ(5,111)IDIA
         READ(5,111)ODIA
    111  FORMAT(F4.3)
         READ(5,112)SL
         READ(5,112)STF
    112  FORMAT(F3.1)
         READ(5,109)NTUBES
    109  FORMAT(I1)
         REH=REPLUS*REC
         READ(5,1104)FINS
   1104  FORMAT(I1)
         IF(FINS.EQ.0)GOTO1204
         READ(5,1205) THEORY
   1205  FORMAT(I1)
         READ(5,1101)HT
   1101  FORMAT(F7.4)
         READ(5,1102)EDIA
   1102  FORMAT(F5.3)
         READ(5,1103)FINNUM
   1103  FORMAT(F3.0)
   1204  CONTINUE
         WRITE(6,701)
    701  FORMAT('-','THE TYPE HEAT EXCHANGER IS EITHER COUNTERFLOW(1),OR PA
        *RALLEL FLOW(2)')
         WRITE(6,702)TYPE
    702  FORMAT('-',4X,I1)
         WRITE(6,803)THEORY
    803  FORMAT('-','THE THEORY BEING USED IS NUMBER',I1)
         WRITE(6,703)
    703  FORMAT('-','THE PIPES CAN BE ARRANGED EITHER INLINE(1),OR STAGGERE
        7D(0)')
         WRITE(6,704)INL
    704  FORMAT('-',4X,I1)
         WRITE(6,705)TFH
    705  FORMAT('-','THE TEMPERATURE OF THE HOT FLOW IS ',F6.2)
         WRITE(6,706)TFC
    706  FORMAT('-','THE TEMPERATURE OF THE COLD FLOW IS ',F6.2)
```

```
      WRITE(6,709)N
  709 FORMAT('-','THE NUMBER OF ROWS IS:',I2)
      WRITE(6,710)LPLUS
  710 FORMAT('-','THE RATIO OF THE LENGTHS IS:',F4.2)
      WRITE(6,711)REPLUS
  711 FORMAT('-','THE RATIO OF THE REYNOLDS NUMBERS ARE:',F4.2)
      WRITE(6,712)REC,REH
  712 FORMAT('-','THIS MEANS THE REYNOLDS NUMBER FOR THE COLD SIDE IS:',
     *F8.2,'AND THE HOT SIDE IS:',F8.2)
      WRITE(6,713)
  713 FORMAT('-','THE TUBES HAVE THE FOLLOWING CHARACTERISTICS')
      WRITE(6,714)ODIA,IDIA,K
  714 FORMAT('-',/,' ','OUTSIDE DIAMETER',F5.3,/,' ','INSIDE DIAMETER',F
     ?5.3,/,' ','CONDUCTIVITY',F5.1)
      WRITE(6,715)
  715 FORMAT('-','THE HEAT EXCHANGER HAS THE FOLLOWING CHARACTERISTICS')
      WRITE(6,716)TALL,WIDTH,STE,SL
  716 FORMAT('-','HEIGHT',F4.1,/,' ','WIDTH',F4.1,/,' ','TRANSVERSE SPAC
     ?ING',F3.1,/,' ','LONGITUDINAL SPACING',F3.1)
      ODIA=ODIA/12.0
      IDIA=IDIA/12.0
      WIDTH=WIDTH/12.0
      TALL=TALL/12.0
      SL=SL/12.0
      STE=STE/12.0
      LC=TALL/(1.0+LPLUS)
      LH=TALL-LC
      CALL FLUID(TEC,AV,AK,ACP,AD)
      MC=(REC*AV*LC*WIDTH)/(ODIA)
      CALL FLUID(TEH,AV,AK,ACP,AD)
      MH=(REH*AV*LH*WIDTH)/(ODIA)
      WRITE(6,707)MC
  707 FORMAT('-','THE MASS FLOW OF COLD AIR IS',F6.2)
      WRITE(6,708)MH
  708 FORMAT('-','THE MASS FLOW OF HOT AIR IS',F6.2)
      IF(FINS.EQ.0)GOTO1105
      FINNC=FINNUM*(LC/(LC+LH))
      FINNH=FINNUM*(LH/(LC+LH))
      WRITE(6,1106)
 1106 FORMAT('-','THE  EXCHANGER USES TUBES WITH RADIAL FINS')
      WRITE(6,1107)HT
 1107 FORMAT('-','THE THICKNESS OF EACH FIN IS',F7.4)
      WRITE(6,1108) EDIA
 1108 FORMAT('-','THE EXTREME DIAMETER OF THE FIN IS',F5.3,'MEASURED IN
     ?INCHES')
      WRITE(6,1109)FINNUM
 1109 FORMAT('-','THE NUMBER OF FINS ARE',F3.0)
      WRITE(6,1110)FINNC,FINNH
 1110 FORMAT('-','THE NUMBER OF FINS ON THE COLD SIDE IS',F5.2,'AND THE
     ?NUMBER OF FINS ON THE HOT SIDE IS',F5.2)
      HT=HT/12.0
      SPACE=(TALL-HT*FINNUM)/(FINNUM+1.0)
      WRITE(6,1111)SPACE
 1111 FORMAT('-','THE SPACING IS',F8.6)
      EDIA=EDIA/12.0
 1105 CONTINUE
      IF(TYPE.EQ.1)GOTO302
      CALL PARLEL(TCOLD,THOT,HEAT,TEC,TEH,MC,MH,N,AT,QT,J,Q,ACP,LMTD,UT,
     *TS,TIC,TIH,CZ,CZ1,CZ7,INL,LC,LH,FINNUM,FINNC,FINNH,SPACE,HT,EDIA,F
     ?INS,NTUBES,WIDTH,SL,STE,ODIA,IDIA,K,THEORY,TALL)
      GOTO 303
  302 CALL CONFLO(TCOLDE,THOTE,HEATE,TC,TH,MC,MH,N,AT,QT,KAY,L,TEH,TEC,Q
     *,J,ACP,TCDELT,TCABS,LMTD,UT,TS,CZ,CZ1,CZ7,40,INL,JOERLO,LC,LH,FINN
     ?UM,FINNC,FINNH,SPACE,HT,EDIA,FINS,NTUBES,WIDTH,SL,STE,ODIA,IDIA,K,
     1THEORY,TALL)
  303 STOP
      END
      SUBROUTINE PARLEL(TCOLD,THOT,HEAT,TEC,TEH,MC,MH,N,AT,QT,J,Q,ACP,L
     ?MTD,UT,TS,TIC,TIH,CZ,CZ1,CZ7,INL,LC,LH,FINNUM,FINNC,FINNH,SPACE,HT
     ?,EDIA,FINS,NTUBES,WIDTH,SL,STE,ODIA,IDIA,K,THEORY,TALL)
      DOUBLE PRECISION DELTTS(80)
      REAL K
      REAL LMTD
      INTEGER FINS
      REAL M,PERC5,PERC3,PR,H1,H3,H,H5
      REAL HT
      INTEGER THEORY
      DOUBLE PRECISION PERC1(70)
      REAL MC,MH,PI,IDIA,LC,LH,IRAD
      REAL ILC7(25),STAGCZ(25)
      REAL TS(170),PS(170)
      REAL THOT(20),TCOLD(20),HEAT(20)
```

```
      QT=0.0
      ATC=0.0
      ATH=0.0
      J=1
      TCOLD(J)=TEC
      THOT(J)=TEH
  104 TIC=TCOLD(J)
      TIH=THOT(J)
      IF(INL.EQ.0)GOTO701
      CALL INLINE(J,INCZ,CZ)
      NTHERM=NTUBES
      GOTO702
  701 CALL STAGER(J,STAGCZ,CZ)
      NTHERM=NTUBES-(J+1-((J+1)/2)*2)
  702 CZ1=CZ
      CZZ=CZ
      IF(FINS.EQ.0)GOTO1112
      CALL FONDUC(MC,MH,TIC,TIH,I,PI,LC,LH,Q,JOFHLO,INL,HT,EDIA,FINNC,FI
     ?NNH,FINNUM,SPACE,CZ1,CZZ,J,NTHERM,WIDTH,SL,STE,ODIA,IDIA,K,AREABH,
     ?AREAFH,AREAHC,AREAFC,THEORY,TALL)
      ATC=(AREAFC+AREABC)*NTHERM+ATC
      ATH=(AREAFH+AREABH)*NTHERM+ATH
      GOTO1113
 1112 CONTINUE
      CALL CONDUC(MC,MH,TIC,TIH,I,PI,CZ1,CZZ,AHO,ACO,K,ODIA,IDIA,LC,R6,L
     *H,R2,PERC1,DELTT0,P1,R7,AHI,ACI,DELTT1,TWHO,Q,TWHI,DELTT2,PERC3,DE
     *LTT3,TS,P3,Q3,PERC5,DELTT5,TWCI,R5,Q5,TWCO,TCCALC,DELTTI,AV,AK,ACP
     *,B,H,Z,PR,VM,H1,AD,H3,WK,WD,H,WV,T,WVD,ST,BETA,Y,HS,HZ,IRAD,J,JOFB
     ?LO,INL,NTHERM,WIDTH,SL,STE)
      ATC=ACO*NTHERM+ATC
      ATH=AHO*NTHERM+ATH
 1113 HEAT(J)=Q
      QT=QT+HEAT(J)
      J=J+1
      CALL FLUID(TIC,AV,AK,ACP,AD)
      TCOLD(J)=TCOLD(J-1)+Q/(MC*ACP)
      CALL FLUID(TIH,AV,AK,ACP,AD)
      THOT(J)=THOT(J-1)-Q/(MH*ACP)
      LMTD=((THOT(1)-TCOLD(1))-(THOT(J)-TCOLD(J)))/(ALOG((THOT(1)-TCOLD(
     ?1))/(THOT(J)-TCOLD(J))))
      AT=ATC+ATH
      UTC=QT/(LMTD*ATC)
      UTH=QT/(LMTD*ATH)
      UT=QT/(LMTD*AT)
      WRITE(6,370)
  370 FORMAT('-','THE TOTAL HEAT TRANSFER IS')
      WRITE(6,371)QT
  371 FORMAT('-',7X,F11.4)
      WRITE(6,372)LMTD
  372 FORMAT('-','THE LOG-MEAN-TEMPERATURE DIFFERENCE IS',F8.4)
      WRITE(6,373)
  373 FORMAT('-','THE OVERALL HEAT TRANSFER COEFFICIENTS ARE')
      WRITE(6,374)UTH,UTC,UT
  374 FORMAT('-','HOT SIDE',F8.4,/,' ','COLD SIDE',F8.4,/,' ','TOTAL',F8
     ?.4)
      IF(J.EQ.N+1)GOTO103
      GOTO104
  103 CONTINUE
      WRITE(6,10)
   10 FORMAT('-','THE FINAL TEMPERATURES ARE:')
      WRITE(6,11)
   11 FORMAT('-',6X,'TC',6X,'TH')
      WRITE(6,12)TCOLD(J),THOT(J)
   12 FORMAT('-',4X,F6.2,4X,F6.2)
      WRITE(6,105)
  105 FORMAT('-','THE OVERALL HEAT TRANSFER COEFFICIENT AND THE TOTAL HE
     *AT TRANSFER ARE:')
      WRITE(6,106)
  106 FORMAT(3X,'UT',8X,'QT')
      WRITE(6,107)UT,QT
  107 FORMAT('-',F8.4,2X,F11.4)
      WRITE(6,620)
  620 FORMAT('-','THE VALUES OF THE LOG MEAN TEMPERATURE DIFFERENCE IS:'
     *)
      WRITE(6,621)
  621 FORMAT('-',3X,'LMTD')
      WRITE(6,623)LMTD
  623 FORMAT('-',1X,F16.8)
      RETURN
      END
      SUBROUTINE CONFLO(TCOLDE,THOTE,HEATE,TIC,TIH,MC,MH,N,AT,QT,KAY,L,T
     *EH,TEC,Q,I,ACP,TCDELT,TCABS,LMTD,UT,TS,CZ,CZ1,CZZ,MO,INL,JOFHLO,LC
     ?,LH,FINNUM,FINNC,FINNH,SPACE,HT,EDIA,FINS,NTUBES,WIDTH,SL,STE,ODIA
     ?,IDIA,K,THEORY,TALL)
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      INTEGER THEORY
      REAL LMTD
      REAL HT
      INTEGER FINS
      REAL K
      DOUBLE PRECISION DELTTS(80)
      DOUBLE PRECISION PERC1(70)
      REAL ILCZ(25),STAGCZ(25)
      REAL MC,MH,PI,IDIA,LC,LH,IRAD
      REAL M,PERC5,PERC3,PR,H1,H3,H,H5
