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20. Abstract
 This report documents a study of a prototype air-to-air heat exchanger of which the thermal conductivity is governed by the reservoir temperature of the heat pipes. The work is introduced with a brief review of the basic operating principles of heat pipes and of the various existing schemes for variable conductance. An experimental facility, incorporating a computerized data acquisition system, was developed to investigate the steady state performance characteristics of the heat exchanger over a wide range of air inlet temperatures, mass flow rates, and reservoir temperatures. Some tests were conducted imposing a single operating temperature on all of the heat pipes, while in others the heat pipe rows were allowed to operate independently. The results are expressed in terms of heat exchanger effectiveness versus normalized reservoir temperature for various values of the ratio of the mass flow rates of the air streams. Conclusions are drawn as to the internal behaviour of this type of heat pipes under various operating regimes. An original computer model is presented which combines a quasi analytical solution for the air and fin temperature profiles in the heat exchanger with a heat pipe simulation routine based on the hydrodynamic equations involved in the working fluid cycle. Computational results are shown, and a brief comparison with the experiments is made. Possible modifications that would improve the accuracy and the versatility of the program are discussed.

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This technical report has been reviewed and is approved for publication.

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ABSTRACT

This report documents a study of a prototype air-to-air heat exchanger of which the thermal conductivity is governed by the reservoir temperature of the heat pipes. The work is introduced with a brief review of the basic operating principles of heat pipes and of the various existing schemes for variable conductance.

An experimental facility, incorporating a computerized data acquisition system, was developed to investigate the steady state performance characteristics of the heat exchanger over a wide range of air inlet temperatures, mass flow rates, and reservoir temperatures. Some tests were conducted imposing a single operating temperature on all of the heat pipes, while in others the heat pipe rows were allowed to operate independently. The results are expressed in terms of heat exchanger effectiveness versus normalized reservoir temperature for various values of the ratio of the mass flow rates of the air streams. Conclusions are drawn as to the internal behaviour of this type of heat pipes under various operating regimes.

An original computer model is presented which combines a quasi-analytical solution for the air and fin temperature profiles in the heat exchanger with a heat pipe simulation routine based on the hydrodynamic equations involved in the working fluid cycle. Computational results are shown, and a brief comparison with the experiments is made. Possible modifications that would improve the accuracy and the versatility of the program are discussed.

LIST OF SYMBOLS

c_p	specific heat of air at constant pressure	J/kg°C
d_p	external diameter of heat pipe	m
e	plate thickness	m
f	friction coefficient	
g	gravitational constant	m/s ²
h	total convective heat transfer coefficient	W/m ² °C
k	thermal conductivity	W/m°C
l	length of heat pipe section	m
\dot{m}	mass flow rate	kg/s
p	static pressure	N/m ²
s	fin spacing	m
t	fin thickness	m
u	bulk velocity	m/s
w	width of heat exchanger	m
x	flow direction	m
y	lateral direction	m
z	heat pipe axis	m
<u>A</u>	coefficient matrix in the numerical solution of the heat exchanger node temperatures	
<u>B,C</u>	coefficient vectors in the numerical solution	
A_{HX}	wetted area in one side of heat exchanger	m ²
A_n	normal flow area	m ²
A_f	projected fin area	m ²
MR	mass flow ratio	

N_p	number of heat pipes	
Q	heat transfer rate	W
T	temperature	$^{\circ}\text{C}, ^{\circ}\text{K}$
ϵ	effectiveness	
ϕ	heat flux in a row of heat pipes	W
$\bar{\phi}$	normalized heat flux in a row of heat pipes	$^{\circ}\text{C}/\text{m}$
τ	wall shear stress	N/m
ρ	density	kg/m
x	non dimensionalized x coordinate	
\bar{x}	normalized platelet length	

Subscripts

a	"adiabatic" or "air"
c	"condenser"
e	"evaporator"
i	platelet number, node number
l	"liquid"
n	"normalized"
R	"reservoir"
v	"vapour"
0	"leading edge of a platelet"
1	"trailing edge of a platelet"
∞	"upstream of the heat exchanger"

Superscript

i	platelet number
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1. INTRODUCTION

A problem that has always been associated with the design of effective air-to-air heat exchangers is their high bulk and weight, due to the very large surfaces required to transfer significant amounts of heat from one airstream to another. The birth and rapid growth of heat pipe technology in the 1960s led to the eventual development of more compact heat exchangers taking advantage of the very high thermal conductivity of these devices. The continued development of such heat exchangers brought the promise of many new applications in the fields of energy conservation and air-conditioning. At the same time, however, a significant limitation of existing heat pipe heat exchangers became apparent, namely, a lack of controllability. While it is undisputably a high performance heat transfer device, the standard heat pipe's thermal conductivity is highly dependent on its operating temperature and on the thermal load placed across it; in other words, the overall heat transfer coefficient is a function of the heat sink and source conditions. With the standard heat pipe, it is impossible to adjust this heat transfer coefficient, and therefore impossible to regulate the heat flux in a heat exchanger using them.

This limitation poses problems for heat recovery and temperature control applications. In heat recovery systems, in which waste heat is recovered from exhausted air (or process gases) and used to heat cold incoming air, it is desirable to maximize the heat transfer rate, i.e., the heat exchanger's effectiveness. Uncontrollable heat pipe heat exchangers work, however, at an effectiveness which is not necessarily their optimum. The working point is passively determined by the temperature and mass flow rate of the two airstreams. Such heat exchangers are also inadequate to meet the requirements of air-conditioning systems, where one must control the exit temperature of one of the airstreams, since the air exit conditions are uniquely determined by the inlet conditions. This lack of

controllability is the motive behind the research on variable conductance heat pipe heat exchangers. Much work has been done to develop both passively and actively controlled heat pipes. Some of the resulting systems will be mentioned in the following sections.

This report documents both an experimental and numerical study of a prototype variable conductance heat pipe (VCHP) heat exchanger based on the principle of operating temperature modulation. This particular design, which makes use of liquid reservoirs of excess working fluid, offers the possibility of heat flux maximization and temperature control without moving parts. The objectives of this study are to determine the steady state performance characteristics of this heat exchanger over a wide range of operating conditions and to apply heat pipe theory to develop a numerical model of the heat exchanger capable of predicting temperature profiles and heat flux for specified air inlet conditions. Such a model would be useful as a tool for both the design and control of this type of heat exchanger.

2. HEAT PIPE PRINCIPLES

2.1 The standard heat pipe

Before discussing the details of the liquid reservoir VCHP, it is useful to review the fundamental principles of heat pipes. The heat pipe is a high performance heat transfer device, able to transfer heat at high rates over considerable distances with almost no temperature drop. In its basic form the heat pipe consists of a sealed tube lined with a porous capillary wick that is saturated with a volatile fluid as shown in figure 1. Heat absorbed by the evaporator zone vaporizes the fluid; vapour flows to the other end where it condenses, giving up the heat. The fluid cycle is then completed as the condensed fluid flows in the wick back to the evaporator. The axial heat transfer is thus equal to the mass circulation rate times the latent heat of vaporization of the working fluid.

Frequently the evaporator and condenser sections are separated by an adiabatic zone, where there is no heat flux into or out of the pipe. A heat pipe can, in principle, operate with the evaporator above the condenser as shown in figure 1, in which case the capillary head generated in the wick must overcome the gravitational head, but in many cases the evaporator is intentionally placed below the condenser so that these two forces work together, which facilitates the return of the condensate to the evaporator.

What appears to be a simple device actually involves a complex set of thermal and hydrodynamic processes, which have made it difficult for researchers to draw firm conclusions about the exact nature of the heat transfer processes in a particular heat pipe for a given set of heat source and sink conditions. What has been determined is that the heat pipe's performance is bounded by an envelope of limiting conditions as shown in figure 2.

2.2 Limits to heat transfer

The range of vapour temperature (also called operating temperature) over which a particular heat pipe functions is determined principally by the working fluid used. At the low end of this range, fluid viscosity imposes a limit on the heat transfer capacity of the heat pipes, as shown in figure 2.

For slightly higher operating temperatures, what is called the sonic limit is the dominant limiting factor. It can be shown analytically that there is a correspondence between constant area flow in a heat pipe with mass addition (evaporator) and removal (condenser) and constant mass flow in a converging-diverging nozzle. Consequently, just as there is a sonic (Mach = 1) limitation on the flow velocity through a nozzle, there is a similar limit on the flow velocity through the adiabatic section of the heat pipe.

Another phenomenon which may limit heat flux over a certain range of operating temperature is liquid entrainment in the vapour. In a heat pipe, the vapour and the liquid generally flow in opposite directions. Since they are in contact at the wick surface (if the wick is saturated), this sets up a mutual drag at the vapour-liquid interface. If the relative velocity between the liquid and vapour becomes too great, the interface becomes unstable and droplets of liquid will be torn from the wick and entrained in the vapour. Since this liquid never reaches the evaporator, it cannot contribute to the heat transferred by the heat pipe. Thus, the maximum axial heat transfer is no longer equal to the maximum fluid circulation rate times the latent heat of vaporization, but is some lower value.

The fact that there exists a maximum capillary head for any wick-fluid combination results in a hydrodynamic limit on heat pipe capacity. In order for the heat pipe to operate, the following pressure balance equation must be satisfied

between any two points along the heat pipe :

$$\Delta p_{c,max} + \Delta p_g \geq \Delta p_l + \Delta p_v \quad (2.1)$$

where $\Delta p_{c,max}$ \equiv maximum capillary head

Δp_g gravitational head acting on liquid (evaporator assumed to be positioned below the condenser)

Δp_l pressure drop in liquid due to viscous drag

Δp_v pressure drop in vapour due to viscous drag and mass transfer.

This equation states that for any fluid path defined by two points along the heat pipe, the sum of the driving forces (capillary head and gravitational head) must be equal to the sum of the pressure losses undergone by the liquid and vapour. These pressure losses are dependent on the fluid circulation rate (and hence the thermal load), whereas the maximum capillary head and the gravitational head are essentially independent of load. When the load increases so that the two sides of the equation are equal anywhere along the heat pipe, the capillary pumping limited has been reached.

Finally, there exists a boiling (burnout) limit which corresponds to the formation of a stable vapour film within the wick due to the excessively rapid production of bubbles, whose escape is inhibited by the wick structure. This condition tends to arise for high operating temperatures where there is a large temperature differential across the evaporator wick.

In all cases, the heat pipe operates within the area bounded by the curves shown in figure 2. For the standard heat pipe, the actual operating point is determined solely by the heat source and sink temperatures. The goal of variable conductance designs is to be able to displace this point at will, and in some cases even to change the shape of the operating envelope defined by these limiting conditions.

2.3 Variable conductance heat pipes

For reasons already stated, the concept of variable conductance heat pipes was extensively developed beginning in the late 1960s. Most variable conductance designs employ a second fluid, an inert non-condensable gas, to act as a buffer to the heat transfer process. The inert gas, which in principle remains unmixed with the working fluid, is compressed into a reservoir at the condenser end of the heat pipe by the total pressure of the working vapour (Fig. 3). The non-condensable gas blocks a portion of the condenser whose length depends on the thermal load on the heat pipe. As the thermal load increases, the inert gas is further compressed, exposing more of the condenser to the working vapour, thus increasing the heat transfer rate and limiting variations in heat source and/or sink temperature.

The heat transfer rate may either be controlled passively, as just described, or actively, in which case the inert gas reservoir is heated or cooled in order to regulate more precisely the position of the vapour/gas interface and hence the effective length of the condenser section.

There exist other schemes for variable conductance heat pipes; these are mostly the so called vapour flow modulation or liquid flow modulation types, where a throttling mechanism is placed either in the vapour or liquid phase of the flow circuit. This may consist of a variable orifice through which the vapour passes on its way from the evaporator to the condenser, or it may be some sort of controllable interruption or impediment to condensate flow in the wick. Unlike the non-condensable gas reservoir type of VCHP, both of these latter concepts require the use of moving parts.

All of the above mentioned methods of heat pipe control have several properties in common. First, the quantity of working fluid present in the heat pipe remains

constant regardless of the control setting. Secondly, these control systems can only attenuate the heat flux from some maximum value attainable were the control system not present. These systems, then, meet the needs of only one class of applications discussed in the introduction, namely temperature control, and this only when the thermal potential placed across the heat pipes is always greater than or equal to the required heat transfer.