      REAL PS(170),TS(170)
      REAL THOTE(10,10),TCOLDE(10,10),HEATE(10,10)
      QT=0.0
      ATH=0.0
      ATC=0.0
      I=1
      KAY=N+1
      J=1
      TCOLDE(KAY,L)=TEC+1.0*FINNUM
  207 THOTE(J,L)=TEH
  203 TIH=THOTE(J,L)
      TIC=TCOLDE(KAY,L)
      IF(TIC.LT.TEC)GOTO510
      MO=KAY-1
      IF(INL.EQ.0)GOTO703
      CALL INLINE(MO,ILCZ,CZ)
      CZ7=CZ
      CALL INLINE(J,ILCZ,CZ)
      CZ1=CZ
      NTHERM=NTUBES
      GOTO704
  703 CALL STAGER(MO,STAGCZ,CZ)
      CZ7=CZ
      CALL STAGER(J,STAGCZ,CZ)
      CZ1=CZ
      NTHERM=NTUBES-(J+1-((J+1)/2)*2)
  704 CONTINUE
      IF(FINS.EQ.0)GOTO1114
      CALL FONDUC(MC,MH,TIC,TIH,I,PI,LC,LH,Q,JOEBLO,INL,HT,EDIA,FINNC,FI
     ?NNH,FINNUM,SPACE,CZ1,CZ7,J,NTHERM,WIDTH,SL,STE,ODIA,IDIA,K,AREABH,
     ?AREAFH,AREABC,AREAFC,THEORY,TALL)
      ATC=(AREAFC+AREABC)*NTHERM+ATC
      ATH=(AREAFH+AREABH)*NTHERM+ATH
      GOTO1115
 1114 CONTINUE
      CALL CONDUC(MC,MH,TIC,TIH,I,PI,CZ1,CZ7,AHO,ACO,K,ODIA,IDIA,LC,R6,L
     *H,R2,PERC1,DELTTO,R1,R7,AHI,ACI,DELTT1,TWHO,Q,TWHI,DELTT2,PERC3,DE
     *LTT3,TS,P3,Q3,PERC5,DELTT5,TWCI,R5,Q5,TWCO,TCCALC,DELTTI,AV,AK,ACP
     *,B,M,Z,PR,VM,H1,AD,H3,WK,WO,H,WV,T,WVD,ST,BETA,Y,H5,H7,IRAD,J,JOEB
     ?LO,INL,NTHERM,WIDTH,SL,STE)
      ATH=AHO*NTHERM+ATH
      ATC=ACO*NTHERM+ATC
 1115 CONTINUE
      IF(JOEBLO.EQ.2)GOTO510
      HEATE(J,L)=Q
      QT=QT+HEATE(J,L)
      J=J+1
      CALL FLUID(TIH,AV,AK,ACP,AD)
      THOTE(J,L)=THOTE(J-1,L)-Q/(MH*ACP)
      CALL FLUID(TIC,AV,AK,ACP,AD)
      KAY=KAY-1
      TCOLDE(KAY,L)=TCOLDE(KAY+1,L)-Q/(MC*ACP)
      LMTD=((THOTE(J,L)-TCOLDE(KAY,L))-(THOTE(1,L)-TCOLDE(N+1,L)))/(ALOG
     ?((THOTE(J,L)-TCOLDE(KAY,L))/(THOTE(1,L)-TCOLDE(N+1,L))))
      AT=ATC+ATH
      UTC=QT/(LMTD*ATC)
      UTH=QT/(LMTD*ATH)
      UT=QT/(LMTD*AT)
      WRITE(6,370)
  370 FORMAT('-','THE TOTAL HEAT TRANSFER IS')
      WRITE(6,371)QT
  371 FORMAT('-',7X,F7.2)
      WRITE(6,372)LMTD
  372 FORMAT('-','THE LOG-MEAN-TEMPERATURE DIFFERENCE IS',F8.4)
      WRITE(6,373)
  373 FORMAT('-','THE OVERALL HEAT TRANSFER COEFFICIENTS ARE')
      WRITE(6,374)UTH,UTC,UT
  374 FORMAT('-','HOT SIDE',F8.4,/,' ','COLD SIDE',F8.4,/,' ','TOTAL',F8
     ?.4)
      IF(J.NE.N+1)GOTO203
      TCDELT=TCOLDE(1,L)-TEC
      TCABS=ABS(TCDELT)
      IF(TCABS.LT.0.1)GOTO204
  205 I=I+1
      KAY=N+1
      JOEBLO=1
      QT=0.0
      ATC=0.0
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      ATH=0.0
      J=1
      TCOLDE(KAY,L)=TCOLDE(N+1,L-1)-((TCOLDE(1,L-1)-TFC)/1.4)
      GOTO207
  510 L=L+1
      JOEBLO=1
      ATH=0.0
      ATC=0.0
      KAY=N+1
      QT=0.0
      J=1
      TCOLDE(KAY,L)=1.01*TCOLDE(KAY,L-1)
      GOTO207
  204 IF(ABS(THOTE(1,L)-TCOLDE(N,L)).LT.0.0005)GOTO205
      LMTD=((THOTE(N+1,L)-TCOLDE(1,L))-(THOTE(1,L)-TCOLDE(N+1,L)))/(ALOG
     *((THOTE(N+1,L)-TCOLDE(1,L))/(THOTE(1,L)-TCOLDE(N+1,L))))
      AT=ATC+ATH
      UT=QT/(LMTD*AT)
      WRITE(6,887)
  887 FORMAT('-','THE FINAL TEMPERATURES ARE:')
      WRITE(6,888)
  888 FORMAT('-',6X,'TC',6X,'TH')
      WRITE(6,889)TCOLDE(KAY,L),THOTE(J,L)
  889 FORMAT('-',4X,F6.2,4X,F6.2)
      WRITE(6,213)
  213 FORMAT('-','THE OVERALL HEAT TRANSFER COEFFICIENT AND THE TOTAL HE
     *AT TRANSFER ARE:')
      WRITE(6,208)
  208 FORMAT(3X,'UT',8X,'QT')
      WRITE(6,209)UT,QT
  209 FORMAT('-',F8.4,2X,F12.4)
      WRITE(6,920)
  920 FORMAT('-','THE VALUE OF THE LOG MEAN TEMPERATURE DIFFERENCE IS:'
     *)
      WRITE(6,921)
  921 FORMAT('-',3X,'LMTD')
      WRITE(6,923)LMTD
  923 FORMAT('-',1X,F16.8)
      RETURN
      END
      SUBROUTINE CONDUC(MC,MH,TC,TH,I,PI,CZ1,CZ7,AHO,ACO,K,ODIA,IDIA,LC,
     *R6,LH,R2,PERC1,DELTT0,R1,R7,AHI,ACI,DELTT1,TWHO,Q,TWHI,DELTT2,PERC
     *3,DELTT3,TS,R3,Q3,PERC5,DELTT5,TWCI,R5,Q5,TWCO,TCCALC,DELTTI,AV,AK
     *,ACP,B,M,Z,PR,VM,H1,AD,H3,WK,WO,H,WV,T,WVD,ST,BETA,Y,H5,H7,IRAD,J,
     ?JOEBLO,INL,NTHERM,WIDTH,SL,STE)
      DOUBLE PRECISION DELTT5(80),BULL,TORO
      REAL H3,H,IRAD,IDIA,PI,H5,M,N,PR,H7,H1,MH,MC,ODIA,LH,LC
      REAL TS(170),PS(170)
      DOUBLE PRECISION FURD,FAB,PERC1(70)
      REAL K
      I=0
      PI=3.14159
      BULL=0.0
      TORO=0.0
      ERROR=0.1
      AHO=PI*ODIA*LH*NTHERM
      ACO=PI*ODIA*LC*NTHERM
      R6=((1.0/(2.0*PI*K*LC))*ALOG(ODIA/IDIA))/NTHERM
      R2=((1.0/(2.0*PI*K*LH))*ALOG(ODIA/IDIA))/NTHERM
      PERC1(I+1)=0.35
      ZING=PERC1(I+1)
      GOTO918
  917 I=0
      BULL=0.0
      TORO=0.0
      PERC1(I+1)=ZING*0.8
      ZING=PERC1(I+1)
  918 DELTT0=TH-TC
      CALL FINDH1(TH,AV,AK,ACP,B,M,Z,PR,VM,CZ1,H1,ODIA,MH,INL,LH,NTHERM,
     ?WIDTH,SL,STF)
      CALL FINDH7(TC,AV,AK,ACP,AD,B,M,Z,PR,VM,CZ7,H7,ODIA,MC,INL,LC,NTHE
     ?RM,WIDTH,SL,STF)
      R1=(1.0/(H1*AHO))
      R7=(1.0/(H7*ACO))
      AHI=IDIA*PI*LH*NTHERM
      ACI=IDIA*PI*LC*NTHERM
      TEST=100.0
      IRUN=1
      ICOUNT=0
      JRON=1
  904 I=I+1
      DELTT1=PERC1(I)*(TH-TC)
      IF(I.EQ.1)GOTO912
      TEST=TC-TCCALC
  912 TWHO=TH-DELTT1
      Q=DELTT1/R1
      TWHI=TWHO-Q*R2
      DELTT2=TWHO-TWHI
      PERC3=0.1
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      DELTT3=PERC3*(TH-TC)
  901 TS(I)=TWHI-DELTT3
      CALL FINDH3(H3,WK,WD,H,WV,T,WVD,TB,PB,BI,TWHI,I)
      R3=(1.0/(H3*AHI))
      Q3=DELTT3/R3
      IF(ABS(Q-Q3).LT.0.1)GOTO902
      DELTT3=ABS((((Q-Q3)/(Q*2.0))+1.0)*DELTT3)
      GOTO901
  902 PERC5=0.1
      ID=1
      JCOUNT=1
      DELTT5(ID)=PERC5*(TH-TC)
  906 TWCI=TS(I)-DELTT5(ID)
      IF(TWCI.LT.TC)GOTO917
      ID=ID+1
      CALL FINDH5(IRAD,IDIA,T,TS,I,TWCI,WK,WD,H,WV,WVD,PS,BETA,LC,PI,Y,H
     *5)
      R5=(1.0/(H5*ACI))
      Q5=DELTT5(ID-1)/R5
      IF(Q.LT.Q5)GOTO843
      BULL=DELTT5(ID-1)
      GOTO842
  843 TORO=DELTT5(ID-1)
  842 IF(JCOUNT.EQ.2)GOTO844
      IF(ID.EQ.2)GOTO841
      IF(QTEST*(Q-Q5).GT.0.0)GOTO841
      JCOUNT=2
  844 IF(ABS(Q-Q5).LT.6.0)GOTO905
      DELTT5(ID)=(BULL+TORO)/2.0
      GOTO906
  841 IF(ABS(Q-Q5).LT.6.0)GOTO905
      DELTT5(ID)=DABS((((Q-Q5)/(Q*3.0))+1.0)*DELTT5(ID-1))
      QTEST=Q-Q5
      GOTO906
  905 TWCO=TWCI-Q*R6
      TCCALC=TWCO-Q*R7
      DELTTI=TH-TCCALC
      IF(TCCALC.LT.TC)GOTO915
      FURD=PERC1(I)
      GOTO916
  915 FAR=PERC1(I)
  916 IF(JRON.EQ.1)GOTO914
      IF(ABS(TC-TCCALC).LT.ERROR)GOTO903
      PERC1(I+1)=(FURD+FAR)/2.0
      ERROR=ERROR*1.07
      GOTO904
  914 IF(I.EQ.1)GOTO913
      IF(ABS(TC-TCCALC).LT.ERROR)GOTO903
      IF(TEST*(TC-TCCALC).LT.0)GOTO911
  913 PERC1(I+1)=(((DELTTO-DELTTI)/(DELTTO*2.0))+1.0)*PERC1(I)
      GOTO904
  911 PERC1(I+1)=(PERC1(I)+PERC1(I-1))/2.0
      IRON=2
      GOTO904
  903 WRITE(6,711)
  711 FORMAT('1','THE FOLLOWING VALUES WERE CALCULATED FOR THE INTERMEDI
     *ATE TEMPERATURES')
      WRITE(6,712)J
  712 FORMAT('-','THE ROW IS',I2)
      WRITE(6,713)
  713 FORMAT('-',6X,'TC',7X,'TWCO',6X,'TWCI',7X,'TS',7X,'TWHI',6X,'TWHO'
     *,7X,'TH')
      WRITE(6,714)TC,TWCO,TWCI,TS(I),TWHI,TWHO,TH
  714 FORMAT('-',4X,F6.2,4X,F6.2,4X,F6.2,4X,F6.2,4X,F6.2,4X,F6.2,4X,F6.2
     *)
      WRITE(6,716)
  716 FORMAT('-','THE VALUES OF THE RESISTANCES ARE:')
      WRITE(6,717)
  717 FORMAT('-',8X,'R1',9X,'R2',9X,'R3',7X,'R5',8X,'R6',7X,'R7')
      WRITE(6,718)R1,R2,R3,R5,R6,R7
  718 FORMAT('-',6X,F9.6,1X,F9.6,1X,F9.6,1X,F9.6,1X,F9.6,1X,F9.6)
      WRITE(6,350)
  350 FORMAT('-','THE HEAT IN THIS STEP IS')