2.4 Liquid reservoir variable conductance heat pipes

The liquid reservoir VCHP is based on a different principle than the preceding types. It uses no direct intervention in the fluid circulation process to control heat flux; instead, the thermal conductance is varied by modulating the heat pipe's vapour temperature. Referring to the cutaway view in figure 4, one sees that the evaporator end of each heat pipe is connected to an unwicked reservoir capable of holding condensed working fluid. Vapour can flow freely from the reservoir to the heat pipe, or vice versa, whereas liquid condensate can only run down from the heat pipe into the reservoir, since the heat pipe is operated in the vertical position. The heat pipe/reservoir combination contains slightly more working fluid than is required to completely saturate the wick; thus, a liquid/vapour interface always exists within the reservoir. A possible exception is when the heat pipe is functioning at a high heat transfer rate with a high reservoir temperature, a situation in which reservoir dryout could occur. Since heat pipe operation depends on the existence of pressure differentials between the liquid and vapour phases in the evaporator and condenser, as well as pressure gradients within each phase (as expressed by equation 2.1), it cannot truly be said that the liquid and vapour phases are in equilibrium. The variations in vapour pressure and temperature are so small, however, that the vapour can be considered saturated everywhere

in the heat pipe and in the reservoir. Consequently, the operating temperature (and hence the operating pressure) is determined by the temperature of the liquid in the reservoir. For the experimental work described in this report, the assumption was made that due to the high conductivity of the copper reservoir walls, the externally measured reservoir temperature is approximately equal to the vapour temperature in the heat pipe.

Variable conductance is achieved in this context by variation of the reservoir temperature, which, by imposing an operating temperature and pressure on the heat pipe, causes a particular operating regime to be established by the interplay of thermal and hydrodynamic processes in the heat pipe. This operating regime, with its corresponding thermal conductivity, is characterized by the amount of fluid present in the heat pipe and by its circulation rate.

2.5 The prototype

The heat exchanger being investigated is shown schematically in figure 3 and its photograph appears in figure 4. It consists of an inline array of sixteen finned heat pipes, where each row of four heat pipes, transverse to the flow direction, has a single reservoir serving all of the heat pipes in that row. There is a divider plate located midway along the heat pipe which divides the heat exchanger into evaporator and condenser sections. As seen in the photograph, a short length of the divider plate is bent over at the ends, forming tabs which effectively block part of the heat exchanger face, thereby creating a short adiabatic zone between the evaporator (bottom) and condenser (top). A summary of the key heat exchanger dimensions, as well as the known and estimated heat pipe parameters is given in Appendices A and C.

3. THE EXPERIMENTAL INVESTIGATION

3.1 The test facility

To make optimum use of the VKI laboratory facilities, the heat pipe heat exchanger was adapted to the Institute's already existing solar energy test set-up (Ref. 1). The modified system, with the heat exchanger in place, is sketched in figure 6. The system consists simply of two open circuits, the hot air path passing through the evaporator side of the heat exchanger, the cold air path through the condenser side. Each circuit is equipped with a fixed speed blower, an orifice plate flow meter, and a valve to regulate the mass flow rate. The hot air circuit is equipped with an electrical resistance heater that has power settings of 2, 4 and 6 Kw. The cold air circuit is equipped with a water-to-air heat exchanger capable of cooling the air to approximately 15°C (depending on the water flow rate) before it passes through the condenser. All of the tests referred to in this report were conducted in the counterflow configuration.

As indicated in figure 6, the test facility is outfitted with a microprocessor-controlled data acquisition system linked to a mini-computer capable of doing real time data reduction. Both the microprocessor and the mini-computer control programs (Ref. 2) are adaptations of the programs used to perform data acquisition in the SUN experiments. The versions used for this study permit the scanning of up to 48 thermocouples as well as two pressure transducers, with a total acquisition time of about 30 seconds and a variable interval between acquisitions.

Temperature control of the liquid reservoirs was achieved in one of three different ways, depending on the type of test :

1. All four reservoirs were placed in a large oil bath (Fig. 7), whose temperature was maintained nearly uniform by means of a rheostatically controlled electrical resistance heater and a rotating agitator.
2. All four reservoirs were placed in insulating sleeves to effectively eliminate heat transfer between the reservoirs and the surroundings.
3. The reservoirs were placed in individual baths, one of which contained heated oil as in case 1, while the other three contained ice water. This permitting the testing of the heat exchanger with only one row of heat pipes active at a time.

3.2 Instrumentation of the heat exchanger

The heat exchanger was sandwiched between two coupling sections, which were designed to provide a smooth transition of the flow in the 160 mm diameter circular ducting to the rectangular flow section of the heat exchanger. Near the inlet and outlet of each side of the heat exchanger there were apertures in the coupling sections which gave access to the test sections. The whole assembly consisting of heat exchanger and coupling sections was mounted on a support designed for testing the heat exchanger at 0°, 30°, 45°, 60° and 90° from the vertical. All measurements discussed in this report were performed with the heat exchanger in the vertical position.

Static pressure taps were drilled in the walls of the coupling sections opposite the test section apertures (Fig. 7). A pressure transducer was connected to each pair of taps to measure the pressure differential across the evaporator and condenser sections of the heat exchanger. Although several types (Setra, Validyne) of transducers were used during the course of experiments, they all had a high sensitivity (range ~ 70 mm H₂O) and they all gave an output on the order of one volt, so as to be compatible with the data acquisition system without the use of intermediate amplifiers.

Two types of probes were placed in the test sections, a vane anemometer to measure the velocity of the air stream and thermocouples to measure its temperature. A plexiglas sliding mechanism (Fig. B) was designed to hold the anemometer and seal the test section apertures, while allowing manual displacement of the probe in the y and z directions (x being the flow direction). All velocity measurements were made without heating or cooling of the airstream and the data acquisition system was not used. For the thermal measurements, thermocouples were placed at the inlet and outlet of both sides of the heat exchanger. A plate, through which were passed two thermocouple wires, was placed over the aperture at each inlet section. These thermocouple wires were drawn taut by fine copper wires which passed through the static pressure taps and were clamped there by the plastic pressure tubing. The wires were positioned such that the temperature sensitive thermocouple junctions were situated midway across the test section. This simple but effective system was adapted in order to minimize perturbation of the flow upstream of the heat exchanger.

In both the evaporator and condenser outlet test sections was placed a grid of fifteen thermocouples, arranged in five rows of three thermocouples (Fig. 9). The thermocouple heads protrude from hollow steel tubing, which can slide with respect to the mounting plate, thus permitting a degree of freedom in the lateral position of the thermocouple columns. For the tests described in this paper, the three columns (in each grid) of thermocouples were aligned with the spaces between the columns of heat pipes.

There were additional thermocouples connected to the data acquisition system. One of these was, of course, a reference, which was placed in an ice bath; the others measured the ambient air temperature and the external temperature of each reservoir.

3.3 Preliminary measurements :
flow characteristics

3.3.1 Velocity profiles

Velocity profile measurements were conducted for two reasons : first, to get an idea of the uniformity of the flow at the inlet and outlet of the heat exchangers, and second, to determine the optimal placement of the thermocouples at the outlet section.

It was found that when the probe was placed directly downstream of a column of heat pipes, the measured air velocity varied quite erratically as the probe was displaced in the z direction, indeed, falling to zero (or at least below the sensitivity threshold of the anemometer) at some points along the measurement line. We concluded from this that there remains a turbulent wake downstream of the heat exchanger behind each column of heat pipes in which the vane anemometer is unsuitable for velocity measurement. The non-uniformity of this wake (as indicated by the vane anemometer) may be due to a high degree of sensitivity, in the wake region, to small irregularities in fin geometry and spacing. This possibility is suggested by the fact that the lightweight fins had been deformed at various points on both faces of the heat exchanger due to previous mishandling. Due to the presence of these highly irregular wake regions, it was decided to represent the velocity profiles in the outlet section in terms of measurements made along lines in the streamplanes midway between the heat pipe columns. This was also the reason for positioning the thermocouple columns midway between the heat pipe wake regions.

The two dimensional inlet and outlet profiles were measured for the evaporator section with a mass flow rate of approximately .1 kg/s. At the inlet section there were

velocity gradients in both the vertical and lateral directions. A careful examination of the heat exchanger assembly revealed that the non-uniformity of the inlet velocity profile corresponded to a misalignment of the air inlet pipe. The outlet velocity profile, though based on a much sparser data field, indicates that as the air passes through the heat exchanger, a high degree of flow uniformization occurs in the lateral direction, whereas the average vertical velocity gradients tended to remain constant. This is intuitively comprehensible; air that enters fins with velocity gradients in both the y and z directions has a chance to redistribute itself in a plane parallel to the fins, thereby reducing lateral gradients, but not in a plane normal to the fins, once the air is trapped between the fins. Hence, except for some slight vertical flow redistribution that may take place immediately upstream of the heat exchanger, the vertical velocity gradients are preserved. These findings lead one to conclude that, although it may be difficult to achieve a totally uniform flow field at the inlet, special attention should be paid toward minimizing velocity gradients normal to the fins.

The most important conclusion to be drawn from these tests is that the outlet velocity profile was sufficiently uniform (out of the wake zone) to justify a simple averaging of all the temperatures given by the thermocouple grid to determine the bulk temperature of the outlet air stream.

3.3.2 Pressure drop

As part of the preliminary testing program, a series of measurements was conducted to determine the functional relationship between the mass flow rate through the heat exchanger and the resulting pressure drop. This information is used for two purposes :

1. During the thermal measurements, the mass flow rates are computed from the pressure drop across each side of the heat exchanger. This is a very direct method of flow metering, one which eliminates errors due to leaks at other points in the flow circuit, which in fact were found to be a problem in the SUN installation.

2. A derivation of the convective heat transfer coefficient in the heat exchanger, necessary for the computer model, is possible knowing the function relating mass flow rate to pressure drop.

We define the friction factor with the following equation, which relates the average component in the flow direction of the wall shear stress in the heat exchanger to the dynamic pressure of the air based on its bulk velocity in the inlet section

$$\tau = f \frac{\rho u^2}{2} \quad (3.1)$$

where f = friction factor.

τ = average axial component of wall shear stress in the heat exchanger

ρ = air density

u = bulk velocity of air in inlet section

The mass flow rate through the heat exchanger is given by :

$$\dot{m} = \rho u A_n \quad (3.2)$$

where A_n = cross sectional flow area (inlet section)

Taking the heat exchanger itself as a control volume, a force balance on the air gives

$$\Delta p A_n = \tau A_{HX} \quad (3.3)$$

where A_{HX} = heat exchange surface area.

Combining equations 3.1 and 3.3, one gets

$$u = \sqrt{\frac{2c}{\rho f}} \sqrt{\Delta p}$$

where $c = \frac{A_n}{A_{HX}}$. Substituting this into equation 3.2 gives

$$\dot{m} = A_n \sqrt{\frac{2\rho c}{f}} \sqrt{\Delta p} \quad (3.4)$$

The lengths of the evaporator and condenser sections are not exactly equal; there are different values for A_n and A_{HX}

A_n = height \times width

$$(A_n)_e = .24 \quad .13 = .0312 \text{ m}$$

$$(A_n)_c = .22 \quad .13 = .0286 \text{ m}$$

A_{HX} = 2 (number of fins) (exposed fin area) +
+ (number of heat pipes) (exposed pipe area)

$$A_{HX} = \frac{L}{s} \left[2 \left(A_f - N_p \frac{\pi d_p^2}{4} \right) + N_p \pi d_p (s-t) \right]$$

l = height of section
 s = fin spacing
 t = fin thickness
 A_f = projected area of fin
 N_p = number of heat pipes
 d_p = diameter of heat pipes

$$(A_{HX})_e = 4.23 \text{ m}^2$$

$$(A_{HX})_c = 3.88 \text{ m}^2$$

Hence, $C = \frac{A_n}{A_{HX}} = .00738$ for both sides of the heat exchanger.

Figures 11 and 12 show the mass flow rate (as measured with a reference orifice plate flow meter, Ref. 3) plotted with respect to the square root of the pressure drop measured across the evaporator and condenser sections of the heat exchanger, respectively. These measurements showed very little random error and were highly repeatable. Clearly, the most striking feature of the data is its linearity above a certain threshold mass flow rate of about .04 kg/s. This means that in equation 3.4, the friction factor f can be considered constant in this range, provided a constant term is added to the right hand side of the equation to compensate for the fact that the extrapolation of the linear portion of the curve does not pass through the origin. There is no a priori reason that this must be so; indeed, the friction factor in any internal flow situation is a function of Reynolds number. It appears, however, that above a certain mass flow rate, the friction factor for this particular geometry achieves Reynolds number independence, as does the friction factor for turbulent flow in rough pipes.