      WRITE(6,351)
  351 FORMAT('-','    Q')
      WRITE(6,352)Q
  352 FORMAT('-',F7.3)
      GOTO1002
 1002 RETURN
      END
      SUBROUTINE FONDUC(MC,MH,TC,TH,I,PI,LC,LH,Q,JOFALO,TNL,HT,EDIA,FINN
     2C,FINNH,FINNUM,SPACE,CZ1,CZ7,J,NTHERM,WIDTH,SL,STE,ODIA,IDIA,K,ARE
     2AH,AREAFH,AREAHC,AREAFC,THEORY,TAIL)
      INTEGER THEORY
      REAL H12,H3,H5,H67,H,IRAD,IDIA,PI,M,N,PR,MH,MC,ODIA,LC,LH,HOI12,HO
     2I67
      REAL H12F,H12R,H67F,H67R
      REAL K
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      REAL TS(170),PS(170)
      DOUBLE PRECISION FURD,FAR,PERC1(100)
      DOUBLE PRECISION DELTTS(80),BULL,TORO,Y,BETA
      I=0
      PI=3.14159
      BULL=0.0
      TORO=0.0
      ERROR=0.1
      PERC1(I+1)=0.35
      ZING=PERC1(I+1)
      GOTO918
  917 I=0
      BULL=0.0
      TORO=0.0
      PERC1(I+1)=ZING*0.8
      ZING=PERC1(I+1)
  918 DELTTO=TH-TC
      IF(THEORY.EQ.1)GOTO731
      IF(THEORY.EQ.2)GOTO732
      IF(THEORY.EQ.3)GOTO733
      IF(THEORY.EQ.4)GOTO734
      IF(THEORY.EQ.5)GOTO735
      GOTO 736
  731 CALL EQLEN(EQLENG,PI,EDIA,ODIA,AREABC,AREABH,AREAFC,AREAFH,HT,FINN
     1H,FINNC,LH,LC,FINNUM)
      CALL THEOR1(TH,AV,ACP,AD,AK,H12F,ODIA,SPACE,HT,EDIA,MH,LH,FINNH,CZ
     11,EQLENG,NTHERM,WIDTH,K,PI,R12,R67,MC,LC,FINNUM,INL,SL,STE,IDIA,TC
     1,FINNC,C77,AREABC,AREABH,AREAFC,AREAFH)
      GOTO737
  732 CALL EQLEN(EQLENG,PI,EDIA,ODIA,AREABC,AREABH,AREAFC,AREAFH,HT,FINN
     1H,FINNC,LH,LC,FINNUM)
      CALL HYDDIA(HDIA,SL,STE,FINNUM,PI,ODIA,EDIA,HT,TALL,AREAFC,AREAFH,
     1AREABC,AREABH,FINNH,FINNC,LC,LH)
      CALL THEOR2(HT,EDIA,ODIA,HDIA,SPACE,REH,REC,R12,R67,TC,EQLENG,AREA
     1BC,AREABH,AREAFC,AREAFH,PI,LC,LH,NTHERM,FINNUM,TH,WIDTH,MC,MH,CZ1,
     2CZ7,IDIA,FINNC,FINNH)
      GOTO737
  733 CALL EQLEN(EQLENG,PI,EDIA,ODIA,AREABC,AREABH,AREAFC,AREAFH,HT,FINN
     1H,FINNC,LH,LC,FINNUM)
      CALL THEOR3(ODIA,EDIA,SPACE,HT,MH,MC,LH,LC,K,IDIA,WIDTH,NTHERM,EQL
     2ENG,CZ1,CZ7,AREAFH,AREABH,AREAFC,AREABC,PI,TC,TH,R12,R67,FINNC,FIN
     2NH)
      GOTO737
  734 CALL HYDDIA(HDIA,SL,STE,FINNUM,PI,ODIA,EDIA,HT,TALL,AREAFC,AREAFH,
     1AREABC,AREABH,FINNH,FINNC,LC,LH)
      CALL THEOR1(TH,AV,ACP,AD,AK,H12F,ODIA,SPACE,HT,EDIA,MH,LH,FINNH,CZ
     11,HDIA,NTHERM,WIDTH,K,PI,R12,R67,MC,LC,FINNUM,INL,SL,STE,IDIA,TC
     1,FINNC,C77,AREABC,AREABH,AREAFC,AREAFH)
      GOTO737
  735 CALL HYDDIA(HDIA,SL,STE,FINNUM,PI,ODIA,EDIA,HT,TALL,AREAFC,AREAFH,
     1AREABC,AREABH,FINNH,FINNC,LC,LH)
      CALL THEOR2(HT,EDIA,ODIA,HDIA,SPACE,REH,REC,R12,R67,TC,HDIA,AREA
     1BC,AREABH,AREAFC,AREAFH,PI,LC,LH,NTHERM,FINNUM,TH,WIDTH,MC,MH,CZ1,
     2CZ7,IDIA,FINNC,FINNH)
      GOTO737
  736 CALL HYDDIA(HDIA,SL,STE,FINNUM,PI,ODIA,EDIA,HT,TALL,AREAFC,AREAFH,
     1AREABC,AREABH,FINNH,FINNC,LC,LH)
      CALL THEOR3(ODIA,EDIA,SPACE,HT,MH,MC,LH,LC,K,IDIA,WIDTH,NTHERM,HDI
     2A,CZ1,C77,AREAFH,AREABH,AREAFC,AREABC,PI,TC,TH,R12,R67,FINNC,FINNH
     2)
  737 AHI=IDIA*PI*LH*NTHERM
      ACI=IDIA*PI*LC*NTHERM
      TEST=100.0
      IRUN=1
      ICOUNT=0
      JRON=1
  904 I=I+1
      DELTT1=PERC1(I)*(TH-TC)
      IF(I.EQ.1)GOTO912
      TEST=TC-TCCALC
  912 CONTINUE
      Q=DELTT1/R12
      TWHI=TH-Q*R12
      PERC3=0.18
      DELTT3=PERC3*(TH-TC)
  901 TS(I)=TWHI-DELTT3
      CALL FINDH3(H3,WK,WD,H,WV,T,WVD,TS,PS,ST,TWHI,I)
      R3=(1.0/(H3*AHI))
      Q3=DELTT3/R3
      IF(ABS(Q-Q3).LT.0.1)GOTO902
      DELTT3=ABS((((Q-Q3)/(Q*2.0))+1.0)*DELTT3)
      GOTO901
  902 CONTINUE
      PERC5=0.001
      ID=1
      JCOUNT=1
      DELTTS(ID)=PERC5*(TH-TC)
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  906 TWCI=TS(I)-DELTTS(ID)
      IF(TWCI.LT.TC)GOTO917
      ID=ID+1
      CALL FINDHS(IRAD,IDIA,I,IS,1,TWCI,MR,MD,M,MV,MVD,FS,BETA,LW,FI,T,H
     75)
      R5=(1.0/(H5*ACI))
      Q5=DELTTS(ID-1)/R5
      IF(Q.LT.Q5)GOTO843
      BULL=DELTTS(ID-1)
      GOTO842
  843 TORD=DELTTS(ID-1)
  842 IF(JCOUNT.EQ.2)GOTO844
      IF(ID.EQ.2)GOTO841
      IF(QTEST*(Q-Q5).GT.0.0)GOTO841
      JCOUNT=2
  844 IF(ABS(Q-Q5).LT.10.0)GOTO905
      DELTTS(ID)=DABS((((Q-Q5)/(Q*3.0))+1.0)*DELTTS(ID-1))
      QTEST=Q-Q5
      GOTO906
  841 IF(ABS(Q-Q5).LT.10.0)GOTO905
      DELTTS(ID)=DABS((((Q-Q5)/(Q*3.0))+1.0)*DELTTS(ID-1))
      QTEST=Q-Q5
      GOTO906
  905 TCCALC=TWCI-Q*R67
      DELTTI=TH-TCCALC
      IF(TCCALC.LT.TC)GOTO915
      FURD=PERC1(I)
      GOTO916
  915 FAR=PERC1(I)
  916 IF(JRON.EQ.1)GOTO914
      IF(ABS(TC-TCCALC).LT.ERROR)GOTO903
      PERC1(I+1)=(FURD+FAR)/2.0
      ERROR=ERROR*1.07
      GOTO904
  914 IF(I.EQ.1)GOTO913
      IF(ABS(TC-TCCALC).LT.ERROR)GOTO903
      IF(TEST*(TC-TCCALC).LT.0)GOTO911
  913 PERC1(I+1)=(((DELTTO-DELTTI)/(DELTTO*2.0))+1.0)*PERC1(I)
      GOTO904
  911 PERC1(I+1)=(PERC1(I)+PERC1(I-1))/2.0
      JRON=2
      GOTO904
  903 WRITE(6,711)
  711 FORMAT('1','THE FOLLOWING VALUES WERE CALCULATED FOR THE INTERMEDI
     *ATE TEMPERATURES')
      WRITE(6,712)J
  712 FORMAT('-','THE ROW IS',I2)
      WRITE(6,713)
  713 FORMAT('-',6X,'TH',7X,'TWHI',7X,'TS',7X,'TWCI',7X,'TC')
      WRITE(6,714)TH,TWHI,TS(I),TWCI,TC
  714 FORMAT('-',4X,F6.2,4X,F6.2,4X,F6.2,4X,F6.2,4X,F6.2)
      WRITE(6,716)
  716 FORMAT('-','THE VALUES OF THE RESISTANCES ARE;')
      WRITE(6,717)
  717 FORMAT('-',7X,'R12',8X,'R3',7X,'R5',7X,'R67')
      WRITE(6,718)R12 ,R3,R5,R67
  718 FORMAT('-',6X,F9.6,1X,F9.6,1X,F9.6,1X,F9.6)
      WRITE(6,350)
  350 FORMAT('-','THE HEAT IN THIS STEP IS:')
      WRITE(6,351)
  351 FORMAT('-','    Q')
      WRITE(6,352)Q
  352 FORMAT('-',F8.3)
      GOTO1002
 1002 RETURN
      END
      SUBROUTINE THEOR3(ODIA,EDIA,SPACE,HT,MH,MC,LH,LC,K,IDIA,WIDTH,NTHE
     7RM,EQLENG,C71,C77,AREAFH,AREABH,AREAFC,AREABC,PI,TC,TH,R12,R67,FIN
     7NC,FINNH)
      REAL HT,LO,M3,MH,MC,LH,LC,K,IDIA,NU12,NU67
      LO=(2.0*ODIA/PI)*((EDIA-ODIA)/(SPACE+HT)+1)
      M3=12.9/(20.8+.5*((EDIA-ODIA)/(SPACE+HT)+1))
      A=0.5*(EDIA-ODIA)*LO/(ODIA*SPACE)
      EX=(A/302.0)
      CALL BESSEL(BIO,AI1,BK0,BK1,EX)
      CALL FLUID(TH,AV,AK,ACP,AD)
      VM=MH/(AD*(LH*(WIDTH-ODIA*NTHERM)-NTHERM*(EDIA-ODIA)*HT*FINNH))
      NU12=0.18*(1.0+93.0*AI1)*((AD*VM*EQLENG/AV)**M3)
      H12=C71*(AK/EQLENG)*NU12
      HOI12=(AREAFH+AREABH)*H12/(LH*PI*IDIA)
      CALL FLUID(TC,AV,AK,ACP,AD)
      VM=MC/(AD*(LC*(WIDTH-ODIA*NTHERM)-NTHERM*(EDIA-ODIA)*HT*FINNC))
      NU67=0.18*(1.0+93.0*AI1)*((AD*VM*EQLENG/AV)**M3)
      H67=C77*(AK/EQLENG)*NU67
      HOI67=(AREAFC+AREABC)*H67/(LC*PI*IDIA)
      R12=1.0/((HOI12*(LH*PI*IDIA))*NTHERM)
      R67=1.0/((HOI67*(LC*PI*IDIA))*NTHERM)
      RETURN
      END
```

```
      SUBROUTINE THEOR1(TH,AV,ACP,AD,AK,H12F,ODIA,SPACE,HT,EDIA,MH,LH,FI
     1NNH,CZ1,EQLENG,NTHERM,WIDTH,K,PI,R12,R67,MC,LC,FINNUM,INL,SL,STE,I
     1DIA,TC,FINNC,CZ7,AREABC,AREABH,AREAFC,AREAFH)
      REAL MH,LH,K,MC,LC,M,IDIA
      CALL FNH12F(TH,AV,ACP,AD,AK,H12F,ODIA,SPACE,HT,EDIA,MH,LH,FINNH,CZ
     21,EQLENG,NTHERM,WIDTH)
      EM=((2*H12F)/(K*HT))**0.5
      EX=EM*ODIA/2.0
      CALL BESSEL(CI0,CI1,CK0,CK1,EX)
      EX=EM*EDIA/2.0
      CALL BESSEL(DI0,DI1,DK0,DK1,EX)
      EFF=(4.0/(EM*ODIA*(1.0-(EDIA/ODIA)**2.)))*((CI1*DK1-DI1*CK1)/(CI0*
     2DK1+DI1*CK0))
      CALL FNH12B(TH,AV,AK,ACP,AD,B,M,Z,PR,VM,CZ1,H12B,ODIA,MH,INL,LH,ED
     2IA,FINNH,HT,NTHERM,WIDTH,SL,STE)
      H12=CZ1*(AREABH*H12B+EFF*AREAFH*H12F)/(AREABH+AREAFH)