A best fit to the points in the linear range gives the following expressions for mass flow rate.

Condenser

$$\dot{m} = .0142 \sqrt{\Delta p} - .0036 \quad [\Delta p] = \text{Pa} \quad (3.5a)$$

$$\dot{m} = .0444 \sqrt{\Delta p} - .0336 \quad [\Delta p] = \text{mm H}_2\text{O} \quad (3.5b)$$

$$\dot{m} = .202 \ell_c \sqrt{\Delta p} - .0336 \quad [\Delta p] = \text{mm H}_2\text{O} \quad (3.5c)$$

$$[\ell_c] = \text{meters}$$

Evaporator

$$\dot{m} = .0150 \sqrt{\Delta p} - .0340 \quad [\Delta p] = \text{Pa} \quad (3.6a)$$

$$\dot{m} = .0471 \sqrt{\Delta p} - .0340 \quad [\Delta p] = \text{mm H}_2\text{O} \quad (3.6b)$$

$$\dot{m} = .0196 \ell_e \sqrt{\Delta p} - .0340 \quad [\Delta p] = \text{mm H}_2\text{O} \quad (3.6c)$$

$$[\ell_e] = \text{meters}$$

The slope of these lines is equal to the grouping $A_n \sqrt{2 \rho c / f}$, from which f can be determined. The following values are found.

condenser : $f = .078$

evaporator : $f = .074$

Since the friction factor (assuming Reynolds independence) is only a function of flow geometry, it is reasonable to assume that the friction factor for both sides of the heat exchanger should be the same. For simplicity, then, we will use the average value $f = .076$.

3.4 Thermal measurements

3.4.1 Data acquisition and processing

Figure 12 is a close-up view of the data acquisition system (DAS), which rapidly gathers and processes all relevant pressure and temperature data. Figure 13 shows the heat exchanger mounted on its support, outfitted with thermocouples and pressure transducers, and connected to the DAS. During each acquisition cycle, the MEKII microprocessor feeds data to the PET minicomputer, where BASIC control program then performs calculations and prints out the results. What follows is a description of the calculations in the data reduction process.

Mass flow rate

The mass flow rates through the evaporator and condenser sections are computed on the basis of the pressure transducer output using equations 3.5c and 3.6c.

Heat transfer rate

The heat transfer rate between air streams is not measured directly. The measurements only allow the calculation of the heat given up by the hot air and the heat absorbed by the cold air. Since the heat pipes are connected to liquid reservoirs, which may act as either heat sources or sinks, these two heat transfer rates are not necessarily the same. They are given by the expressions

$$Q_e = \dot{m}_e c_p (T_{1,e} - T_{2,e}) \quad (3.7)$$

$$Q_c = \dot{m}_c c_p (T_{2,c} - T_{1,c}) \quad (3.8)$$

The bulk outlet temperatures $T_{2,e}$ and $T_{2,c}$ are determined as mentioned in section 3.3.1, by averaging the 15 values given

by each thermocouple grid. The reservoir power is defined as the difference between the condenser and evaporator heat transfer rates :

$$Q_R = Q_c - Q_e \quad (3.9)$$

Effectiveness

The effectiveness of a heat exchanger is a dimensionless parameter used to compare the performance of different heat exchangers or of the same heat exchanger under different operating conditions. This quantity, denoted ϵ , is defined as the ratio of the actual heat transfer rate in the heat exchanger to the maximum thermodynamically possible heat transfer rate. The maximum possible heat transfer rate is given by the maximum temperature difference existing within the heat exchanger, the difference between the two inlet temperatures, times the lower value of the heat capacity rate of the flow :

$$Q_{\max} = (\dot{m}c_p)_{\min} (T_{1,e} - T_{1,c})$$

In the present case, the heat capacity c_p of the air is assumed constant; thus, $(\dot{m}c_p)_{\min}$ always pertains to the flow with the minimum mass flow rate.

The two principal heat transfer rates in the heat exchanger, Q_e and Q_c , lead to two different definitions of the effectiveness :

$$\text{Evaporator effectiveness} \quad \epsilon_e = \frac{Q_e}{Q_{\max}}$$

$$\text{Condenser effectiveness} \quad \epsilon_c = \frac{Q_c}{Q_{\max}}$$

There are two dimensionless parameters computed during the data reduction process which are useful both in prescribing the test conditions and organizing the effectiveness data. There are the mass flow ratio and the normalized reservoir temperature.

Mass flow ratio, MR

The mass flow ratio is defined as the air mass flow rate through the condenser section divided by the air mass flow rate through the evaporator :

$$MR = \frac{\dot{m}_c}{\dot{m}_e}$$

Normalized reservoir temperature, T_n

Since the relationship of reservoir temperature to effectiveness is of primary interest in this study, it is necessary to express reservoir temperature in some way that permits comparisons of tests under different thermal loads for various reservoir temperatures. In order that the chosen transformation be valid, it must be independent of the performance of the heat exchanger, i.e., it must be independent of all temperatures downstream of the inlets. This leaves little choice but to define a parameter that indicates the reservoir temperature's relative position between the cold and hot air inlet temperatures :

$$T_n = \frac{T_R - T_{1,c}}{T_{1,e} - T_{1,c}}$$

It should be mentioned here that the liquid reservoir VCHP heat exchanger could be considered a reservoir power modulated system rather than a reservoir temperature modulated

one, in which case it would be appropriate to define a normalized reservoir power, Q_R/Q_{max} . A somewhat different test facility would have been required to conduct measurements from this point of view, one in which reservoir power could be accurately measured (now it can only be determined by taking the difference between Q_c and Q_e , which already are of only marginal accuracy). The author feels that the temperature formulation is more fundamental to the physical processes responsible for the variable conductance.

3.4.2 Global heat exchanger performance - Tests in large oil bath

A test matrix was established in which each point was defined by three main parameters : heater power, mass flow ratio and reservoir temperature. The heat exchanger was tested at thermal loads corresponding to heater settings of 2, 4 and 6 kW : at mass flow ratios of .4, .6, .8, and 1.0; and at irregular intervals in the normalized temperature range $0 < T_n < 1$. In addition, several tests were run at $T_n < 0$ and $T_n > 1$ to see the effect of operating outside the normal reservoir temperature range.

For each test point, the data acquisition system was activated once the test conditions (valve, air heater and rheostat settings) were established. The program output (heat transfer rates, effectiveness and temperature profile) was monitored until steady state was reached, typically taking about 20 minutes; then the DAS was stopped and reactivated, with subsequent acquisitions being recorded on magnetic cassette. An example of the DAS output appears in Appendix B.

The evaporator and condenser effectiveness for $MR = 1$ are plotted in figure 14. The points, although corresponding to tests under different thermal loads, admit of a relatively smooth curve fit, which validates the method of non-dimensionalizing the reservoir temperature. The two curves, evaporator

and condenser effectiveness, cross at a point designated the adiabatic operating point. At normalized reservoir temperatures below the adiabatic operating temperature, the evaporator effectiveness is higher than the condenser effectiveness, the difference corresponding to the rate of heat transfer from the heat pipes to the reservoirs. Under steady state conditions, this heat loss can only be by conduction, since if it were by convection the heat pipe wick would gradually empty of liquid. This corresponds to a condition in which there is dryout at the lower end of the evaporator section in at least some of the heat pipes, a condition manifested by a steep gradient in the air temperature profile measured at the exit of the evaporator section (Fig. 15). The temperature profile shown probably represents a superposition of several staggered temperature steps, since the dryout boundary (if any) may be at a different location in each row of heat pipes.

At the adiabatic operating point, there must be no influx of liquid into the reservoir, since that would necessitate reservoir heating in order to revaporize the fallen liquid. There are, then, two possible adiabatic operating situations: One is where the reservoirs are completely dried out, with all of the liquid circulating in the heat pipes. In this case, heat input to the reservoir serves only to maintain it at an elevated temperature (near the hot air inlet temperature), compensating for heat losses by conduction, radiation, etc. The second is where there is some condenser liquid in the reservoir; the adiabatic operating temperature in this case is that at which the reservoir liquid is in equilibrium with the heat pipe vapour. This situation is plausible with or without dryout in the heat pipe evaporator section.

Above the adiabatic operating point, the condenser heat transfer is greater than that of the evaporator, being the sum of the heat input to the evaporator and the heat input to the reservoirs. The true effectiveness of the heat exchanger, at any given operating temperature, must be regarded as the

lower of these two curves, since it is always the lower curve that represents the heat actually exchanged between the two air streams, not taking into account heat which might have been lost or gained from the reservoirs. The significant feature of the super-adiabatic operating range ($T_n > T_a$) is that after increasing nearly linearly with T_n , the evaporator effectiveness reaches a peak and then decreases as the normalized reservoir temperature approaches 1.0. This decline in evaporator effectiveness is explained by the fact that at high normalized reservoir temperatures there is a low thermal gradient across the liquid layer in the evaporator section, and hence a low heat flux. The maximum in the evaporator effectiveness curve indicates the maximum heat exchange rate between air streams that can be achieved by the heat exchanger for these inlet conditions (and with uniform reservoir temperature).

The usable heat flux range can effectively be extended if one's principal purpose is to heat cold air. Although the maximum heat recovery from the hot air is given by the evaporator effectiveness, the heat supplied to the reservoirs is also recovered at the condenser; thus, in cases where the thermal potential of the evaporator is insufficient to meet the heat demand at the condenser, power can be supplied to meet the deficit. It should be noted, however, that this is an inefficient mode of operation, since increasing the normalized reservoir temperature beyond T_{max} reduces the effectiveness of the evaporator, so the additional reservoir power required to attain the desired rate of heat transfer at the condenser will be greater than the original deficit.

Figure 16 is a plot of the true heat exchanger effectiveness versus normalized reservoir temperature. It represents the ensemble of the large oil bath test results; all thermal loads and mass flow ratios are included. The effect of decreasing the ratio of mass flow rates from 1.0 is to displace the effectiveness curve upward and slightly to the right, giving both a higher maximum effectiveness and

higher optimal operating temperature. The rise in effectiveness with decreasing mass flow ratio is typical of all fluid-to-fluid heat exchangers.

3.4.3 Adiabatic operating point - Individually insulated reservoirs

For these tests each reservoir was wrapped in a tightly fitting sleeve made of closed-cell foam. This independent insulation allowed each row of heat pipes to establish its own adiabatic operating regime. Indeed, it was found during these tests that each heat pipe row had a slightly different adiabatic operating temperature. To compare the results obtained under these adiabatic test conditions with the "adiabatic points" given by the intersection of the evaporator and condenser effectiveness curves, one must average the temperatures of the insulated reservoirs in order to obtain an equivalent normalized reservoir temperature. The comparison is therefore not rigorous, and some data suggests that it may be misleading in the case where the heat transfer rate is a small fraction of the heat exchanger's rated capacity. The point corresponding to the highest heat transfer rate tested (MR = 1, air heater power = 6 kw) is plotted against the general effectiveness curves in figure 14.

An unexpected and still unexplained finding made during the tests with individually insulated reservoirs is that the first and third reservoirs had a higher temperature than the second and fourth, the first row being the one closest to the hot air inlet. One would expect the reservoir temperatures to either increase or decrease monotonically from one row to the next, depending on the direction of the flow. That this is not the case indicates that for some reason the heat pipe rows are not all operating in the same regime, as characterized by their respective normalized reservoir temperature evaluated with respect to the local evaporator and condenser temperatures.

The results of these tests are summarized in table 1. It is worth noting that the heat losses associated with adiabatic operation increase, as a percentage of condenser effectiveness, as the overall heat transfer rate decreases. This is seen in the relatively large spread between the evaporator and condenser effectiveness found in the last five rows of the table.

3.4.4 Performance of a single row of heat pipes - Individual reservoir baths

To compare the individual performance of each row of heat pipes, four tall narrow baths were used for reservoir temperature control, three of them containing ice water and the fourth containing oil with a resistance heater. The reservoirs were tested consecutively under the following (nominal) conditions : MR = .8, heater power = 4 kW, normalized reservoir temperature = 0.7.

The test results are summarized in table 2. They show that there are no large-order differences in effectiveness among the four rows of heat pipes. It is not clear whether the differences that do exist result from differences in heat flux path length in the fins, which varies with the position of the heat pipe row tested, or rather differences in the heat pipes themselves, such as non-uniformities in wick porosity, quantity of working fluid, or in other basic heat pipe characteristics. This question could be resolved with a numerical model of the heat exchanger, capable of simulating single row operation.