      HOI12=(AREAFH+AREABH)*H12/(LH*PI*IDIA)
      R12=1.0/((HOI12*(LH*PI*IDIA))*NTHERM)
      CALL FNH67F(TC,AV,AK,ACP,AD,H67F,ODIA,SPACE,HT,EDIA,MC,LC,FINNC,CZ
     27,EQLENG,NTHERM,WIDTH)
      EM=((2.0*H67F)/(K*HT))**0.5
      EX=EM*ODIA/2.0
      CALL BESSEL(CI0,CI1,CK0,CK1,EX)
      EX=EM*EDIA/2.0
      CALL BESSEL(DI0,DI1,DK0,DK1,EX)
      EFF=(4.0/(EM*ODIA*(1.0-(EDIA/ODIA)**2.)))*((CI1*DK1-DI1*CK1)/(CI0*
     2DK1+DI1*CK0))
      CALL FNH67B(TC,AV,AK,ACP,AD,B,M,Z,PR,VM,CZ7,H67B,ODIA,MC,INL,LC,ED
     2IA,FINNC,HT,NTHERM,WIDTH,SL,STE)
      H67=CZ7*(AREABC*H67B+EFF*AREAFC*H67F)/(AREABC+AREAFC)
      HOI67=(AREAFC+AREABC)*H67/(LC*PI*IDIA)
      R67=1.0/((HOI67*(LC*PI*IDIA))*NTHERM)
      RETURN
      END
      SUBROUTINE HYDDIA(HDIA,SL,STE,FINNUM,PI,ODIA,EDIA,HT,TALL,AREAFC,A
     1REAFH,AREABC,AREABH,FINNH,FINNC,LC,LH )
      REAL HDIA,PI,HT,LC,LH
      AREAFH=((PI/2.0)*(EDIA**2.0-ODIA**2.0)+HT*EDIA*PI)*FINNH
      AREABH=(PI*ODIA*(LC+LH-FINNUM*HT))*(LH/(LH+LC))
      AREAFC=((PI/2.0)*(EDIA**2.0-ODIA**2.0)+HT*EDIA*PI)*FINNC
      AREABC=(PI*ODIA*(LC+LH-FINNUM*HT))*(LC/(LC+LH))
      HDIA=(SL*STE*TALL-(FINNUM*(PI/4.0)*(ODIA**2-EDIA**2)*HT)-(PI/4.0)
     1*(ODIA**2)*TALL)/(AREAFC+AREAFH+AREABC+AREABH)
      RETURN
      END
      SUBROUTINE EQLEN(EQLENG,PI,EDIA,ODIA,AREABC,AREABH,AREAFC,AREAFH,H
     1T,FINNH,FINNC,LH,LC,FINNUM)
      REAL PI,LH,LC
      AREAFH=((PI/2.0)*(EDIA**2.0-ODIA**2.0)+HT*EDIA*PI)*FINNH
      AREABH=(PI*ODIA*(LC+LH-FINNUM*HT))*(LH/(LH+LC))
      AREAFC=((PI/2.0)*(EDIA**2.0-ODIA**2.0)+HT*EDIA*PI)*FINNC
      AREABC=(PI*ODIA*(LC+LH-FINNUM*HT))*(LC/(LC+LH))
      EQLENG=PI*(EDIA**2-ODIA**2)/(8.0*(EDIA**2-ODIA**2))
      RETURN
      END
      SUBROUTINE THEOR2(HT,EDIA,ODIA,HDIA,SPACE,REH,REC,R12,R67,TC,EQLEN
     1G,AREABC,AREABH,AREAFC,AREAFH,PI,LC,LH,NTHERM,FINNUM,TH,WIDTH,MC,M
     1H,CZ1,CZ7,IDIA,FINNC,FINNH)
      REAL HT,HDIA,LC,LH,MC,MH,NU12,NU67,IDIA
      C1=0.159*((HT/(0.5*(EDIA-ODIA)))**0.141)*((HDIA/HT)**0.065)
      C2=-0.323*((HT/(0.5*(EDIA-ODIA)))**0.049)*((SPACE/HT)**0.077)
      CALL FLUID(TH,AV,AK,ACP,AD)
      VM=MH/(AD*(LH*(WIDTH-ODIA*NTHERM)-NTHERM*(EDIA-ODIA)*HT*FINNH))
      NU12=C1*((AD*VM*EQLENG/AV)**(C2+1))*((AV*ACP/AK)**0.333)
      H12=CZ1*(AK/EQLENG)*NU12
      HOI12=(AREAFH+AREABH)*H12/(LH*PI*IDIA)
      CALL FLUID(TC,AV,AK,ACP,AD)
      VM=MC/(AD*(LC*(WIDTH-ODIA*NTHERM)-NTHERM*(EDIA-ODIA)*HT*FINNC))
      NU67=C1*((AD*VM*EQLENG/AV)**(C2+1))*((AV*ACP/AK)**0.333)
      H67=CZ7*(AK/EQLENG)*NU67
      HOI67=(AREAFC+AREABC)*H67/(LC*PI*IDIA)
      R12=1.0/((HOI12*(LH*PI*IDIA))*NTHERM)
      R67=1.0/((HOI67*(LC*PI*IDIA))*NTHERM)
      RETURN
      END
      SUBROUTINE BESSEL(BI0,BI1,BK0,BK1,EX)
      IF(EX.GT.0.0.AND.EX.LE.0.25)GOTO1010
      IF(EX.GT.0.25.AND.EX.LE.0.5)GOTO1011
      IF(EX.GT.0.5.AND.EX.LE.0.76)GOTO1012
      IF(EX.GT.0.76.AND.EX.LE.1.0)GOTO1013
      IF(EX.GT.1.0.AND.EX.LE.1.26)GOTO1014
      IF(EX.GT.1.26.AND.EX.LE.1.5)GOTO1015
      IF(EX.GT.1.5.AND.EX.LE.1.76)GOTO1016
      IF(EX.GT.1.76.AND.EX.LE.2.0)GOTO1017
      IF(EX.GT.2.0.AND.EX.LE.2.26)GOTO1018
      IF(EX.GT.2.26.AND.EX.LE.2.5)GOTO1019
      IF(EX.GT.2.5.AND.EX.LE.2.76)GOTO1020
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      IF(EX.GT.2.76.AND.EX.LE.3.0)GOTO1021
      IF(EX.GT.3.0.AND.EX.LE.3.26)GOTO1022
      IF(EX.GT.3.26.AND.EX.LE.3.5)GOTO1023
      IF(EX.GT.3.5.AND.EX.LE.3.76)GOTO1024
      IF(EX.GT.3.76.AND.EX.LE.4.0)GOTO1025
      IF(EX.GT.4.0.AND.EX.LE.4.26)GOTO1026
      IF(EX.GT.4.26.AND.EX.LE.4.5)GOTO1027
      IF(EX.GT.4.5.AND.EX.LE.4.76)GOTO1028
      IF(EX.GT.4.76.AND.EX.LE.5.0)GOTO1029
      GOTO1030
1010  BI0=6.5226E-2*EX+0.9971
      BI1=0.5038*EX-0.0002
      BK0=-10.322*EX+3.7584
      BK1=-7.8007*EX+5.3998
      GOTO1031
1011  BI0=0.1937*EX+0.9643
      BI1=0.5038*EX-0.0002
      BK0=-2.3942*EX+2.0901
      BK1=-7.8007*EX+5.3998
      GOTO1031
1012  BI0=0.3371*EX+0.8909
      BI1=0.5797*EX-3.365E-2
      BK0=-1.2061*EX+1.5069
      BK1=-2.6542*EX+2.9143
      GOTO1031
1013  BI0=0.4911*EX+0.7725
      BI1=0.6576*EX-9.3847E-2
      BK0=-0.734*EX+1.1504
      BK1=-1.32821*EX+1.919
      GOTO1031
1014  BI0=0.6121*EX+0.6408
      BI1=0.7675*EX-0.2058
      BK0=-0.4804*EX+0.8957
      BK1=-0.7716*EX+1.3618
      GOTO1031
1015  BI0=0.8778*EX+0.3265
      BI1=0.9147*EX-0.3929
      BK0=-0.3268*EX+0.99999
      BK1=-0.4837*EX+0.99999
      GOTO1031
1016  BI0=1.1298*EX-5.5576E-2
      BI1=1.1077*EX-0.6858
      BK0=-0.2284*EX+0.5540
      BK1=-0.3189*EX+0.752
      GOTO1031
1017  BI0=1.436*EX-0.5979
      BI1=1.3558*EX-1.1252
      BK0=-0.1623*EX+0.4376
      BK1=-0.217*EX+0.5727
      GOTO1031
1018  BI0=1.81359*EX-1.359
      BI1=1.6733*EX-1.7657
      BK0=-0.1169*EX+0.3466
      BK1=-0.151*EX+0.44035
      GOTO1031
1019  BI0=2.2799*EX-2.4175
      BI1=2.076*EX-2.68
      BK0=-8.49791E-2*EX+0.2744
      BK1=-0.1068*EX+0.3404
      GOTO1031
1020  BI0=2.8607*EX-3.8795
      BI1=2.5869*EX-3.9662
      BK0=-6.2316E-2*EX+0.2176
      BK1=-7.6643E-2*EX+0.26476
      GOTO1031
1021  BI0=3.5643*EX-5.884
      BI1=3.2168*EX-5.7086
      BK0=-4.5916E-2*EX+0.1723
      BK1=-5.5448E-2*EX+0.2062
      GOTO1031
1022  BI0=4.4911*EX-8.62
      BI1=4.048*EX-8.2155
      BK0=-3.4058E-2*EX+0.1366
      BK1=-4.0437E-2*EX+0.16108
      GOTO1031
1023  BI0=5.6257*EX-12.3307
      BI1=5.0769*EX-11.5805
      BK0=-2.5346E-2*EX+0.1082
      BK1=-0.02975*EX+0.1262
      GOTO1031
1024  BI0=7.0526*EX-17.3493
      BI1=6.3786*EX-16.1588
      BK0=-1.8945E-2*EX+8.5748E-2
      BK1=-2.1978E-2*EX+9.8971E-2
      GOTO1031
1025  BI0=8.8427*EX-24.0989
      BI1=8.0195*EX-22.3459
      BK0=-1.4201E-2*EX+6.7905E-2
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      BK1=-1.6315E-2*EX+7.7671E-2
      GOTO1031
 1026 BI0=11.0992*EX-24.7019
      BI1=10.0962*EX-30.6887
      BK0=-1.0665E-2*EX+5.3731E-2
      BK1=-1.2157E-2*EX+6.1009E-2
      GOTO1031
 1027 BI0=13.9353*EX-45.2749
      BI1=12.7157*EX-41.8752
      BK0=-8.0349E-3*EX+0.04252
      BK1=-9.0752E-3*EX+4.7878E-2
      GOTO1031
 1028 BI0=17.516*EX-61.4498
      BI1=16.0332*EX-56.8614
      BK0=-6.0687E-3*EX+3.366E-2
      BK1=-6.6951E-3*EX+3.7127E-2
      GOTO1031
 1029 BI0=22.0231*EX-82.9511
      BI1=20.2207*EX-76.8379
      BK0=-4.5909E-3*EX+2.6626E-2
      BK1=-5.1154E-3*EX+2.9599E-2
      GOTO1031
 1030 WRITE(6,1032)
 1032 FORMAT('-','THE PROGRAM SCREWED UP BECAUSE THERE WAS NO BESSEL FUN
     ?CTION DESCRIBED FOR THIS POINT')
 1031 RETURN
      END
      SUBROUTINE FNH67B(TC,AV,AK,ACP,AD,B,M,Z,PR,VM,CZ7,H67B,ODIA,MC,INL
     ?,LC,EDIA,FINNC,HT,NTHERM,WIDTH,SL,STE)
      REAL LC
      REAL M,PR,H67B,MC
      CALL FLUID(TC,AV,AK,ACP,AD)
      VM=(MC/AD)/((LC*(WIDTH-ODIA*NTHERM))-(NTHERM*FINNC*(EDIA-ODIA)*HT)
     ?)
      IF(INL.EQ.0)GOTO350
      CALL INLBM(ATLAS,SL,ODIA,TIRE,STE,B,M)
      GOTO351
  350 CALL STAGBM(ODIA,SL,TAX,FORM,STE,M,B)
  351 H67B=(AK*B*(((AD*VM*ODIA)/(AV))**M)/ODIA)*((AV*ACP/AK)**0.33)
      RETURN
      END
      SUBROUTINE FNH67F(TC,AV,AK,ACP,AD,H67,ODIA,SPACE,HT,EDIA,MC,LC,FIN
     ?NC,CZ7,EQLENG,NTHERM,WIDTH)
      REAL LC,M,MC,HT,H67
      CALL FLUID(TC,AV,AK,ACP,AD)
      VM=(MC/AD)/((LC*(WIDTH-ODIA*NTHERM))-(NTHERM*FINNC*(EDIA-ODIA)*HT)
     ?)