4. DEVELOPMENT OF A NUMERICAL MODEL OF THE HEAT EXCHANGER

4.1 Simplifying assumptions

In addition to the experimental investigation described above, it was undertaken to numerically model the heat exchanger. This represented a substantial challenge, for if the detailed behaviour of a single liquid reservoir VCHP is not fully understood, the overall comportment of a finned tube heat exchanger containing sixteen such heat pipes in cross-flow is understood still less. Having no prior indications as to the feasibility or validity of a complete computer model of the heat exchanger, we* considered it appropriate to begin with the following simplifying assumptions :

- (1) There is no longitudinal temperature gradient in either the evaporator or condenser sections of the heat exchanger. Similarly, there are no transverse temperature gradients. The only non-uniform temperature profiles, whether of the fins or of the air, are in the flow direction.
- (2) As in the experimental study, only steady state operation is considered.
- (3) The liquid and vapour phases in the reservoirs are in equilibrium.

Consequently, there is no mass interchange between the heat pipes and the reservoirs. This condition corresponds to the adiabatic operation described in sections 3.4.2 and 3.4.3.

* This program was developed in conjunction with Paul-Hervé Theunissen, doctoral candidate at the Université Libre de Bruxelles.

4.2 Representation of the fin/tube system

As a consequence of assumption 1 of the preceding section we may effectively compress all of the fins in the evaporator section into a single flat plate, and likewise those in the condenser section. Because the fins are thin and have high thermal conductivity, the plate temperature is assumed to be constant across its thickness. These two plates, one representing the evaporator side of the heat exchanger, the other the condenser side, are coupled to each other by the four rows of heat pipes. To facilitate an analytical (and therefore more elegant and efficient) solution of the air and fin temperature profiles, this coupling is simulated as a series of four linear heat sinks/sources normal to the flow direction, their position corresponding to that of the heat pipe rows. These four linear heat sinks and sources subdivide the evaporator and condenser plates, respectively, each into five "platelets" (Fig. 17), for which an analytical solution to the convective and conductive heat transfer equations can be developed.

Because of the high thermal conductivity of the fins and their large surface area relative to the heat pipe area, we assume that all of the heat transferred between the air and the heat pipes passes through the fins. In reality, the isothermal lines in the fins (plates) would tend to form contours around the heat pipes; due to the linearization of the heat sinks/sources, however, we make the further idealization that the isothermal lines are linear and perpendicular to the flow direction (in accordance with assumption 1 of the preceding section).

4.2.1 The plate equations

In any given platelet, the two governing heat transfer equations are the one dimensional conduction equation,

$$ke \frac{d^2T}{dx^2} + h(T_a - T) = 0 \quad (4.1a)$$

and the convective heat transfer equation,

$$\frac{\dot{m}c_p}{w} \frac{dT_a}{dx} + h(T_a - T) = 0 \quad (4.2a)$$

where k = thermal conductivity of plate

e = plate thickness = (thickness of fin) \times
(number of fins)

h = total convective heat transfer coefficient =
(convective heat transfer coefficient) \times
(number of fin sides)

w = width of plate

\dot{m} = total mass flow rate through one side of heat
exchanger

T = local plate temperature

T_a = local bulk air temperature

These equations can be simplified using the following
non-dimensionalizations :

$$x = \frac{wh}{\dot{m}c_p}$$

$$\alpha = \frac{(\dot{m}c_p)^2}{w^2ehk} \quad \text{giving}$$

$$\frac{d^2T}{dx^2} + \alpha(T_a - T) = 0 \quad (4.1b)$$

$$\frac{dT_a}{dx} + T_a - T = 0 \quad (4.2b)$$

4.2.2 Solution for the plate temperature, T

Consider figure 18, which represents the i^{th} platelet.

$$\text{Let } T_0 = T_i(0)$$

$$T_1 = T(X_i)$$

$$T_{a0} = (T_a)_i \Big|_{x=0}$$

$$T_{a1} = (T_a)_i \Big|_{x=X_i}$$

Integrating equation 4.1b once, after substituting the term $(T_a - T)$, one obtains

$$\frac{dT}{dx} - \frac{dT}{x} \Big|_0 = \alpha(T_a - T_{a0})$$

or

$$\frac{1}{\alpha} \frac{dT}{dx} + \left(T_{a0} - \frac{1}{\alpha} \frac{dT}{dx} \Big|_0 \right) = T_a$$

$$\text{Define } T_\alpha \equiv \left(T_{a0} - \frac{1}{\alpha} \frac{dT}{dx} \Big|_0 \right); \text{ then} \quad (4.3)$$

$$\frac{1}{\alpha} \frac{dT}{dx} + T_\alpha = T_a$$

Substituting back into equation (4.2b), one gets

$$\frac{d^2T}{dx^2} + \frac{dT}{dx} - \alpha T + \alpha T_\alpha = 0$$

or

$$\frac{d^2(T-T_\alpha)}{dx^2} + \frac{d(T-T_\alpha)}{dx} - \alpha(T-T_\alpha) = 0$$

Equation 4.4 is a second order linear differential equation with the characteristic equation

$$r^2 + r - \alpha = 0$$

The general solution of this equation is

$$T - T_\alpha = Ae^{\omega_1 X} + Be^{\omega_2 X}$$

$$\text{where } \left. \begin{array}{l} \omega_1 \\ \omega_2 \end{array} \right\} = \frac{-1 \pm \sqrt{1+4\alpha}}{2} \quad (4.6ab)$$

4.2.3 Determination of constants A and B

We start with the implicit boundary conditions on T_0 and T_1 :

$$T_0 - T_\alpha = A+B$$

$$T_1 - T_\alpha = Ae^{\omega_1 X} + Be^{\omega_2 X}$$

Then,

$$A = (T_0 - T_\alpha) - B$$

Define $\Delta \equiv T - T_\alpha$. Then,

$$\Delta_1 = [\Delta_0 - B] e^{\omega_1 X} + B e^{\omega_2 X}$$

$$\therefore B = \frac{\Delta_1 - \Delta_0 e^{\omega_1 X}}{e^{\omega_2 X} - e^{\omega_1 X}} \quad (4.7a)$$

$$A = \frac{\Delta_1 - \Delta_0 e^{\omega_2 X}}{e^{\omega_1 X} - e^{\omega_2 X}} \quad (4.7b)$$

The complete solution of the temperature profile for any given platelet is thus determined if T_0 , T_1 and T_α are known. To determine T_α , $\left. \frac{dT}{dx} \right|_{x=0}$ must be determined in terms of

the known temperatures, assumed for the moment to be T_0 , T_1 and $T_{\alpha 0}$. Differentiating equation 4.5 gives, at the platelet's leading edge :

$$\left. \frac{dT}{dx} \right|_{x=0} = \omega_1 A = \omega_2 B \quad (4.8)$$

$$\left. \frac{dT}{dx} \right|_{x=0} = \frac{\omega_1 (-\Delta_1 + \Delta_0 e^{\omega_2 X}) + \omega_2 (\Delta_1 - \Delta_0 e^{\omega_1 X})}{e^{\omega_2 X} - e^{\omega_1 X}}$$

$$\text{Define } \beta_1 \equiv \frac{\omega_2 - \omega_1}{\begin{pmatrix} \omega_2 X & \omega_1 X \\ e & -e \end{pmatrix}} \quad \text{and} \quad \beta_2 \equiv \frac{\begin{pmatrix} \omega_2 X & \omega_1 X \\ \omega_1 e & -\omega_2 e \end{pmatrix}}{\begin{pmatrix} \omega_2 X & \omega_1 X \\ e & e \end{pmatrix}} \quad (4.9ab)$$

then

$$\left. \frac{dT}{dx} \right|_{x=0} = \Delta_1 \beta_1 + \Delta_0 \beta_2$$

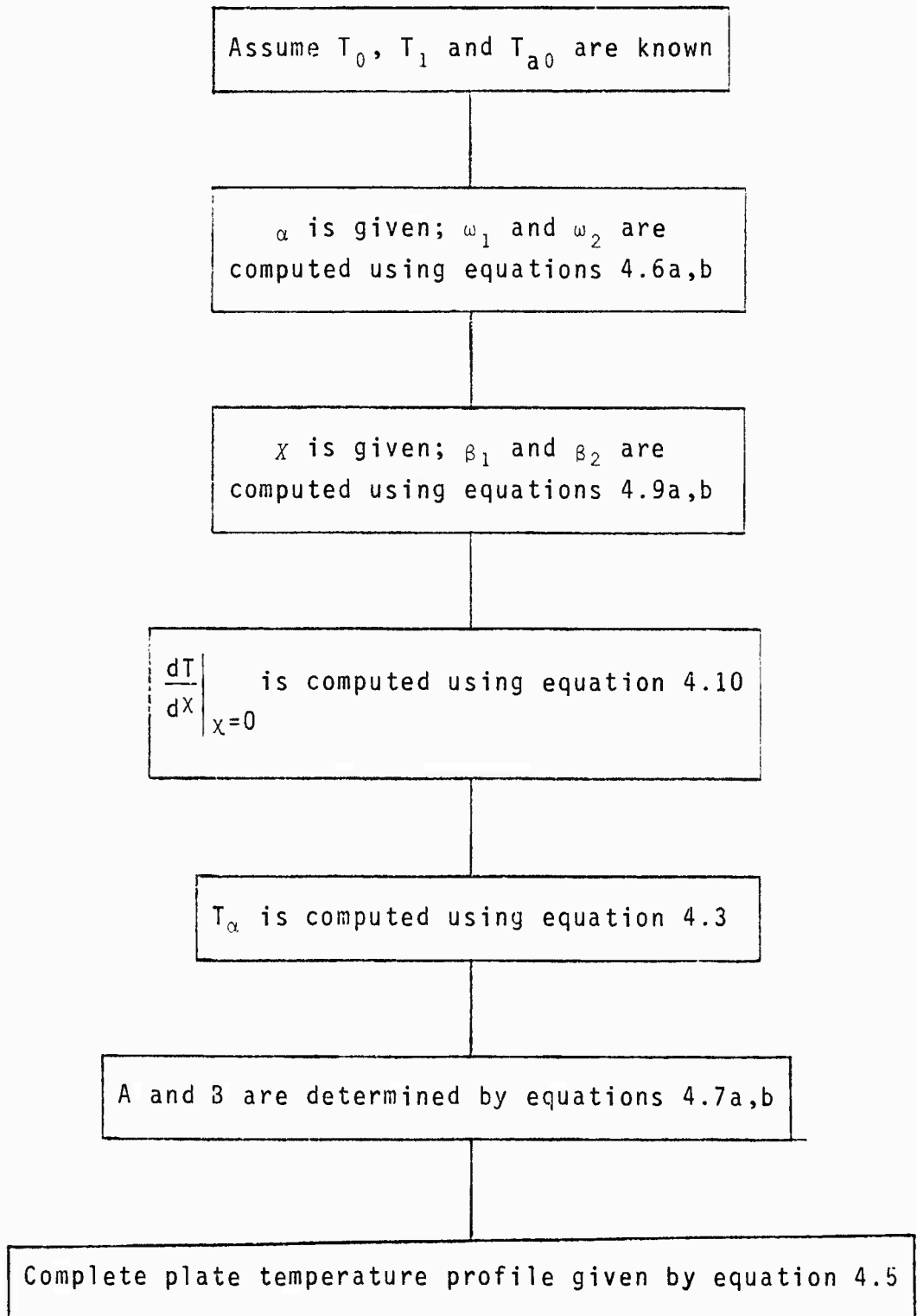
If one defines $\tilde{\Delta}_1 \equiv T_1 - T_{a0}$ and $\tilde{\Delta}_0 \equiv T_0 - T_{a0}$, it can be shown that

$$\left. \frac{dT}{dx} \right|_{x=0} = \frac{\tilde{\Delta}_1 \beta_1 + \tilde{\Delta}_0 \beta_2}{1 - \frac{1}{\alpha} (\beta_1 + \beta_2)} \quad (4.10)$$

Thus, $\left. \frac{dT}{dx} \right|_{x=0}$ hence T_α , has been determined in terms of the

"known" temperatures T_0 , T_1 and T_{a0} , thereby enabling a complete solution of the platelet temperature profile.

4.2.4 Summary of platelet temperature calculation



4.2.5 Solution for the air temperature, T_a

The expression for the air temperature is built up from equations 4.2b and 4.5.

$$\frac{dT_a}{dX} + T_a = T_\alpha + Ae^{\omega_1 X} + Be^{\omega_2 X} \quad (4.11)$$

The general solution to this differential equation is of the form

$$T_a = T_\alpha + Ce^{-X} + \xi_1 e^{\omega_1 X} + \xi_2 e^{\omega_2 X} \quad (4.12)$$

The particular solution is obtained as follows :

$$(\xi_1 + \omega_1 \xi_1) e^{\omega_1 X} + (\xi_2 + \omega_2 \xi_2) e^{\omega_2 X} = Ae^{\omega_1 X} + Be^{\omega_2 X}$$

$$\therefore \xi_1 = \frac{A}{1 - \omega_1} \quad \xi_2 = \frac{B}{1 + \omega_2}$$

The constant C is determined by setting X equal to zero :

$$T_{a0} - T_\alpha = C + \xi_1 + \xi_2$$

$$C = T_{a0} - T_\alpha - \xi_1 - \xi_2$$

$$C = \frac{1}{\alpha} \left. \frac{dT}{dX} \right|_0 - \xi_1 - \xi_2$$

The air temperature at the platelet's trailing edge is then given as

$$T_{a1} = T_{\alpha} + Ce^{-X} + \xi_1 e^{\omega_1 X} + \xi_2 e^{\omega_2 X} \quad (4.13)$$

4.2.6 The temperature gradient in the platelet at the trailing edge

Equation 4.10 expresses the temperature gradient (which is proportional to the heat flux) in the platelet at its leading edge. A similar expression can be obtained for the trailing edge.

$$\begin{aligned} \left. \frac{dT}{dX} \right|_X &= \omega_1 A e^{\omega_1 X} + \omega_2 B e^{\omega_2 X} \\ &= \frac{\omega_1 \rho^{\omega_1 X} \begin{bmatrix} -\Delta_1 + \Delta_0 e^{\omega_2 X} \\ e^{\omega_2 X} & -e^{\omega_1 X} \end{bmatrix} + \omega_2 e^{\omega_2 X} \begin{bmatrix} \Delta_1 - \Delta_0 \\ \Delta_1 - \Delta_0 & \omega_1 X \end{bmatrix}}{\begin{pmatrix} \omega_2 X & \omega_1 X \\ e & -e \end{pmatrix}} \end{aligned}$$

$$\text{Define } S_1 \equiv \frac{-\omega_1 e^{\omega_1 X} + \omega_2 e^{\omega_2 X}}{e^{\omega_2 X} - e^{\omega_1 X}}$$

$$S_2 \equiv \frac{(\omega_1 + \omega_2) X e^{(\omega_1 + \omega_2) X}}{\begin{pmatrix} \omega_2 X & \omega_1 X \\ e & -e \end{pmatrix}}$$

Then

$$\left. \frac{dT}{dX} \right|_X = \Delta_1 S_1 + \Delta_0 S_2 \quad (4.14)$$

4.2.7 Linearization of the equations

Equations 4.10, 4.13 and 4.14 are all implicitly linear equations in terms of the three temperatures, T_0 , T_1 , and T_{a0} , which up to now have been treated as known boundary conditions. The explicit expression of $\frac{dT}{dX}\Big|_0$, $\frac{dT}{dX}\Big|_X$ and $T_{\alpha 1}$ in terms of these three variable involves length algebraic manipulations that will not be shown here. The resulting equations are given below :

$$(a) \quad \frac{dT}{dX}\Big|_0 = \gamma_1 T_1 + \gamma_2 T_0 - (\gamma_1 + \gamma_2) T_{a0} \quad (4.15)$$

$$\text{where } \gamma_1 \equiv \frac{\beta_1}{1 - \frac{1}{\alpha} (\beta_1 + \beta_2)}$$

$$\gamma_2 \equiv \frac{\beta_2}{1 - \frac{1}{\alpha} (\beta_1 + \beta_2)}$$

$$(b) \quad \frac{dT}{dX}\Big|_X = \mu_1 T_1 - \mu_2 T_0 - (\mu_1 + \mu_2) T_{a0} \quad (4.16)$$

$$\mu_1 = S_1 + \frac{(S_1 + S_2) \gamma_1}{\alpha}$$

$$\mu_2 = S_2 + \frac{(S_1 + S_2) \gamma_2}{\alpha}$$

$$(c) \quad T_{a1} = v_1 T_1 + v_2 T_0 - (v_1 + v_2 - 1) T_{a0} \quad (4.17)$$

$$v_1 = \epsilon_3 - \epsilon_1 + \gamma_1 \epsilon_6$$

$$v_2 = \epsilon_2 - \epsilon_4 + \gamma_2 \epsilon_6$$

$$\epsilon_1 = \frac{-e^{-X} + e^{\omega_1 X}}{(1 + \omega_1) \begin{pmatrix} e^{\omega_2 X} & \omega_1 X \\ -e & \end{pmatrix}}$$

$$\epsilon_2 = e^{\omega_2 X} \epsilon_1$$

$$\epsilon_3 = \frac{-e^{-X} + e^{\omega_2 X}}{(1 + \omega_2) \begin{pmatrix} e^{\omega_2 X} & \omega_1 X \\ -e & \end{pmatrix}}$$

$$\epsilon_4 = e^{\omega_1 X} \epsilon_3$$

$$\epsilon_5 = e^{-X} - 1$$

$$\epsilon_6 = \frac{\epsilon_2 + \epsilon_3 + \epsilon_5 - \epsilon_1 - \epsilon_4}{\alpha}$$

4.2.8 Assembly of the platelets

Up to now, the entire analytical development has been with regard to a single platelet, a segment of the composite plate which represents one side of the heat exchanger. These segments, or platelets, are joined by lines of heat addition or removal, which represent the rows of heat pipes. For each side of the heat exchanger, our mathematical model links the platelets together using the physically imposed boundary conditions of continuity of temperature in the air and conservation

of energy in the platelet. At each node (boundary between platelets), the following conditions apply :

$$T_{a_0}^{i+1} = T_{a_1}^i \quad (4.18)$$

$$\phi_i = kwe \left[\frac{dT}{dx} \Big|_{x_i}^i - \frac{dT}{dx} \Big|_0^{i+1} \right] \quad (4.19)$$

where ϕ_i = heat addition rate at the boundary between platelets i and $i+1$ (heat transfer rate in the i^{th} row of the pipes)

The boundary conditions at the entry and exit of the heat exchanger are the following:

- (1) At the leading edge, the plate temperature is equal to the air temperature. This temperature is not the upstream inlet temperature of the air; due to stagnation effects at the leading edge of the fins, the air is preheated or pre-cooled before actually passing the $x_0 = 0$ line.
- (2) At the trailing edge, it is assumed that there is no longitudinal temperature gradient (no heat conduction) in the fin. The exit air temperature is given by an equation of the type 4-17.

The above-stated conditions lead to a system of twelve equations in twelve unknowns for each side of the heat exchanger. Referring to figure 17, the equations for the top plate (condenser section) are as follows :

Letting $\phi = \frac{\phi}{kwe}$

entry $\left[\begin{array}{l} T_0^0 = T_{a0}^0 \end{array} \right. \quad (4.20)$

$\left[\begin{array}{l} T_{a0}^0 = T_{a\infty} + \frac{1}{\alpha} \left[\frac{dT}{dX} \right]_0^0 \end{array} \right. \quad (4.21)$

row 1 $\left[\begin{array}{l} \phi_1 = \left[\frac{dT}{dX} \right]_{X_0}^0 - \left[\frac{dT}{dX} \right]_0^1 \end{array} \right. \quad (4.22)$

$\left[\begin{array}{l} T_{a0}^1 = T_a^0 \end{array} \right. \quad (4.23)$

row 2 $\left[\begin{array}{l} \phi_2 = \left[\frac{dT}{dX} \right]_{X_1}^1 - \left[\frac{dT}{dX} \right]_0^2 \end{array} \right. \quad (4.24)$

$\left[\begin{array}{l} T_{a0}^2 = T_a^1 \end{array} \right. \quad (4.25)$

row 3 $\left[\begin{array}{l} \phi_3 = \left[\frac{dT}{dX} \right]_{X_2}^2 - \left[\frac{dT}{dX} \right]_0^3 \end{array} \right. \quad (4.26)$

$\left[\begin{array}{l} T_{a0}^3 = T_{a1}^2 \end{array} \right. \quad (4.27)$

row 4 $\left[\begin{array}{l} \phi_4 = \left[\frac{dT}{dX} \right]_{X_3}^3 - \left[\frac{dT}{dX} \right]_0^4 \end{array} \right. \quad (4.28)$

$\left[\begin{array}{l} T_{a0}^4 = T_a^3 \end{array} \right. \quad (4.29)$

exit $\left[\begin{array}{l} \left[\frac{dT}{dX} \right]_{X_4}^4 = 0 \end{array} \right. \quad (4.30)$

$\left[\begin{array}{l} T_{a1}^4 = \left[v_1 T_1 + v_2 T_0 - (v_1 + v_2 - 1) T_{a0} \right]_{i=4} \end{array} \right. \quad (4.31)$

A similar set of equations is obtained for the lower plate. To solve this system of equations a matrix is composed of the coefficients γ , μ , and ν given in section 4.9. The matrix equation takes the form shown in figure 19, non-zero elements indicated by squares and is given in tensorial notation by

$$\underline{A} \cdot \underline{x} = \underline{B} T_{a0}^{\circ} + \underline{C} \alpha$$

We have, then, a system of equations in which the only appearing unknowns, the vector \underline{x} , are the air and fin temperatures at each discontinuity in the plate, precisely the values needed to couple, by means of a mathematical model of the heat pipes (to be described in the next section), the two sides of the heat exchangers.

4.3 Numerical simulation of the heat pipes

Ideally, it would have been desirable to develop a complete numerical model of the variable conductance heat pipe, capable of representing its operation in all possible regimes, thereby permitting a complete comparison with the experimental results. Due to the complexity of the internal heat pipe phenomena, however, such a model was beyond the scope of this preliminary study. Instead, we limited ourselves to a model of the heat pipe in its adiabatic reservoir regime, i.e., the point at which heat is exchanged only between the evaporator and condenser sections. This approach is justified, moreover, by the fact that we have been until now unable to obtain much of the information on the physical parameters of the heat pipe construction (eg. wick thickness, wick porosity, effective pore radius, wick material, etc) necessary to an accurate prediction of its performance. Thus, even restricting ourselves to the adiabatic operation of the heat pipes, we could only hope to achieve qualitative agreement with the experimental results by making appropriate guesses at the unknown parameters.

Our model is based on the simultaneous resolution of the equations expressing the relevant thermodynamic and hydrodynamic phenomena in the heat pipe. To attack the problem, it was first necessary to have a geometrical description of the liquid layer in the wick. On the basis of a survey of the longitudinal temperature distributions we measured under adiabatic operating conditions, a number of different idealized liquid profiles seemed plausible (see Fig. 20). Number 3 is the only one compatible with the assumption of constant heat flux over the entire evaporator section and over the entire condenser section, a condition necessary to the incorporation of this model into the finned tube model described in section 4.1. What follows is the development of heat and mass transfer equations for the model based on profile

4.3.1 Primary and secondary unknowns

These equations are formulated in terms of three primary unknowns and three secondary unknowns. The primary unknowns are

- x_1 thickness of the liquid layer in the condenser wick
- x_2 thickness of the liquid layer in the evaporator wick
- x_3 working fluid mass flow rate through the adiabatic zone, $(\dot{m})_{\max}$

The secondary unknowns are the temperatures on which the various fluid properties such as density and viscosity are dependent.

T_v vapour temperature

$\left(T_2\right)_e$ average temperature of the liquid in the evaporator

$\left(T_2\right)_c$ average temperature of the liquid in the condenser

4.3.2 Thermal resistance network

The heat transferred by the heat pipe must traverse a network of thermal resistances, which is represented in figure 21.