      H67=CZ7*0.036*AD*ACP*VM*((AD*VM*EQLENG/AV)**(-0.2))/((AV*ACP/AK)**
     ?0.67)
      RETURN
      END
      SUBROUTINE FNH12F(TH,AV,ACP,AD,AK,H12,ODIA,SPACE,HT,EDIA,MH,LH,FIN
     ?NH,CZ1,EQLENG,NTHERM,WIDTH)
      REAL LH,M,H12,MH
      REAL HT
      CALL FLUID(TH,AV,AK,ACP,AD)
      VM=MH/(AD*(LH*(WIDTH-ODIA*NTHERM)-NTHERM*(EDIA-ODIA)*HT*FINNH))
      H12=CZ1*0.036*AD*ACP*VM*((AD*VM*EQLENG/AV)**(-0.2))/((AV*ACP/AK)**
     ?0.67)
      RETURN
      END
      SUBROUTINE FNH12B(TH,AV,AK,ACP,AD,B,M,Z,PR,VM,CZ1,H12B,ODIA,MH,INL
     ?,LH,EDIA,FINNH,HT,NTHERM,WIDTH,SL,STE)
      REAL LH
      REAL M,PR,H12B,MH
      CALL FLUID(TH,AV,AK,ACP,AD)
      VM=MH/(AD*(LH*(WIDTH-ODIA*NTHERM)-NTHERM*(EDIA-ODIA)*HT*FINNH))
      IF(INL.EQ.0)GOTO250
      CALL INLBM(ATLAS,SL,ODIA,TIRE,STE,B,M)
      GOTO251
  250 CALL STAGBM(ODIA,SL,TAX,FORM,STE,M,B)
  251 H12B=(AK*B*(((AD*VM*ODIA)/(AV))**M)/ODIA)*((AV*ACP/AK)**0.33)
      RETURN
      END
      SUBROUTINE FINDH1(TH,AV,AK,ACP,B,M,Z,PR,VM,CZ1,H1,ODIA,MH,INL,LH,N
     ?THERM,WIDTH,SL,STE)
      REAL LH
      REAL M,PR,H1,MH
      CALL FLUID(TH,AV,AK,ACP,AD)
      VM=MH/(AD*LH*(WIDTH-ODIA*NTHERM))
      IF(INL.EQ.0)GOTO950
      CALL INLBM(ATLAS,SL,ODIA,TIRE,STE,B,M)
      GOTO951
  950 CALL STAGBM(ODIA,SL,TAX,FORM,STE,M,B)
  951 H1=(AK*B*CZ1*(((AD*VM*ODIA)/(AV))**M)/ODIA)
      RETURN
      END
      SUBROUTINE FINDH7(TC,AV,AK,ACP,AD,B,M,Z,PR,VM,CZ7,H7,ODIA,MC,INL,L
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      2C,NTHERM,WIDTH,SL,STE)
       REAL LC
       REAL M,PR,H7,MC
      CALL FLUID(TC,AV,AK,ACP,AD)
      VM=MC/(AD*LC*(WIDTH-ODIA*NTHERM))
      IF(INL.EQ.0)GOTO650
      CALL INLRM(ATLAS,SL,ODIA,TIRE,STE,B,M)
      GOTO651
  650 CALL STAGRM(ODIA,SL,TAX,FORM,STE,M,B)
  651 H7=(AK*B*CZ7*(((AD*VM*ODIA)/(AV))**M)/ODIA)
      RETURN
      END
      SUBROUTINE FINDH5(IRAD,IDIA,T,TS,I,TWCI,WK,WD,H,WV,WVD,PS,BETA,LC,
     *PI,Y,H5)
      DOUBLE PRECISION Y,BETA
      REAL IRAD,IDIA,H,LC,PI,H5
      REAL TS(170),PS(170)
      T=(TS(I)+TWCI)/2.0
      CALL H2O(WK,WD,H,WV,T)
      CALL VAP(WVD,TS,PS,I)
      IRAD=IDIA/2.0
      BETA=(2.0*PI*WV*WK*(TS(I)-TWCI)*LC)/(WD*(WD-WVD)*H*(IRAD**4.0)*(60
     ?.0**2))
      CALL BISECT(BETA,Y)
      H5=(WK*(0.25-(Y**2.0)+0.75*(Y**4.0)-(Y**4.0)*(DLOG(Y))))/(BETA*IDI
     *A)
      RETURN
      END
      SUBROUTINE FINDH3(H3,WK,WD,H,WV,T,WVD,TS,PS,ST,TWHI,I)
      REAL H3,H
      REAL TS(170),PS(170)
      T=(TS(I)+TWHI)/2.0
      CALL H2O(WK,WD,H,WV,T)
      CALL SURTEN(ST,T)
      CALL VAP(WVD,TS,PS,I)
      H3=(((TWHI-TS(I))/(H*0.013*(((WV*3600.0*32.2)/WK)**1.7)))**3.03)*(
     *((WD-WVD)/ST)**0.5)*((32.2*3600.0*H*WV)/(TWHI-TS(I)))
      RETURN
      END
      SUBROUTINE FLUID(T,FV,FK,FCP,FD)
C     THIS SUBROUTINE USES AIR PROPERTIES
      IF(T.LE.50.0)GOTO81
      IF(T.GT.50.0.AND.T.LE.100.0)GOTO82
      IF(T.GT.100.0.AND.T.LE.150.0)GOTO83
      IF(T.GT.150.0.AND.T.LE.200.0)GOTO84
      IF(T.GT.200.0.AND.T.LE.300.0)GOTO85
      FV=(0.000515*T)+0.042074
      FK=(0.0000215*T)+0.013912
      FCP=(0.000021*T)+0.23676
      FD=(-0.0000591*T)+0.069724
      GOTO86
   81 FV=(0.00006725*T)+0.039395
      FK=(0.000026*T)+0.01312
      FCP=(0.0000025*T)+0.240283333
      FD=(-0.000249*T)+0.08675
      GOTO86
   82 FV=(0.00006375*T)+3.956666E-2
      FK=(0.000025*T)+0.01316
      FCP=(0.000005*T)+0.2401
      FD=(-0.0001365*T)+8.4486E-2
      GOTO86
   83 FV=(0.0000605*T)+0.03995
      FK=(0.0000245*T)+0.01321
      FCP=(0.00001*T)+0.2395
      FD=(-0.0001145*T)+0.08217
      GOTO86
   84 FV=(0.0000585*T)+4.023333E-2
      FK=(0.0000235*T)+0.01336
      FCP=(0.00001*T)+0.2395
      FD=(-0.000097*T)+0.07951
      GOTO86
   85 FV=(0.000055*T)+0.040994
      FK=(0.00002265*T)+0.013553
      FCP=(0.000016*T)+0.23826
      FD=(-0.0000768*T)+0.069724
   86 CONTINUE
      RETURN
      END
      SUBROUTINE H2O(WK,WD,H,WV,T)
      IF(T.LE.100.0)GOTO12
      IF(T.GT.100.0.AND.T.LE.200.0)GOTO13
      IF(T.GT.200.0)GOTO14
   12 WK=0.2096*T**0.119
      WD=62.5-3.88E-6*T**2.554
      H=1095.2-0.5826*T
      IF(T.GT.50.0)GOTO15
      WV=(0.095411*T**0.317422)*1.0E-5
      GOTO80
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      IF(TS(I).LE.50.0)GOTO91
      IF(TS(I).GT.50.0.AND.TS(I).LE.100.0)GOTO92
      IF(TS(I).GT.100.0.AND.TS(I).LE.150.0)GOTO93
      WD=62.5-2.08E-5*T**2.2
      H=1095.2-0.5826*T
      IF(T.GT.150.0)GOTO16
      WV=(0.99929*T**1.46671E-2)*1.0E-5
      GOTO80
   16 WV=(0.99965/(T**6.1013E-2))*1.0E-5
      GOTO80
   14 WK=0.396
      WD=62.5-2.08E-5*T**2.2
      H=1110.0-0.6587*T
      WV=(0.99965/(T**6.1013E-2))*1.0E-5
   80 CONTINUE
      RETURN
      END
      SUBROUTINE VAP(WVD,TS,PS,I)
      REAL TS(170),PS(170)
      IF(TS(I).LE.50.0)GOTO91
      IF(TS(I).GT.50.0.AND.TS(I).LE.100.0)GOTO92
      IF(TS(I).GT.100.0.AND.TS(I).LE.150.0)GOTO93
      IF(TS(I).GT.150.0.AND.TS(I).LE.200.0)GOTO94
      IF(TS(I).GT.200.0.AND.TS(I).LE.250.0)GOTO95
      IF(TS(I).GT.250.0.AND.TS(I).LE.350.0)GOTO96
      IF(TS(I).GT.350.0)GOTO97
   91 PS(I)=1.00509*(TS(I)**(-0.15998))
      GOTO97
   92 PS(I)=1.00204*(TS(I)**0.16176)
      GOTO97
   93 PS(I)=1.00108*(TS(I)**0.38676)
      GOTO97
   94 PS(I)=1.00057*(TS(I)**0.54210)
      GOTO97
   95 PS(I)=1.00148*(TS(I)**0.73438)
      GOTO97
   96 PS(I)=1.00076*(TS(I)**0.91805)
      GOTO97
   97 WVD=(PS(I)*144.0)/(85.77*(460.0+TS(I)))
      RETURN
      END
      SUBROUTINE BISECT(BETA,Y)
      DOUBLE PRECISION Y1,Y2,F1,F2,DELF,ABF,PRODF,Y3,F3,YF,Y,CALC,BETA
      Y1=1.0
      Y2=0.00002
   34 F1=CALC(Y1,BETA)
   35 F2=CALC(Y2,BETA)
      DELF=F1-F2
      ABF=DABS(DELF)
      IF(ABF.LT.0.0000000001)GOTO36
      PRODF=F1*F2
      IF(PRODF.GT.0.0)GOTO37
      Y3=(Y1+Y2)/2.0
      F3=CALC(Y3,BETA)
      IF(F3.LT.0.0)GOTO31
      IF(F1.LT.0.0)GOTO33
   32 Y1=Y3
      GOTO34
   31 IF(F1.LT.0.0)GOTO32
   33 Y2=Y3
      GOTO35
   36 YF=(Y1+Y2)/2.0
      Y=YF
      RETURN
   37 WRITE(6,38)
   38 FORMAT('-','CHOOSE NEW INITIAL GUESS VALUES')
      RETURN
      END
      DOUBLE PRECISION FUNCTION CALC(Y,BETA)
      DOUBLE PRECISION Y,BETA
      CALC=0.5*(Y**4.0)*DLOG(Y)*(DLOG(Y)-1.0)+.125*(Y**4.0)+(Y**2.0)*(0.