- T_e temperature applied to the external surface of the evaporator
- T_c temperature applied to the external surface of the condenser
- T_v temperature of vapour in core of heat pipe (operating temperature)
- RP_e thermal resistance across the heat pipe wall in the evaporator section
- RW_e thermal resistance to conduction across the liquid layer in the evaporator section
- RP_c thermal resistance across the heat pipe wall in the condenser section
- R_c thermal resistance to axial conduction in the wall of the heat pipe

For a heat pipe in which there is not significant dry out the axial conduction in the walls is negligible with respect to the heat transfer by vapour transport, so R_c is neglected. Thus

$$Q = \dot{m}L = \frac{T_e - T_c}{RP_e + RW_e + RP_c + RW_c} \quad (4.32)$$

where the four resistances are given by

$$RP_e = \frac{\ln(R_o/R_p)}{2\pi k_p l_e}$$

$$RP_c = \frac{\ln(R_0/R_p)}{2\pi k_p \ell_c}$$

$$RW_e = \frac{\ln\left[R_p/(R_p - X_2)\right]}{2\pi k_w \ell_e}$$

$$RW_c = \frac{\ln\left[R_p/(R_p - X_1)\right]}{2\pi k_w \ell_c}$$

4.3.3 Pressure drop equilibrium in the heat pipe

Heat transfer in a heat pipe involves the working fluid in a cyclical process in which the fluid is pumped around a closed loop against certain pressure losses. During steady state operation, there must exist an equilibrium between the pumping forces and the drag forces on the working fluid. The pressure losses undergone by the fluid arise from the viscous resistance to the flow in both the wick and the vapour core, and from the momentum loss (or gain) of the vapour due to the mass transfer at the liquid/vapour interface.

The pumping forces are the capillary force on the liquid, which is due to the variation in the radius of curvature of the liquid menisci along the wick, and the body forces on the fluid, which are limited to the force of gravity in the present case. The weight of the vapour is very small compared to the weight of the liquid and is therefore neglected.

This equilibrium is expressed by the following equation, where pumping pressure gradients are assumed to be positive in the flow direction, and drag pressure gradients negative :

$$\Delta p_c + \Delta p_g + \Delta p_\ell + \Delta p_v = 0 \quad (4.33)$$

where Δp_c pressure difference due to capillary forces

Δp_g pressure difference due to gravity

Δp_ℓ pressure drop in liquid

Δp_v pressure drop in vapour

This equation can be written in integral form, with z defined as the direction of liquid flow :

$$\int_{z_1}^{z_2} \left[\frac{dp_c}{dz} + \frac{dp_g}{dz} + \frac{dp_\ell}{dz} - \frac{dp_v}{dz} \right] dz = 0 \quad (4.34)$$

4.3.4 Pumping forces

Capillary liquid pressure gradient

In the case of a class B wick (in which the reduced capillary force is independent of the wetting angle of the liquid/solid interface, but is rather a function of pore geometry), the maximum capillary pressure gradient is given by

$$\frac{dp_c}{dz} = \frac{2\sigma}{r_p(\ell_e + \ell_a + \ell_c)}$$

where σ surface tension of liquid/vapour interface

r_p effective pore radius in the wick

Gravitational liquid pressure gradient

This term is simply the density of the liquid times the acceleration due to gravity :

$$\frac{dp_g}{dz} = \rho_\ell g$$

4.3.5 Drag forces

Viscous liquid pressure gradient

This term is given by a form of Darcy's law, adapted for flow in porous media. It is usually used for fully saturated wicks, with A denoting the full cross-sectional area of the wick. In this case, however, A represents only the wetted area and depends on the thickness of the liquid layer in the wick. Cotter (Ref. 4) gives the following expression for the liquid pressure gradient :

$$\frac{dp_l}{dz} = \frac{-v_l \dot{m}_l(z)}{K A(z)}$$

where \dot{m}_l local liquid mass flow rate (m/s)
 K wick permeability (m^2)
 A local liquid flow area (m^2)

Vapour pressure gradient

The vapour pressure gradient can be divided into two distinct terms, one due to viscous shear and the other due to mass transfer, indicated by

$$\left(\frac{dp_v}{dz}\right)_f \quad \text{and} \quad \left(\frac{dp_v}{dz}\right)_m$$

respectively. The expression for the viscous pressure loss is similar to that for the liquid. The momentum pressure gradient depends on the rate of mass addition (or removal) at the wick boundary and thus contains the derivative of the mass transfer rate with respect to position. Katsoff's (Ref. 5) equations for each of these terms are

$$\left(\frac{dp_v}{dz}\right)_f = \frac{8 \nu_v \dot{m}(z)}{\pi R_v^4}$$

$$\left(\frac{dp_v}{dz}\right)_m = \frac{8 \dot{m}(z)}{3 \pi^2 \rho_v R_v^4} \frac{d\dot{m}(z)}{dz}$$

where R_v radius of vapour core
 ν_v kinematic viscosity of vapour
 ρ_v density of vapour

4.3.6 The heat pipe equations

The liquid profile in the wick chosen for this model imposes certain boundary conditions on the pressure balance equations. Since the thickness of the liquid layer is constant over the whole condenser section and the whole evaporator section, according to the resistance model the heat transfer rate also must be constant. This implies that in the pressure balance equations, $\frac{d\dot{m}}{dz}$ is constant over each section. Additionally, since there is no mass interchange with the reservoir, the mass flow rate is zero at both ends of the heat pipe.

Expanding equation 4.32, and integrating 4.34 over the length of the evaporator and over the length of the condenser, one obtains the following set of three non-linear algebraic equations in the primary and secondary unknowns introduced in section 4.21 :

$$f_1(x_1, x_2, x_3) = 0 = \left[\frac{4}{3 \pi^2 \ell_c \rho_v (T_v) R_v^4} \right] x_3^2 - \left[\frac{\nu_\ell (T_\ell)_c}{2KA_\ell (x_1)} + \frac{4 \nu_v (T_v)}{\pi R_v^4} \right] x_3 + \left[g_{\rho \ell} (T_\ell)_c + \frac{2 \sigma (T_v)}{r_p (\ell_e + \ell_a + \ell_c)} \right] \quad (4.35)$$

$$f_2(x_1, x_2, x_3) = 0 = \left[\frac{4}{3\pi^2 \ell_e \rho_V (T_V) R_V^4} \right] x_3^2 + \left[\frac{v_\ell (T_\ell)_e}{2KA_\ell (x_2)} + \frac{4v_V (T_V)}{\pi R_V} \right] x_3 - \left[g_{\rho_\ell (T_\ell)_e} + \frac{2\sigma (T_V)}{r_p (\ell_e + \ell_a + \ell_c)} \right] \quad (4.36)$$

$$f_3(x_1, x_2, x_3) = 0 = x_3 L -$$

$$- \frac{T_e - T_c}{\left(\frac{1}{\ell_e} + \frac{1}{\ell_c} \right) \left[\frac{\ln(R_0/R_p)}{2\pi k_p} \right] + \left(\frac{1}{2\pi k_w} \right) \left[\frac{\ln[R_p/(R_p - x_2)]}{\ell_e} + \frac{\ln[R_p/(R_p - x_1)]}{\ell_c} \right]} \quad (4.37)$$

A complementary set of relations, based on the thermal resistance model of figure 21, is used to evaluate the secondary unknowns :

$$T_V = T_e - x_3 L (R_{P_e} + R_{W_e}) \quad (4.38)$$

$$(T_\ell)_e = T_V + \frac{1}{2} x_3 L (R_{W_e}) \quad (4.39)$$

$$(T_\ell)_c = T_V - \frac{1}{2} x_3 L (R_{W_c}) \quad (4.40)$$

4.3.7 Method of solution

In the FORTRAN program, equations 4.35 to 4.37 are solved iteratively using Newton's method. The input to the heat pipe subroutine consists of the known heat pipe properties and the external temperatures of the evaporator and condenser sections. The output consists of the heat flux and heat pipe operating temperature.

A matrix is constructed of the partial derivatives of the three functions f_1 , f_2 and f_3 , defining a system of linear independent equations which are then solved by Gaussian elimination to yield a correction vector containing the increments to be added to each of the primary unknowns in order to improve the accuracy of the initial estimates. After each iteration, the secondary unknowns are reevaluated using equations 4.38 to 4.40, and the new values are used in determining the values of the fluid properties to be used in the next iteration. This procedure is repeated until the desired tolerances on the x values have been met.

The stability of this method seems to be very sensitive to several of the physical heat pipe parameters, notably the permeability and porosity of the wick. A damping factor is currently used to retard convergence and thereby to avoid accidental divergence from an actual solution.

4.4 Coupling the finned tube model to the heat pipe model

The critical step in modelling the entire heat exchanger is the incorporation of the heat pipe model into the previously described solution of the finned tube system. What follows is a brief explanation of the global solution algorithm.

To start, an overall heat exchanger effectiveness of 50% is assumed, and the heat fluxes in all of the heat pipes are assumed equal. This gives initial values for the heat fluxes $\phi_{i=1,4}$ in vector \underline{C} of the system of equations shown in figure 19. The inlet air temperatures are given, so the corresponding set of equations for both sides of the heat exchanger can be solved, thereby giving the air and fin temperatures at each node. These temperatures are computed on the basis of an arbitrarily assumed heat flux which most probably does not correspond to the actual performance of the heat pipes at the specified air inlet conditions. When the heat pipe subroutine is called, however, new values are provided for the heat flux in each row of heat pipes. These new values replace the previous ones in the vector \underline{C} in the parallel systems of equations (evaporator and condenser side), enabling one to recompute the air and fin temperatures at each node. The process is repeated until convergence is obtained.

The output values of particular interest are T_{a1}^4 for each side of the heat exchanger, which correspond to TE(2) and TC(2) in the terminology of the experiments. With these data, the heat exchanger effectiveness is computed to facilitate comparison with the experimental results. The operating temperature of each row of heat pipes, passively determined in the process of convergence, is included in the program's output, as it is of interest in determining the operating regime of the heat exchanger predicted by the program.

4.5 Computational results

Since we had almost no information on the internal construction of the heat pipes, informed guesses had to be made to provide the computer program with some of the necessary input parameters, particularly those concerning the wick. Although perhaps not the real configuration, the wick was assumed to be of single annular geometry with isotropic and uniform

porosity. Several physical properties seem to play a major role in determining the stability of the procedure, notably the porosity (void fraction) and permeability of the wick. The values of these parameters were chosen so as to obtain convergence of the solution; they do lie, however, within the range of physically reasonable values. A summary of the heat exchanger parameters used in the computations appears in Appendix C.

The program HPIPE was run for a variety of air flow conditions matching those of the experiments. A sample of the program output, with only the final iteration shown, appears in Appendix D. The air and fin temperature is given for each node (fin-tube intersection) as well as the vapour temperature (reservoir temperature for each row of heat pipes). A plotting routine was developed for the program, permitting the visualization of the evaporator and condenser section air and fin temperature profiles given by the numerical solution. Figures 22-29 show the temperature profiles computed for the various air inlet temperatures and mass flow ratios tested. It is important to note the difference in nature between the air and fin temperature profiles, whereas the air's temperature changes monotonically as it passes through the heat exchanger, the fin temperature gradient often changes sign at the heat pipe rows, indicating a heat flux either into (evaporator) or out of (condenser) the heat pipes both upstream and downstream of each row. The existence of these cusps in the fin temperature profile depends on the relative magnitude of the equivalent heat pipe thermal conductivity, the fin thermal conductivity, and the coefficient of convection of the air flows.

For the highest thermal load tested, corresponding to an inlet temperature of 65°C, the four effectivenesses computed for the four different mass flow ratios are in fairly good agreement with the experimental curves when plotted versus their respective average normalized reservoir (Fig. 30).

Thus, it seems that despite the large uncertainty associated with many of the heat pipe parameters, the numerical heat exchanger model does approximate the heat exchanger effectiveness in one particular operating regime. There is a major discrepancy, however, in that the reservoir temperature predicted by the program, which is based on the assumption of adiabatic reservoirs, are consistently higher than the experimentally determined adiabatic operating temperature for the same thermal load. This discrepancy can perhaps be corrected with a more detailed knowledge of the real heat pipe parameters. It is possible, however, that the assumption of a constant liquid profile in the wick of the evaporator and condenser sections is an oversimplification, inadequate to simulate the actual heat pipe behaviour, which may include significant dryout zones or non-linear liquid thickness gradients.

To permit the numerical simulation of the heat exchanger over the entire normalized temperature operating range, as well as to improve the prediction of the adiabatic operating point, two modifications of HPIPE are currently being undertaken :

- (1) The heat pipe simulation will be expanded to take into account an arbitrary heat input to the reservoir, hence a vapour flux from the reservoir to the heat pipe. For given air inlet conditions, the operating temperature and effectiveness of the heat exchanger will be functions of the reservoir input energy.
- (2) Rather than treating the evaporator and condenser sections as single entities with constant liquid profiles, each section will be divided into finite elements, with the possibility of a different liquid thickness, heat transfer rate, and temperature boundary conditions for each element.

5. CONCLUSION

The research presented in this report confirms that the use of heat pipes permits the construction of compact and effective air-to-air heat exchangers. The liquid reservoir VCHT can be considered a sort of thermal transistor (evaporator = emitter, condenser = collector, reservoir = base) in which the primary heat flux between the evaporator and condenser is modulated by a secondary heat flux, the energy input to the reservoir. The normalized reservoir temperature, the mass flow ratio, and the effectiveness are three non-dimensionalizations which allow the coherent representation of heat exchanger performance over a wide range of operating conditions. The effectiveness data, when normalized in this way, lie on curves which exhibit two important properties :

- (1) There exists a peak in the effectiveness curves which indicates both the maximum attainable heat flux and the operating temperature at which this maximum is reached.
- (2) Below this temperature, there is a nearly linear variation of effectiveness over a broad temperature band, which would facilitate the design of a control system.

For control purposes, the heat exchanger can be considered either reservoir temperature modulated or reservoir power modulated. The power modulated concept is more direct, since temperature regulation is ultimately achieved by power modulation anyway.

The HPIPE program, once it has been modified to simulate complete variable conductance operation, will be a useful tool in the design of VCHP heat exchangers and in the development of a suitable control system. It appears that the structure of the program is both sound and efficient and that with the aid of more complete design information and a discretization of the heat exchanger along the length of the heat pipes as well as in the flow direction, a high degree of accuracy can be obtained.

REFERENCES

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4. MARCUS, B.D.: Theory and design of variable conductance heat pipes. NASA CR 2018, April 1972, p 8.
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APPENDIX A - HEAT EXCHANGER SPECIFICATIONS

N_p number of heat pipes : 16 (in a square 4 x 4 array)
number of reservoirs : 4 (1 per row of heat pipes)

Materials

housing : aluminum
fins : aluminum
heat pipes : copper
reservoirs : reservoirs
wick : ?
working fluid : methylchloride (Freon 40)

Dimensions

l_e evaporator length : .24 m
 l_c condenser length : .22 m
 l_a length of adiabatic zone : .02 m
 s fin spacing : 1.72 mm
 w width of heat exchanger : .13 m
depth of heat exchanger (flow path length) : .13 m
 d_p diameter of heat pipes : .013 m
heat pipe spacing : .034 m
diameter of reservoirs : .020 m

APPENDIX B - SAMPLE DATA ACQUISITION SYSTEM OUTPUT

DATE: 7/5/81
TITLE: .8/6/85
CONFIGURATION COUNTER=1, PARALLEL=0: 1
CARACTERISTIQUES DES CANNAUX A/D

CANAL	1	THERM	56
CANAL	2	MEVAP	1
CANAL	3	MCOND	1
CANAL	4	NON RACCORDE	
CANAL	5	NON RACCORDE	
CANAL	6	NON RACCORDE	
CANAL	7	NON RACCORDE	
CANAL	8	NON RACCORDE	

0 TEMPS: 0 SEC
TEMP 0-7 + .0 +32.2 +32.0 +72.2 +72.8 +67.2 + .0 + .0
TEMP 8-15 +58.3 +58.8 +58.8 +58.2 +59.2 +59.4 +58.4 +59.9
TEMP 16-23 +52.3 +53.1 +52.8 +52.3 +51.3 +50.7 +50.7 +50.6
TEMP 24-31 + .0 + .0 + .0 + .0 + .0 + .0 + .0 + .0
TEMP 32-39 + .0 + .0 + .0 + .0 + .0 + .0 + .0 + .0
TEMP 40-47 +59.1 +59.1 +60.0 +60.1 +58.9 +61.0 +60.8 + .0
TEMP 48-55 +50.4 +50.8 +49.8 +50.1 +51.6 +51.1 +50.8 + .0

PE: 8.886696 MM H2O MEVAP: .106228883 KG/SEC
PC: 6.44871396 MM H2O MCOND: .0792523501 KG/SEC
TRES: 67.2292988 °C MASS FLOW RATIO: .746852747
NORMALIZED RESERVOIR TEMPERATURE: .86878022
TE(1): 72.528554 TE(2): 59.3699916
TC(1): 32.1439782 TC(2): 51.2921618
HEAT TRANSFER RATE: QE= 1.40900195 KW QC= 1.52967886 KW
POWER RATIO: QE/QC= .921109642
EE: .436740419 EX: .474145963
NORMALIZED RESERVOIR HEAT TRANSFER: QR/QE= .0856470871 QR/QC= .078890357

LONGITUDINAL TEMPERATURE DISTRIBUTION

TL(1)= 60.2520775
TL(2)= 59.7690912
TL(3)= 59.1913034
TL(4)= 58.9864947
TL(5)= 58.6509912
TL(6)= 51.2399422
TL(7)= 50.290892
TL(8)= 50.6329261
TL(9)= 51.5049626
TL(10)= 52.7920862

1 TEMPS: 109 SEC
TEMP 0-7 + .0 +31.9 +31.7 +72.0 +72.3 +66.7 + .0 + .0
TEMP 8-15 +57.9 +58.4 +58.4 +57.7 +58.9 +59.1 +58.0 +59.4
TEMP 16-23 +52.1 +52.7 +52.4 +51.9 +50.8 +50.4 +50.3 +50.5
TEMP 24-31 + .0 + .0 + .0 + .0 + .0 + .0 + .0 + .0
TEMP 32-39 + .0 + .0 + .0 + .0 + .0 + .0 + .0 + .0
TEMP 40-47 +58.7 +58.6 +59.6 +59.8 +58.5 +60.5 +60.4 + .0
TEMP 48-55 +50.1 +50.5 +49.5 +49.8 +51.3 +50.7 +50.6 + .0
PE: 8.203104 MM H2O MEVAP: .100727553 KG/SEC
PC: 6.72312732 MM H2O MCOND: .0816284496 KG/SEC
TRES: 66.7894924 °C MASS FLOW RATIO: .81038849
NORMALIZED RESERVOIR TEMPERATURE: .865862623
TE(1): 72.2025873 TE(2): 58.982272
TC(1): 31.8477284 TC(2): 50.9584425
HEAT TRANSFER RATE: QE= 1.34230321 KW QC= 1.57245778 KW
POWER RATIO: QE/QC= .853633864
EE: .404252502 EX: .473566619
NORMALIZED RESERVOIR HEAT TRANSFER: QR/QE= .17146243 QR/QC= .146366136
LONGITUDINAL TEMPERATURE DISTRIBUTION
TL(1)= 59.8800687
TL(2)= 59.39658
TL(3)= 58.7437604
TL(4)= 58.6133442
TL(5)= 58.2776065
TL(6)= 50.9365081
TL(7)= 49.9868285
TL(8)= 50.3290574
TL(9)= 51.087948
TL(10)= 52.4518702

2 TEMPS: 218 SEC
TEMP 0-7 + .0 +31.9 +31.8 +71.9 +72.5 +66.3 + .0 + .0
TEMP 8-15 +58.1 +58.5 +58.6 +58.0 +59.1 +59.2 +58.2 +59.6
TEMP 16-23 +52.2 +52.9 +52.7 +52.0 +51.1 +50.6 +50.4 +50.7
TEMP 24-31 + .0 + .0 + .0 + .0 + .0 + .0 + .0 + .0
TEMP 32-39 + .0 + .0 + .0 + .0 + .0 + .0 + .0 + .0
TEMP 40-47 +58.9 +59.0 +59.9 +60.0 +58.9 +60.6 +60.5 + .0
TEMP 48-55 +50.3 +50.7 +49.8 +50.1 +51.5 +51.1 +50.7 + .0
PE: 8.6132592 MM H2O MEVAP: .10405466 KG/SEC
PC: 6.72312732 MM H2O MCOND: .0816284496 KG/SEC
TRES: 66.3492644 °C MASS FLOW RATIO: .784476633
NORMALIZED RESERVOIR TEMPERATURE: .853590829
TE(1): 72.2568525 TE(2): 59.1911218
TC(1): 31.9870019 TC(2): 51.1709153
HEAT TRANSFER RATE: QE= 1.37042657 KW QC= 1.58506325 KW
POWER RATIO: QE/QC= .864587943
EE: .412773454 EX: .477422173
NORMALIZED RESERVOIR HEAT TRANSFER: QR/QE= .15662034 QR/QC= .135412058
LONGITUDINAL TEMPERATURE DISTRIBUTION
TL(1)= 60.0663381
TL(2)= 59.6573855
TL(3)= 58.967586
TL(4)= 58.7999645
TL(5)= 58.464335
TL(6)= 51.1641022
TL(7)= 50.2529397
TL(8)= 50.5189858
TL(9)= 51.2776361
TL(10)= 52.6409128

APPENDIX C - VALUES OF PARAMETERS USED IN THE
HPIPE CALCULATIONS HEAT EXCHANGER

Heat exchanger

length of evaporator : .24 m
length of condenser : .22 m
length of adiabatic zone : .02 m

Pipes

external heat pipe diameter : .013 m
internal heat pipe diameter : .011 m
vapour core diameter : .009 m

Wick

wick thickness : .001 m
porosity 15%
permeability $1. \times 10^{-9} \text{ m}^2$
thermal conductivity of wick material : 100 W/m°C
effective pore radius : .0005 m

Fins

fin spacing .00172 m
fin thickness .0005 m
thermal conductivity 384 W/m°C
length of platelets
 x_1 .014 m
 x_2, x_3, x_4 .034 m
 x_5 .014 m

Working fluid

liquid density :

$$\rho_l(T_l) = -(1.818 \times 10^{-4}) T_l^3 + (1.125 \times 10^{-2}) T_l^2 - (2.128) T + 960 \quad (\text{kg/m}^3)$$

$$[T_l] = ^\circ\text{C}$$

vapour density :

$$\rho_v(T_v) = (1.441 \times 10^{-4}) T_v^3 - (5.762 \times 10^{-3}) T_v^2 + (.2965) T_v + 6.066 \quad (\text{kg/m}^3)$$

$$[T_v] = ^\circ\text{C}$$

liquid viscosity :

$$\mu_l = .6931 T_l^{1.5} \exp \left(\frac{331.9}{T_l} \right) \quad \text{Ns/m}^2$$

$$[T_l] = ^\circ\text{K}$$

vapour viscosity :

$$\mu_v = (15.92 \times 10^{-7}) T_v^{0.5} / (1 + 462.1/T_v) \quad \text{Ns/m}^2$$

$$[T_l] = ^\circ\text{K}$$

latent heat of vaporization :

$$4.2 \times 10^5 \quad \text{J/kg}$$

thermal conductivity of liquid :

$$.07 \quad \text{W/m}^\circ\text{C}$$


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*****
*****
AIR      #      TI      EXP(W1*X)  EXP(W2*X)  EXP(-X)
*****
1      .3000E+02  .1061E+01  .5347E+00  -.2842E-13
2      .3308E+02  .3295E+00  -.9278E+00  -.4619E-13
3      .3611E+02  .3250E+00  -.1036E+01  -.1421E-13
4      .3914E+02  .2990E+00  -.1035E+01  -.3055E-12
5      .4219E+02  .9390E-01  -.1045E+01  .2132E-13
*****

```

SOLUTION DE L'EVAPORATEUR
RACINE EN CARACT. W1= .9503E+00 W2=- .1920E+01

```

*****
*****
*****
PLAQUETTE #      TI      EXP(W1*X)  EXP(W2*X)  EXP(-X)
*****
1      .6500E+02  -.2116E+01  .4860E+00
2      .6174E+02  -.6795E+00  -.8141E+00
3      .5891E+02  -.6730E+00  -.9167E+00
4      .5588E+02  -.6170E+00  -.9214E+00
5      .5280E+02  -.1720E+00  -.9367E+00
*****

```