     *5*DLOG(Y)-0.25)+0.125-BETA
      RETURN
      END
      SUBROUTINE SURTEN(ST,T)
      IF(T.LE.50.0)GOTO41
      IF(T.GT.50.0.AND.T.LE.100.0)GOTO42
      IF(T.GT.100.0.AND.T.LE.150.0)GOTO43
      ST=(-0.00072*T)+0.5558
      GOTO44
   41 ST=(-0.0005*T)+0.534
      GOTO44
   42 ST=(-0.00062*T)+0.542
      GOTO44
   43 ST=(-0.00063*T)+0.5417
   44 ST=ST/100.0
      RETURN
      END
```

```
      END
      SUBROUTINE STAGER(JROW,STAGCZ,CZ)
      DIMENSION STAGCZ(25)
      IF(JROW.EQ.1)GOTO0601
      IF(JROW.EQ.2)GOTO0602
      IF(JROW.EQ.3)GOTO0603
      IF(JROW.EQ.4)GOTO0604
      IF(JROW.EQ.5)GOTO0605
      IF(JROW.EQ.6)GOTO0606
      IF(JROW.EQ.7)GOTO0607
      IF(JROW.EQ.8)GOTO0608
      IF(JROW.EQ.9)GOTO0609
      STAGCZ(JROW)=1.0
      GOTO0610
  601 STAGCZ(JROW)=0.68
      GOTO0610
  602 STAGCZ(JROW)=0.75
      GOTO0610
  603 STAGCZ(JROW)=0.83
      GOTO0610
  604 STAGCZ(JROW)=0.89
      GOTO0610
  605 STAGCZ(JROW)=0.92
      GOTO0610
  606 STAGCZ(JROW)=0.95
      GOTO0610
  607 STAGCZ(JROW)=0.97
      GOTO0610
  608 STAGCZ(JROW)=0.98
      GOTO0610
  609 STAGCZ(JROW)=0.99
  610 CZ=STAGCZ(JROW)
      RETURN
      END
      SUBROUTINE INLINE(IROW,ILCZ,CZ)
      REAL ILCZ(25)
      IF(IROW.EQ.1)GOTO0501
      IF(IROW.EQ.2)GOTO0502
      IF(IROW.EQ.3)GOTO0503
      IF(IROW.EQ.4)GOTO0504
      IF(IROW.EQ.5)GOTO0505
      IF(IROW.EQ.6)GOTO0506
      IF(IROW.EQ.7)GOTO0507
      IF(IROW.EQ.8)GOTO0508
      IF(IROW.EQ.9)GOTO0509
      ILCZ(IROW)=1.0
      GOTO0510
  501 ILCZ(IROW)=0.64
      GOTO0510
  502 ILCZ(IROW)=0.8
      GOTO0510
  503 ILCZ(IROW)=0.87
      GOTO0510
  504 ILCZ(IROW)=0.9
      GOTO0510
  505 ILCZ(IROW)=0.92
      GOTO0510
  506 ILCZ(IROW)=0.94
      GOTO0510
  507 ILCZ(IROW)=0.96
      GOTO0510
  508 ILCZ(IROW)=0.98
      GOTO0510
  509 ILCZ(IROW)=0.99
  510 CZ=ILCZ(IROW)
      RETURN
      END
      SUBROUTINE INLBM(ATLAS,SL,ODIA,TIRE,ST,BUT,MUT)
      REAL ODIA,MUT
      ATLAS=SL/ODIA
      IATLAS=ATLAS*1000.0
      ATLAS=IATLAS/1000.0
      TIRE=ST/ODIA
      ITIRE=TIRE*1000.0
      TIRE=ITIRE/1000.0
      IF(ATLAS.GE.1.2.AND.ATLAS.LT.1.3)GOTO0571
      IF(ATLAS.GE.1.3.AND.ATLAS.LT.1.75)GOTO0572
      IF(ATLAS.GE.1.75.AND.ATLAS.LT.2.4)GOTO0573
      IF(ATLAS.GE.2.4.AND.ATLAS.LT.3.4)GOTO0574
      GOTO0570
  571 IF(TIRE.GE.1.2.AND.TIRE.LT.1.3)GOTO0575
      IF(TIRE.GE.1.3.AND.TIRE.LT.1.75)GOTO0576
      IF(TIRE.GE.1.75.AND.TIRE.LT.2.4)GOTO0577
      IF(TIRE.GE.2.4.AND.TIRE.LT.3.4)GOTO0578
      GOTO0570
  575 BUT=0.348
      MUT=0.592
      GOTO0591
  576 BUT=0.275
```

```
      MUT=0.608
      GOTO591
  577 RUT=0.1
      MUT=0.704
      GOTO591
  578 RUT=0.0633
      MUT=0.752
      GOTO591
  572 IF(TIRE.GE.1.2.AND.TIRE.LT.1.3)GOTO579
      IF(TIRE.GE.1.3.AND.TIRE.LT.1.75)GOTO580
      IF(TIRE.GE.1.75.AND.TIRE.LT.2.4)GOTO581
      IF(TIRE.GE.2.4.AND.TIRE.LT.3.4)GOTO582
      GOTO570
  579 RUT=0.367
      MUT=0.786
      GOTO591
  580 RUT=0.25
      MUT=0.62
      GOTO591
  581 RUT=0.101
      MUT=0.702
      GOTO591
  582 RUT=0.0678
      MUT=0.744
      GOTO591
  573 IF(TIRE.GE.1.2.AND.TIRE.LT.1.3)GOTO583
      IF(TIRE.GE.1.3.AND.TIRE.LT.1.75)GOTO584
      IF(TIRE.GE.1.75.AND.TIRE.LT.2.4)GOTO585
      IF(TIRE.GE.2.4.AND.TIRE.LT.3.4)GOTO586
      GOTO570
  583 RUT=0.418
      MUT=0.57
      GOTO591
  584 RUT=0.299
      MUT=0.602
      GOTO591
  585 RUT=0.229
      MUT=0.632
      GOTO591
  586 RUT=0.198
      MUT=0.648
      GOTO591
  574 IF(TIRE.GE.1.2.AND.TIRE.LT.1.3)GOTO587
      IF(TIRE.GE.1.3.AND.TIRE.LT.1.75)GOTO588
      IF(TIRE.GE.1.75.AND.TIRE.LT.2.4)GOTO589
      IF(TIRE.GE.2.4.AND.TIRE.LT.3.4)GOTO590
      GOTO570
  587 RUT=0.29
      MUT=0.601
      GOTO591
  588 RUT=0.357
      MUT=0.584
      GOTO591
  589 RUT=0.374
      MUT=0.581
      GOTO591
  590 RUT=0.286
      MUT=0.608
      GOTO591
  570 WRITE(6,945)
  945 FORMAT('-','THE WRONG SPACING WAS USED')
  591 RETURN
      END
      SUBROUTINE STAGRM(ODIA,SL,TAX,FORM,ST,MUT,RUT)
      REAL MUT ,ODIA
      TAX=SL/ODIA
      ITAX=TAX*1000.0
      TAX=ITAX/1000.0
      FORM=ST/ODIA
      IFORM=FORM*1000.0
      FORM=IFORM/1000.0
      IF(TAX.GE.0.4.AND.TAX.LT.0.8)GOTO650
      IF(TAX.GE.0.8.AND.TAX.LT.0.95)GOTO651
      IF(TAX.GE.0.95.AND.TAX.LT.1.05)GOTO652
      IF(TAX.GE.1.05.AND.TAX.LT.1.2)GOTO653
      IF(TAX.GE.1.2.AND.TAX.LT.1.3)GOTO654
      IF(TAX.GE.1.3.AND.TAX.LT.1.7)GOTO655
      IF(TAX.GE.1.7.AND.TAX.LT.2.5)GOTO656
      IF(TAX.GE.2.5.AND.TAX.LT.3.5)GOTO657
      GOTO658
  650 IF(FORM.GE.2.5.AND.FORM.LT.3.5)GOTO659
      GOTO658
  659 RUT=0.213
      MUT=0.636
      GOTO660
  651 IF(FORM.GE.1.7.AND.FORM.LT.2.5)GOTO661
      IF(FORM.GE.2.5.AND.FORM.LT.3.5)GOTO662
  661 RUT=0.446
      MUT=0.571
```

```
      GOTO660
  662 BUT=0.401
      MUT=0.581
      GOTO660
  652 IF(FORM.GE.1.3.AND.FORM.LT.1.7)GOTO663
      GOTO658
  663 BUT=0.497
      MUT=0.558
      GOTO660
  653 IF(FORM.GE.1.7.AND.FORM.LT.2.5)GOTO664
      IF(FORM.GE.2.5.AND.FORM.LT.3.5)GOTO665
      GOTO658
  664 BUT=0.478
      MUT=0.565
      GOTO660
  665 BUT=0.518
      MUT=0.560
      GOTO660
  654 IF(FORM.GE.1.2.AND.FORM.LT.1.3)GOTO666
      IF(FORM.GE.1.3.AND.FORM.LT.1.7)GOTO667
      IF(FORM.GE.1.7.AND.FORM.LT.2.5)GOTO668
      IF(FORM.GE.2.5.AND.FORM.LT.3.5)GOTO669
      GOTO658
  666 BUT=0.518
      MUT=0.556
      GOTO660
  667 BUT=0.505
      MUT=0.554
      GOTO660
  668 BUT=0.519
      MUT=0.556
      GOTO660
  669 BUT=0.522
      MUT=0.562
      GOTO660
  655 IF(FORM.GE.1.2.AND.FORM.LT.1.3)GOTO670
      IF(FORM.GE.1.3.AND.FORM.LT.1.7)GOTO671
      IF(FORM.GE.1.7.AND.FORM.LT.2.5)GOTO672
      IF(FORM.GE.2.5.AND.FORM.LT.3.5)GOTO673
      GOTO658
  670 BUT=0.451
      MUT=0.568
      GOTO660
  671 BUT=0.460
      MUT=0.562
      GOTO660
  672 BUT=0.452
      MUT=0.568
      GOTO660
  673 BUT=0.488
      MUT=0.568
      GOTO660
  656 IF(FORM.GE.1.2.AND.FORM.LT.1.3)GOTO674
      IF(FORM.GE.1.3.AND.FORM.LT.1.7)GOTO675
      IF(FORM.GE.1.7.AND.FORM.LT.2.5)GOTO676
      IF(FORM.GT.2.5.AND.FORM.LT.3.5)GOTO677
      GOTO658
  674 BUT=0.404
      MUT=0.572
      GOTO660
  675 BUT=0.416
      MUT=0.568
      GOTO660
  676 BUT=0.482
      MUT=0.556
      GOTO660
  677 BUT=0.449
      MUT=0.570
      GOTO660
  657 IF(FORM.GE.1.2.AND.FORM.LT.1.3)GOTO678
      IF(FORM.GE.1.3.AND.FORM.LT.1.7)GOTO679
      IF(FORM.GE.1.7.AND.FORM.LT.2.5)GOTO680
      IF(FORM.GE.2.5.AND.FORM.LT.3.5)GOTO681
      GOTO658
  678 BUT=0.310
      MUT=0.592
      GOTO660
  679 BUT=0.356
      MUT=0.580
      GOTO660
  680 BUT=0.440
      MUT=0.560
      GOTO660
  681 BUT=0.421
      MUT=0.574
      GOTO660
  658 WRITE(6,946)
  946 FORMAT('-','THE WRONG SPACING PARAMETERS WERE USED')
  660 RETURN
      END
```

# APPENDIX F

## Description of Nomenclature in Program

# APPENDIX F

## Description of Nomenclature in Program

This appendix will contain a description of the nomenclature used in the program. The terms will be defined in the section of the program where they are first introduced, unless their meaning varies drastically from one part of the program to another.

Main Program:

EDIA - The diameter to the outside of the fin.

FINNC - The number of fins on the cold side of the tube.

FINNH - The number of fins on the hot side of the tube.

FINNUM - The total number of fins on the tube.

FINS - A flag which indicates whether or not the tube has fins.

HT - The height of the fin

IDIA - The inside diameter of the thermosyphon tube.

INL - A flag which tells whether the arrangement of tubes in the exchanger is in line or staggered.

JOEBLO - A flag

K - The thermal conductivity of the thermosyphon tube material.

LC - The length of tube on the cold side.

LH - The length of tube on the hot side.

LMTD - The log mean temperature difference.

LPLUS - The ratio of the length of the tube on the hot side to the length of tube on the cold side.

MC - The mass flow of air on the cold side.

MH - The mass flow of air on the hot side.

N - The number of rows of tubes in the exchanger.

NTUBES - The number of tubes per row (It should be noted that for the staggered arrangement this number is the maximum).

ODIA - The outside diameter of the tube.

REH - The Reynolds number for the hot side air flow.

REC - The Reynolds number for the cold side air flow.

REPLUS - The ratio of the Reynolds number on the hot side to that on the cold side.

SL - The lateral spacing of the tubes.

STE - The transverse spacing of the tubes.

TALL - The total height of the flow area on both cold and hot sides.

TEC - The inlet cold temperature.

TEH - The inlet hot temperature.

TYPE - This is used to indicate whether a counter-flow or parallel flow system is desired.

WIDTH - The width of the air flow area.

Subroutine PARLEL :

AT - The total area of heat transfer.

ATC - The area of heat transfer on the cold side.

ATH - The area of heat transfer on the hot side.

CZ -A variable used in the calculation of the convection on the outside. $1^{+}$varies depending upon the row being approached and the arrangement of the tube.

CZ1 - Same as 'CZ' but specifically for the hot side.

CZ7 - Same as 'CZ' but specifically for the cold side.

HEAT - The heat transferred in each row.

J - The row number that the air flow is approaching.

NTHERM - The number of thermosyphon tubes in the specific row being considered.

QT - The total heat transferred.

TCOLD - A cold temperature at any point 'J' in the heat exchanger.

THOT - A hot temperature at any point 'J' in the heat exchanger.

TIC - The same as 'TCOLD' except in a scalar form.

TIH - The same as 'THOT' except in a scalar form.

UT - The overall heat transfer coefficient.

UTC - The overall heat transfer coefficient on the cold side.

UTH - The overall heat transfer coefficient on the hot side.

Subroutine CONFLO :

HEATE - The amount of heat transferred in the present row of thermosyphon tubes.

J - The position in the heat exchanger with reference to the hot side.

KAY - The position in the heat exchanger with reference to the cold side.

L - The number of times the program has looped in the subroutine CONFLO

MO - Numerically equivalent to KAY-1

TCOLDE - The temperature of the cold air flow at any point within the heat exchanger.

THOTE - The temperature of the hot air flow at any point within the heat exchanger.

Subroutine CONDUC :

ACI - The area of the inside of the tube on the cold side of the flow.

ACO - The area of the outside of the tube on the cold side of the flow.

AHI - The area of the inside of the tube on the hot side of the flow.

AHO - The area of the outside of the tube on the hot side of the flow.

BULL - Used in conjunction with 'TORO' to form the limits for the bisection method solution of 'DELTT5'.

DELTTI - The difference between the temperatures of the hot air flow approaching the thermosyphon and the cold air flow which was calculated to be approaching the thermosyphon. (This is not necessarily the true value of the cold flow approaching the thermosyphon)

DELTTO - The difference between the temperatures of the hot air flow approaching the thermosyphon and the real cold air flow approaching the thermosyphon.

DELTT1 - The difference in temperature between the hot air flow and the outside wall of the tube on the hot side.

DELTT2 - The difference in temperature of the hot side wall on the outside and the inside.

DELTT3 - The difference in temperature of the hot side wall on the inside and the saturation temperature of the two phase liquid contained inside the tube.

DELTT5 - The temperature difference between the saturation temperature of the liquid contained inside the tube and the cold side wall on the inside.

ERROR - A variable which changes within the program so as to allow minimum error in the results but still enough so that the values converge within a reasonable amount of time.

FAB - Used in conjunction with 'FURD' to provide the interval for the bisection method approximation of 'PERC1'.

I - A counter.

ICOUNT - A counter.

IRUN - A counter.

JCOUNT - A counter.

JRON - A counter.

PERC1 - A variable percentage used to approximate the value of 'Q' for the whole conduction process in this thermosyphon.

PERC3 - A variable percentage used to approximate the correct value of 'Q3'.

PERC5 - A variable percentage used to approximate the correct value of 'Q5'.

PI - The numerical constant.

Q - The heat flux in this process.

QTEST - The difference between Q and Q5.

Q3 - The heat flux used to calculate 'R3' (This will eventually converge to the value of 'Q')

Q5 - The heat flux used to calculate 'R5' (This value will eventually converge to the value of 'Q')

R1 - The convective film resistance on the hot side.

R2 - The resistance of the wall on the hot side.

R3 - The resistance caused from the evaporation of the liquid in the thermosyphon tube.

R5 - The resistance caused from the condensation of liquid in the thermosyphon tube.

R6 - The resistance of the wall on the cold side.

R7 - The convective film resistance on the cold side.

TC - The temperature of the cold air flow approaching the thermosyphon.

TCCALC - The temperature of the cold air flow calculated to be approaching the thermosyphon.

TH - The temperature of the cold air flow approaching the thermosyphon.

TORO - Used in conjunction with 'BULL' to form the limits for the bisection method solution of 'DELTT5'.

TWCI - The cold side temperature of the wall on the inside.

TWCO - The cold side temperature of the wall on the outside.

TWHI - The hot side temperature of the wall on the inside.

TWHO - The hot side temperature of the wall on the outside.

ZING - A scalar form of the vector 'PERC1'.

Subroutine FONDUC :

AREABC - The area of the base of the tube on the cold side.

AREABH - The area of the base of the tube on the hot side.

AREAFH - The area of the fin on the hot side.

AREAFC - The area of the fin on the cold side.

EFF - The fin efficiency.

EM - A variable described in the equation to find the fin efficiency.

EX - A variable used to find the fin efficiency.

EQLENG - The equivalent length of a fin if it is considered to be a flat plate.

HOI12 - The hot side film coefficient of the outside based on the inside for a finned surface.

HOI67 - The cold side film coefficient of the outside based on the inside for a finned surface.

H12 - The hot side film coefficient for a finned surface.

H67 - The cold side film coefficient for a finned surface.

R12 - The resistance of the hot side for a finned surface.

R67 - The resistance of the cold side for a finned surface.

Subroutine BESSEL :

BIO - The Bessel function $I_o$

BI1 - The Bessel function I

BKO - The Bessel function $K_o$

BK1 - The Bessel function K

Subroutine FNH67B :

H67B - The film coefficient of the base on the cold side.

VM - The velocity of the cold air flow.

Subroutine FNH12B :

H12B - The film coefficient of the base on the hot side.

VM - The velocity of the hot air flow.

Subroutine FINDH1 :

H1 - The film coefficient of the tube on the hot side.

VM - The velocity of the hot air flow.

Subroutine FINDH7 :

H7 - The film coefficient of the outside of the tube on the cold side.

VM - The velocity of the cold air flow.

Subroutine FINDH5 :

BETA - The value of $\beta$ found in Rowsenhows equation.

H5 - The condensation film coefficient.

IRAD - The inside radius of the tube.

T - The mean temperature of the mixture inside the tube at which the properties were defined.

Subroutine FINDH3 :

H3 - The film coefficient for evaporation.

T - The mean temperature between the saturation temperature and the inside wall temperature on the hot side of the tube.

Subroutine FNH67F :

H67 - The outside film coefficient for the fins on the cold side.

VM - The velocity of air on the cold side.

Subroutine FNH12F :

H12 - The outside film coefficient for fins on the hot side.

VM - The velocity of air on the hot side.

Subroutine AIRPRO :

ACP - The specific heat of air at a given temperature.

AD -The density of air at a given temperature.

AK - The conductivity of air at a given temperature.

AV - The viscosity of air at a given temperature.

<u>Subroutine H2O</u> :

| | |
|---|---|
| H | - The enthalpy of water at a given temperature. |
| WD | - The density of water at a given temperature. |
| WK | - The conductivity of water at a given temperature. |
| WV | - The viscosity of water at a given temperature. |

<u>Subroutine VAP</u> :

| | |
|---|---|
| PS | - The saturation pressure at a given temperature. |
| WVD | - The vapor density. |

<u>Subroutine BISECT</u> :

| | |
|---|---|
| ABF | - The absolute value of 'DELF'. |
| DELF | - The difference between F1 and F2. |
| F1 | - The function evaluated at the point Y1. |
| F2 | - The function evaluated at the point Y2. |
| F3 | - The function evaluated at the point Y3. |
| PRODF | - The product of F1 and F2. |
| Y1 | - The upper limit for the bisection interval. |
| Y2 | - The lower limit for the bisection interval. |
| Y3 | - The midpoint of the bisection interval. |

Function CALC :

CALC - The value of the function upon which the bisection method is being used.

Subroutine SURTEN :

ST - The surface tension at a given temperature.

Subroutine STAGER :

STAGCZ - The coefficient Cz for a staggered tube arrangement.

Subroutine INLINE :

INLCZ - The coefficient Cz for the inline tube arrangement.

Subroutine INLBM :

ATLAS - A parameter depending upon the arrangement of tubes and the size of the tube relative to the longitudinal spacing of the tubes.

BUT - A variable which is used in the equation for forced convection.

IATLAS - An integer form of 'ATLAS' used for truncation.

ITIRE - An integer form of 'TIRE' used for truncation.

MUT - A variable which is used in the equation for forced convection.

TIRE - A parameter depending upon the arrangement of tubes and the size of the tube relative to the transverse spacing.

Subroutine STAGBM :

BUT - A variable in the equation for forced convection.

FORM - A parameter which depends upon the arrangement and the magnitude of the transverse spacing in relation to the diameter of the tubes.

IFORM - An integer form of 'FORM'.

ITAX - An integer form of 'TAX'.

MUT - A variable in the equation for forced convection.

TAX - A parameter which depends upon the arrangement and the magnitude of the longitudinal spacing relative to the tube diameter.

# APPENDIX G

## Input for the Program

# APPENDIX G

## How to Provide Input for the Program

The data cards should be arranged as follows:

#1 Put a '1' meaning counterflow or a '0' meaning parallel flow. The number is called 'TYPE' and is read in FORMAT (I1)

#2 Put a '0' meaning staggered or a '1' meaning inline arrangement. The number is called 'INL' and is read in FORMAT (I1)

#3 Enter the cold and hot inlet temperature and the Reynolds number of the cold flow. The numbers are called 'TEC', 'TEH', and 'REC' and are read in FORMAT (F6.2, 4X, F6.2, 4X, F8.2)

#4 Enter the number of rows in the exchanger calling it 'N'. This is read in FORMAT(I2)

#5 Enter 'LPLUS' in FORMAT(F4.2)

#6 Enter 'REPLUS' in FORMAT(F4.2)

#7 Enter 'WIDTH' in FORMAT(F4.1). The measurement is in inches.

#8 Enter 'TALL' in FORMAT(F4.1). The measurement is in inches.

#9 Enter 'K' in FORMAT(F5.1). The measurement is in

#10 Enter 'IDIA' in FORMAT(F4.3). The measurement is taken in inches.

#11 Enter 'ODIA' in FORMAT(F4.3). The measurement is taken in inches.

#12 Enter 'SL' in FORMAT(F3.1). The measurement is taken in inches

#13 Enter 'STE' in FORMAT(F3.1). The measurement is in inches.

#14 Enter 'NTUBES' in FORMAT(I1)

#15 Enter 'FINS' in FORMAT(I1)
Put a ' 0' if there are no fins or a '1' if there are fins.

The following entries only apply to finned surfaces:

#16 Enter 'H' in FORMAT(F7.4). The measurement is taken in inches.

#17 Enter 'EDIA' in FORMAT(F5.3). The measurement is taken in inches.

#18 Enter 'FINNUM' in FORMAT(F3.0)

# APPENDIX H

## Calibration of Orifice Plates and Thermocouple Reader

# APPENDIX H

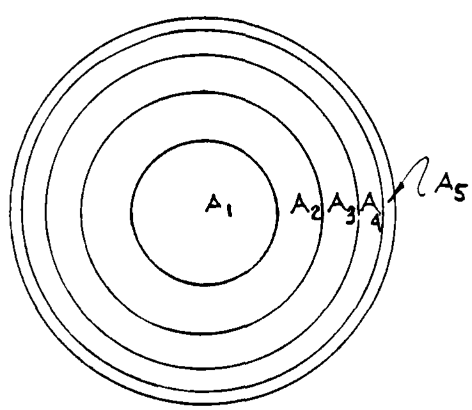
## Calibration of Orifice Plates and Thermocouple Reader

### Orifice Plates

Mass flow rates of the air were found from pressure drops across the orifice plates. The orifices were located at the suction inlets of both the hot and the cold ducting systems.

The orifices were calibrated by using a pitot tube to measure the average velocity through the section. This average velocity was then correlated to the pressure drop across the orifice plate. The area of the inside of the pipe was divided into concentric rings of equal area.(Fig. H.1)

Fig.H.1




The velocity readings are taken in the centres of each of the rings along a straight path through the centre of the pipe. The average velocity is then given by

$$V_{avg} = \sqrt{\frac{(\gamma_{ct} - \gamma_{tp})2g}{\gamma_{air}}} \times \sum_{i=1}^{2n} \frac{\sqrt{h_i}}{2n} \qquad (H.1)$$

The correlation equations used on the orifice plates are

$$V_h = 18.41\sqrt{\Delta P} + 2.17 \qquad (H.2)$$

$$V_c = 18.94\sqrt{\Delta P} + 1.53 \qquad (H.3)$$

for the orifice used in the hot and cold ducts respectively. This also implies the range of flows used experimentally.

## Multi-Channel Digital Thermocouple Reader

Upstream and downstream of both the hot and cold test sections air temperature readings were taken by means of copper-constantan thermocouples attached to a multi-channel digital thermocouple reader with an automatic ice point. The validity of these readings was checked by calibrating the digital reading against a similar thermocouple attached to a potentiometer. They were calibrated in a hot water bath operating over the same range of temperatures as the experimental air temperatures. A standard mercury thermometer was also placed in the hot water bath to insure a very small error in the calibration procedure. The expression used to correct the digital thermocouple reading is

$$T_h = 1.158\, T_h - 14.1 \qquad (H.4)$$

$$T_c = 1.156\, T_c - 15.5 \qquad (H.5)$$

# APPENDIX I

## Experimental Data

# APPENDIX I

## Experimental Data

Study A - Keeping the number of fins per side constant at 16 fins/side, the ratio $Re^*$ was varied.

Experiment A-1 ($Re^* = 0.5$)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 2280 | 0.51 | 740 | 1060 | 900 | 80.1 | 87.9 | 149.9 | 169.6 |
| 2 | 2890 | 0.51 | 880 | 1310 | 1100 | 77.8 | 85.2 | 150.4 | 169.9 |
| 3 | 3380 | 0.51 | 840 | 1570 | 1200 | 76.3 | 82.4 | 148.4 | 168.5 |
| 4 | 3660 | 0.51 | 910 | 1470 | 1190 | 79.5 | 85.5 | 150.7 | 167.9 |

Experiment A-2 ($Re^* = 0.75$)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1930 | 0.76 | 800 | 1270 | 1030 | 76.1 | 86.1 | 149.9 | 168.8 |
| 2 | 2620 | 0.77 | 920 | 1440 | 1180 | 78.8 | 87.3 | 151.0 | 166.7 |
| 3 | 3140 | 0.78 | 1000 | 1550 | 1280 | 81.1 | 88.8 | 152.7 | 166.7 |
| 4 | 3590 | 0.78 | 970 | 1680 | 1330 | 82.8 | 89.3 | 153.1 | 166.4 |
| 5 | 4010 | 0.79 | 1160 | 1910 | 1530 | 80.5 | 87.5 | 152.3 | 165.9 |

Experiment A-3 ($Re^* = 1.0$)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1450 | 1.0 | 750 | 1340 | 1050 | 74.6 | 87.1 | 149.9 | 169.9 |
| 2 | 1990 | 1.0 | 810 | 1560 | 1180 | 75.4 | 85.3 | 150.2 | 167.3 |
| 3 | 2400 | 1.0 | 930 | 1670 | 1300 | 76.1 | 85.5 | 151.5 | 166.6 |
| 4 | 2740 | 1.0 | 1020 | 1780 | 1400 | 77.8 | 86.8 | 152.3 | 166.3 |
| 5 | 3040 | 1.0 | 1080 | 1890 | 1490 | 78.9 | 87.5 | 152.7 | 166.2 |

Experiment A - 4 (Re* = 1.33)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 890 | 1.4 | 680 | 1020 | 1190 | 78.6 | 96.6 | 151.6 | 169.4 |
| 2 | 1470 | 1.4 | 860 | 1390 | 1120 | 78.6 | 92.5 | 151.3 | 166.4 |
| 3 | 2060 | 1.4 | 1030 | 1810 | 1420 | 75.7 | 87.8 | 151.8 | 166.0 |
| 4 | 2270 | 1.4 | 1060 | 1900 | 1480 | 78.1 | 89.4 | 153.7 | 167.2 |
| 5 | 1770 | 1.4 | 1000 | 1440 | 1220 | 78.7 | 92.3 | 152.6 | 165.6 |

Experiment A - 5 (Re* = 2.0)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 960 | 2.1 | 880 | 1240 | 1060 | 78.7 | 100.2 | 153.2 | 166.7 |
| 2 | 1120 | 2.1 | 900 | 1580 | 1240 | 78.9 | 98.0 | 153.2 | 168.1 |
| 3 | 1270 | 2.1 | 820 | 1580 | 1200 | 78.2 | 93.6 | 151.3 | 164.5 |
| 4 | 1380 | 2.1 | 970 | 1730 | 1350 | 79.2 | 96.0 | 154.4 | 167.7 |
| 5 | 1490 | 2.1 | 1020 | 1560 | 1290 | 79.0 | 95.3 | 152.8 | 164.0 |

<u>Study B</u> - Keeping the Re* ratio constant at 1.0 the number of fins per side was varied.

Experiment B - 1 (16 fins/side)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1450 | 1.0 | 750 | 1340 | 1050 | 74.6 | 87.1 | 149.9 | 169.9 |
| 2 | 1990 | 1.0 | 810 | 1560 | 1180 | 75.4 | 85.3 | 150.2 | 167.3 |
| 3 | 2400 | 1.0 | 930 | 1670 | 1300 | 76.1 | 85.5 | 151.5 | 166.6 |
| 4 | 2740 | 1.0 | 1020 | 1780 | 1400 | 77.8 | 86.8 | 152.3 | 166.3 |
| 5 | 3040 | 1.0 | 1080 | 1890 | 1490 | 78.9 | 87.5 | 152.7 | 166.2 |

Experiment B - 2 (12 fins/side)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1450 | 1.0 | 790 | 1380 | 1090 | 74.5 | 87.7 | 150.1 | 170.7 |
| 2 | 1960 | 1.0 | 930 | 1440 | 1180 | 79.1 | 90.5 | 150.8 | 166.4 |
| 3 | 2380 | 1.0 | 990 | 1730 | 1360 | 79.0 | 89.1 | 152.6 | 168.2 |
| 4 | 2710 | 1.0 | 950 | 1850 | 1400 | 82.6 | 91.0 | 152.0 | 166.6 |
| 5 | 3010 | 1.0 | 1040 | 1990 | 1520 | 81.0 | 89.3 | 151.8 | 166.0 |

Experiment B - 3 (8 fins/side)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1430 | 1.0 | 630 | 1130 | 880 | 80.3 | 90.8 | 151.0 | 167.8 |
| 2 | 1960 | 1.0 | 700 | 1250 | 970 | 81.3 | 89.9 | 152.8 | 166.4 |
| 3 | 2360 | 1.0 | 780 | 1450 | 1110 | 82.4 | 90.3 | 153.5 | 166.6 |
| 4 | 2690 | 1.0 | 800 | 1440 | 1120 | 83.7 | 90.9 | 153.6 | 164.0 |
| 5 | 2980 | 1.0 | 830 | 1270 | 1050 | 84.8 | 91.5 | 155.9 | 165.0 |

Experiment B - 4 ( 4 fins/side)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1450 | 1.0 | 490 | 1130 | 810 | 77.8 | 86.0 | 150.6 | 167.4 |
| 2 | 1980 | 1.0 | 550 | 1400 | 980 | 78.3 | 85.0 | 151.5 | 166.6 |
| 3 | 2400 | 1.0 | 620 | 1410 | 1020 | 78.6 | 84.9 | 151.9 | 164.7 |
| 4 | 2750 | 1.0 | 670 | 1590 | 1130 | 79.5 | 85.4 | 152.3 | 164.8 |
| 5 | 3050 | 1.0 | 640 | 1670 | 1160 | 80.2 | 85.3 | 153.0 | 164.9 |

Experiment B - 5 ( 2 fins/side)

| Run | $Re_c$ | $Re^*$ | $Q_c$ | $Q_h$ | $Q_{avg}$ | $T_{cin}$ | $T_{cout}$ | $T_{hin}$ | $T_{hout}$ |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1460 | 1.0 | 470 | 970 | 720 | 75.1 | 82.8 | 152.7 | 167.2 |
| 2 | 2010 | 1.0 | 560 | 1120 | 840 | 74.7 | 81.6 | 152.6 | 164.9 |
| 3 | 2410 | 1.0 | 600 | 1250 | 930 | 77.4 | 83.4 | 153.7 | 165.2 |
| 4 | 2780 | 1.0 | 640 | 1340 | 990 | 76.1 | 81.7 | 153.2 | 163.9 |
| 5 | 3070 | 1.0 | 670 | 1480 | 1080 | 76.8 | 82.1 | 154.4 | 165.0 |

# APPENDIX J

## Experimental Equipment

# APPENDIX J

## Experimental Equipment

Multichannel Digital Thermocouple Reader

- 20 channels calibrated for copper-constantan thermocouples for use between -100 to 1640°F

- 914 Numatron by Leeds and Northrup

1st Fan

- size 130Z-B design PH Arr.4 made by Sheldon Engineering of Galt,Ontario,Canaada.

2nd Fan

- size 32 Volume made by Canadian Blower and Forge Co. Ltd. of Kitchener,Ontario, Canada.

Multimanometer

- filled with manometer fluid(S.G = 0.8)
- made by T.E.M Instruments Ltd.

# APPENDIX K

## Program for Experimental Data

```
//                  JOB          (SH169145,
//   ),'CLEMENTS',MSGLEVEL=(2,0),
//   CLASS=K
//   EXEC WATFIV,TIME=(2,0),REGION=200K
//WATFIV.SYSIN  DD  *
$JOB    SHXXXXXX,PAGES=20,TIME=120
      DIMENSION TCI(5),TCO(5),THI(5),THO(5),P(10)
      REAL MC,MH
      PI=3.1416
      READ(5,100)(TCI(I),I=1,5)
      READ(5,100)(TCO(I),I=1,5)
      READ(5,100)(THI(I),I=1,5)
      READ(5,100)(THO(I),I=1,5)
  100 FORMAT(5(1X,F5.1))
      READ(5,101)(P(I),I=1,10)
  101 FORMAT(10(1X,F5.2))
      READ(5,102)PA
  102 FORMAT(F5.2)
      V4=18.942*(P(9)**0.5)+1.525
      V1=18.409*(P(10)**0.5)+2.173
      DO 10 I=1,5
      TCI(I)=1.156*TCI(I)-15.495
      TCO(I)=1.156*TCO(I)-15.495
      THI(I)=1.158*THI(I)-14.143
      THO(I)=1.158*THO(I)-14.143
   10 CONTINUE
      DO 20 I=1,10
      P(I)=PA-(0.795*62.4*P(I)/12.0**3)
   20 CONTINUE
      TTCI=0.0
      TTCO=0.0
      TTHI=0.0
      TTHO=0.0
      DO 30 I=1,5
      TTCI=TTCI+TCI(I)
      TTCO=TTCO+TCO(I)
      TTHI=TTHI+THI(I)
      TTHO=TTHO+THO(I)
   30 CONTINUE
      TTCI=TTCI/5.0
      TTCO=TTCO/5.0
      TTHI=TTHI/5.0
      TTHO=TTHO/5.0
      MC=(144.0*P(10)/(53.3*(TTCI+460)))*(V1*((PI/4.0)*(3.829/12.0)**2))
      MH=(144.0*P(9)/(53.3*(TTHI+460.0)))*(V4*((PI/4.0)*(3.829/12.0)**2)
     2)
      V2=(53.3*(TTCI+460.0)*MC)/((144.0*(P(1)+P(2))/2.0)*((5.5/12.0)**2)
     2)
      V5=(53.3*(TTHI+460.0)*MH)/((144.0*(P(7)+P(8))/2.0)*((5.5/12.0)**2)
     2)
      V3=(53.3*(TTCO+460.0)*MC)/((144.0*(P(3)+P(4))/2.0)*((5.5/12.0)**2)
     2)
      V6=(53.3*(TTHO+460.0)*MH)/((144.0*(P(5)+P(6))/2.0)*((5.5/12.0)**2)
     2)
      VC=(V2+V3)/2.0
      VH=(V4+V5)/2.0
      TH=(TTHI+TTHO)/2.0
      TC=(TTCI+TTCO)/2.0
      CALL FLUID(TC,FV,FK,FCP,FD)
      QC=MC*FCP*(TTCO-TTCI)*3600.0
      REC=(FD*V2*(.66/12.0))*3600.0/FV
      CALL FLUID(TH,FV,FK,FCP,FD)
      QH=MH*FCP*(TTHO-TTHI)*3600.0
      REH=(FD*V5*(0.66/12.0))*3600.0/FV
      RESTAR=REH/REC
      PRINT,'TCI',TCI,'TTCI',TTCI
      PRINT,'TCO',TCO,'TTCO',TTCO
      PRINT,'THI',THI,'TTHI',TTHI
      PRINT,'THO',THO,'TTHO',TTHO
      PRINT,'P',P
      PRINT,'MC',MC,'V1',V1,'V2',V2,'V3',V3,'VC',VC
      PRINT,'MH',MH,'V4',V4,'V5',V5,'V6',V6,'VH',VH
      PRINT,'QC',QC,'QH',QH,'REC',REC,'REH',REH,'RESTAR',RESTAR
      STOP
      END
      SUBROUTINE FLUID(T,FV,FK,FCP,FD)
C     THIS SUBROUTINE USES AIR PROPERTIES
      IF(T.LE.50.0)GOTO81
      IF(T.GT.50.0.AND.T.LE.100.0)GOTO82
      IF(T.GT.100.0.AND.T.LE.150.0)GOTO83
      IF(T.GT.150.0.AND.T.LE.200.0)GOTO84
```

```
          IF(T.GT.200.0.AND.T.LE.300.0)GOTO85
          FV=(0.0000515*T)+0.042074
          FK=(0.0000215*T)+0.013912
          FCP=(0.000021*T)+0.23676
          FD=(-0.0000591*T)+0.069724
          GOTO86
       81 FV=(0.00006725*T)+0.039395
          FK=(0.000026*T)+0.01312
          FCP=(0.0000025*T)+0.240283333
          FD=(-0.000249*T)+0.08675
          GOTO86
       82 FV=(0.00006375*T)+3.956666E-2
          FK=(0.000025*T)+0.01316
          FCP=(0.000005*T)+0.2401
          FD=(-0.0001365*T)+8.4486E-2
          GOTO86
       83 FV=(0.0000605*T)+0.03995
          FK=(0.0000245*T)+0.01321
          FCP=(0.00001*T)+0.2395
          FD=(-0.0001145*T)+0.08217
          GOTO86
       84 FV=(0.0000585*T)+4.023333E-2
          FK=(0.0000235*T)+0.01336
          FCP=(0.00001*T)+0.2395
          FD=(-0.000097*T)+0.07951
          GOTO86
       85 FV=(0.000055*T)+0.040994
          FK=(0.00002265*T)+0.013553
          FCP=(0.000016*T)+0.23826
          FD=(-0.0000768*T)+0.069724
       86 CONTINUE
          RETURN
          END
$ENTRY
  77.8  77.8  78.2  78.3  78.1
  83.6  84.2  84.0  84.4  83.6
 143.5 144.4 144.4 144.1 143.7
 153.3 154.8 154.8 154.8 155.5
  0.30  0.30  0.37  0.37  6.43  6.43  6.25  6.25  1.00  0.57
14.62
$STOP
```

# APPENDIX L

## A Description of the Heat Exchanger used for the Analysis

# APPENDIX L

## A Description of the Heat Exchanger used for the Analysis

| | |
|---|---|
| Type | Air to Air * |
| Flow Type | Parallel Flow |
| Arrangement | Staggered |
| Inlet Cold Side Temperature | 70°F |
| Inlet Hot Side Temperature | 200°F |
| Reynolds Number of the Cold Side | 6000 * |
| Number of Rows | 10 |
| Length Ratio ($L^+$) | 1.0 * |
| Reynolds Number Ratio ($Re^*$) | 1.0 * |
| Channel Width | 5.5" |
| Tube Material | Brass |
| Tube Material Conductivity | 34 Btu/hr ft F |
| Inside Diameter of Tubes | 0.5" |
| Outside Diameter of Tubes | 0.66" |
| Longitudinal Spacing | 1.5" |
| Transverse Spacing | 1.5" |

| | |
|---|---|
| Number of Tubes in First Row | 5 |
| Number of Fins | 24 * |
| Height of Fins | 0.0625" |
| Diameter of Fins | 0.96" |

* Varied where indicated

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