```

*****
*****
*****
AIR      #      TI      EXP(W1*X)  EXP(W2*X)  EXP(-X)
*****
1      .6500E+02  -.1102E+01  -.5281E+00  .1776E-12
2      .6194E+02  -.3538E+00  .8842E+00  -.1208E-12
3      .5891E+02  -.3505E+00  .9961E+00  -.4476E-12
4      .5588E+02  -.3210E+00  .1001E+01  -.3624E-12
5      .5280E+02  -.8958E-01  .1018E+01  .1421E-13
*****

```

EFFICACITE DE L'ECHANGEUR 0087055511017
EQUIPE 5015 0001 11 1100 1040

Mass flow ratio	Heater power	Normalized reservoir temperatures				ϵ_e	ϵ_c
		1	2	3	4		
1.	6 kW	.37	.33	.36	.32	.29	.30
.8	6	.37	.32	.39	.31	.31	.32
.6	6	.37	.32	.39	.30	.37	.36
.4	6	.36	.32	.40	.31	.41	.40
1.	4	.28	.22	.27	.20	.26	.25
.8	4	.32	.26	.32	.26	.30	.30
.6	4	.33	.27	.34	.28	.35	.34
.4	4	.31	.26	.33	.28	.42	.37
1.	2	.23	.12	.24	.14	.22	.21
.8	2	.19	.10	.22	.11	.24	.22
.6	2	.16	.07	.18	.09	.27	.19
.4	2	.16	.06	.17	.12	.35	.24

TABLE 1 - SUMMARY OF TEST RESULTS WITH INDIVIDUALLY HEATED RESERVOIRS

Row	Mass flow ratio	T_n	ϵ_e	ϵ_c
1	.80	.70	.216	.206
2	.80	.70	.218	.224
3	.80	.70	.213	.218
4	.80	.70	.185	.216

Heater Power = 4 kW

TABLE 2 - SUMMARY OF THE TEST RESULT WITH INDIVIDUAL RESERVOIR BATHS

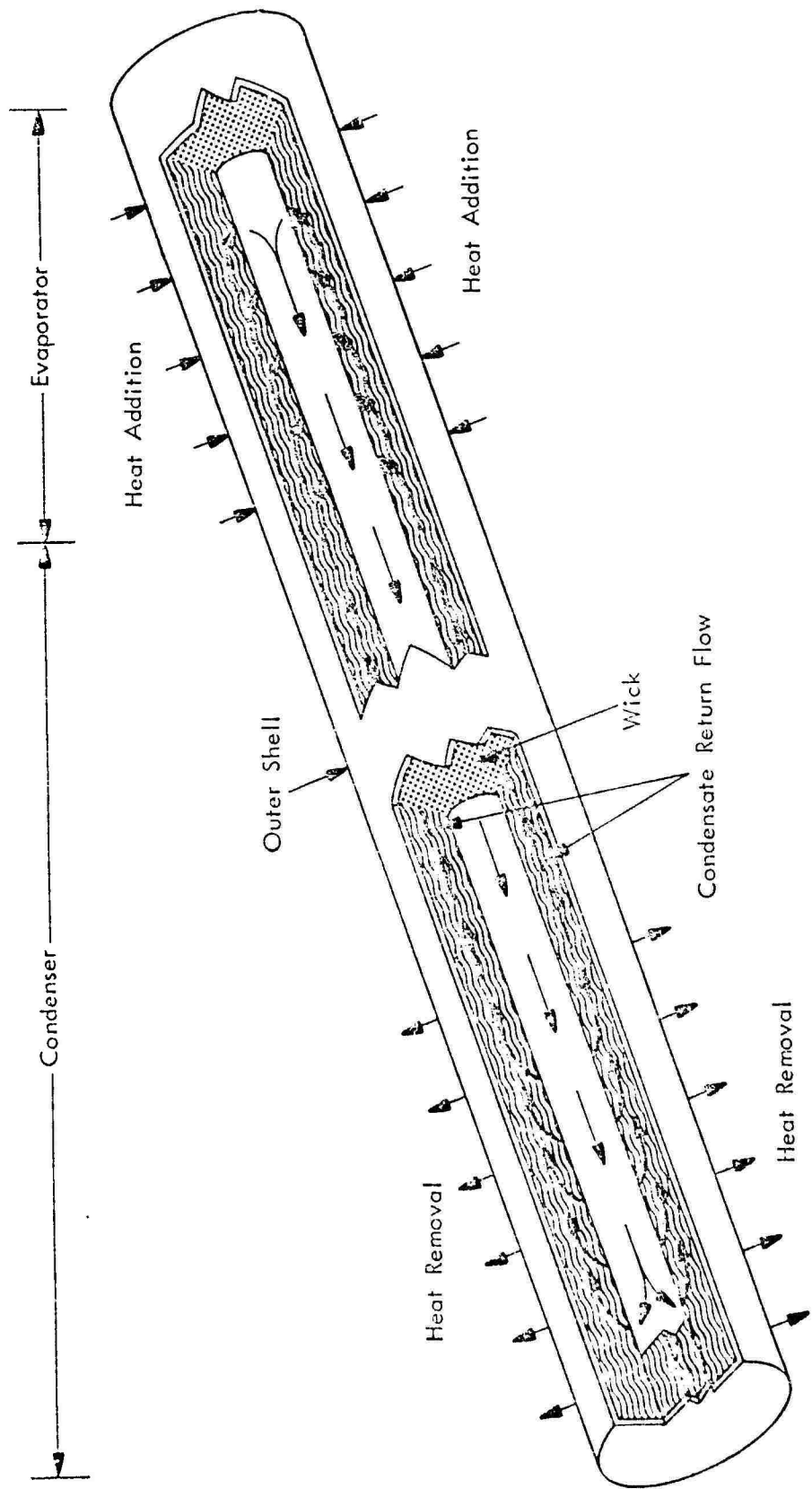


Figure 1 - Parts and Functions of the Basic Heat Pipe

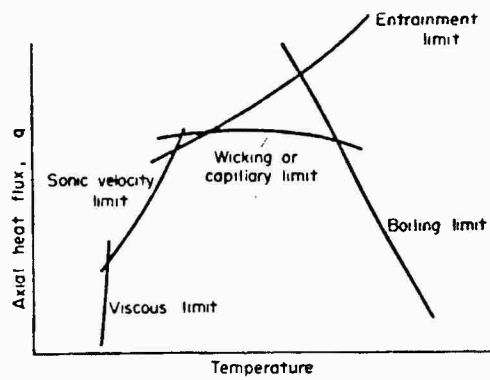


Fig. 2 Limitations to heat transport in the heat pipe

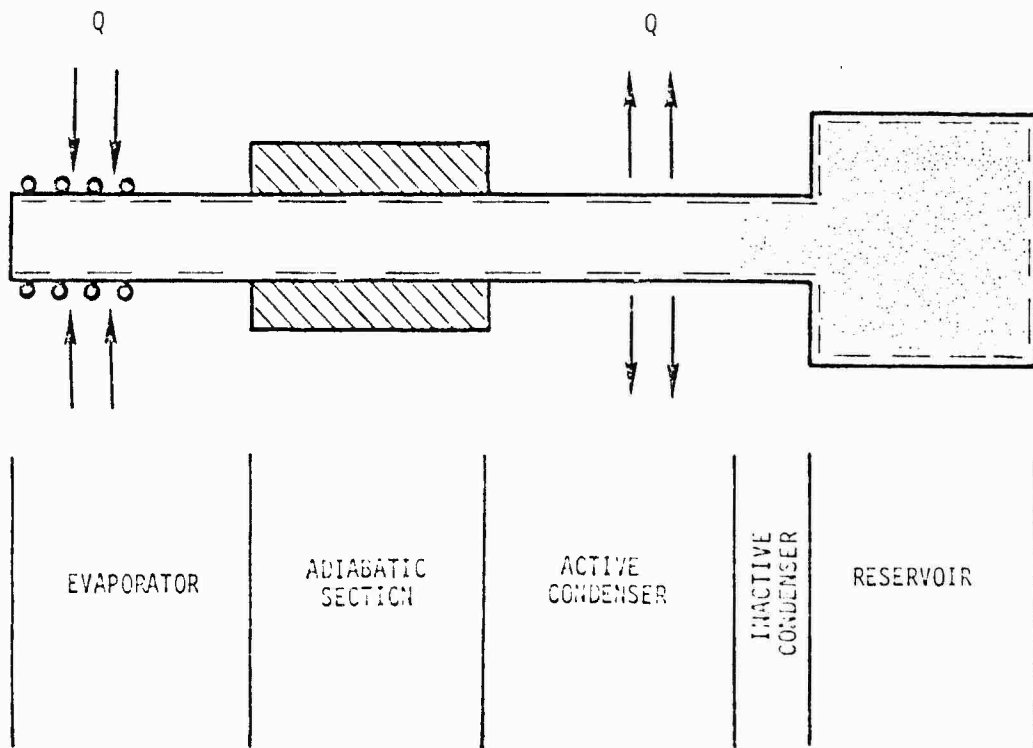


FIG. 3 - NON-CONDENSABLE GAS RESERVOIR VARIABLE CONDUCTANCE HEAT PIPE

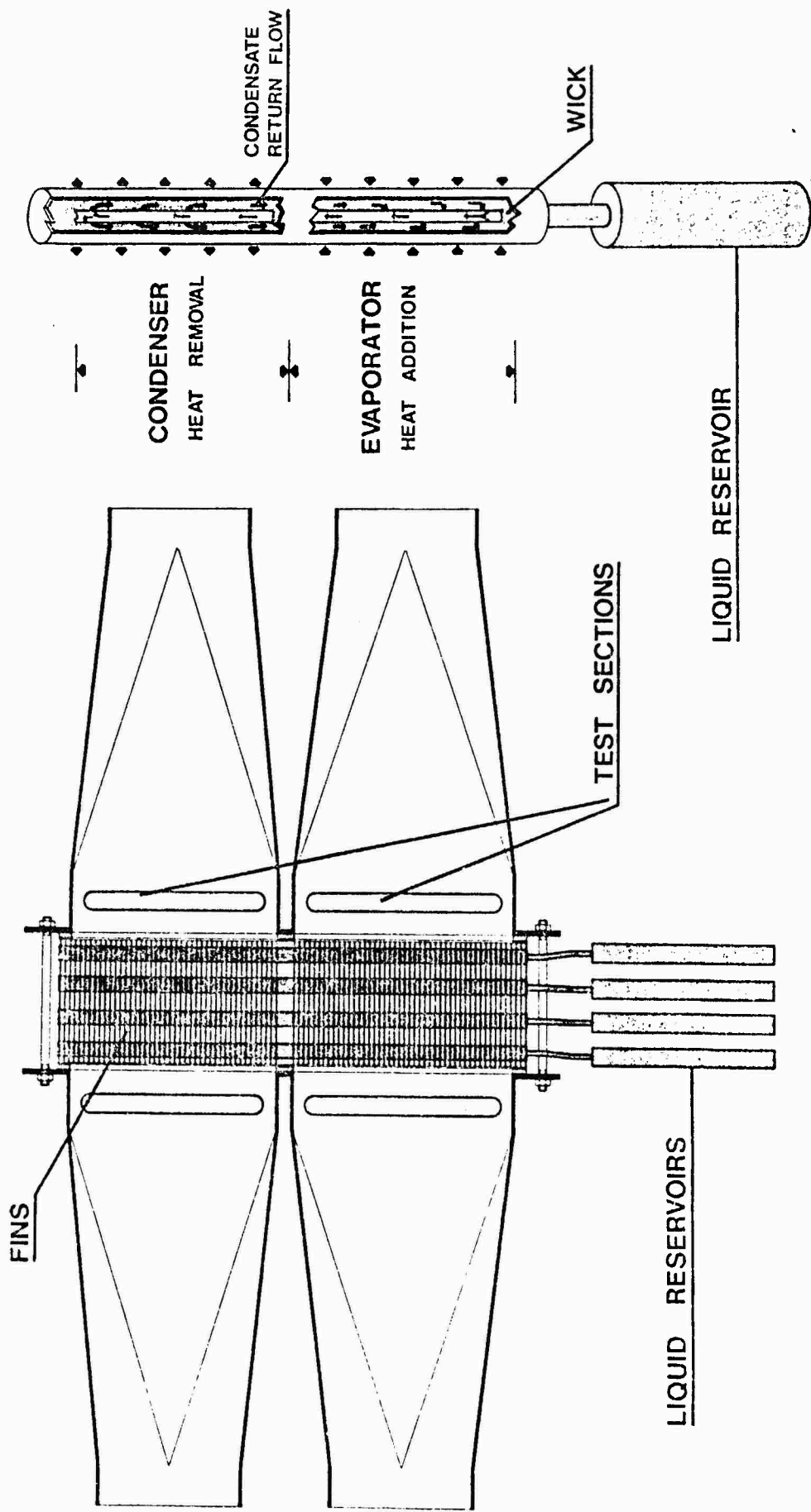


FIG. 4 - Schematic diagram of an air-to-air heat exchanger containing liquid reservoir variable conductance heat pipes.

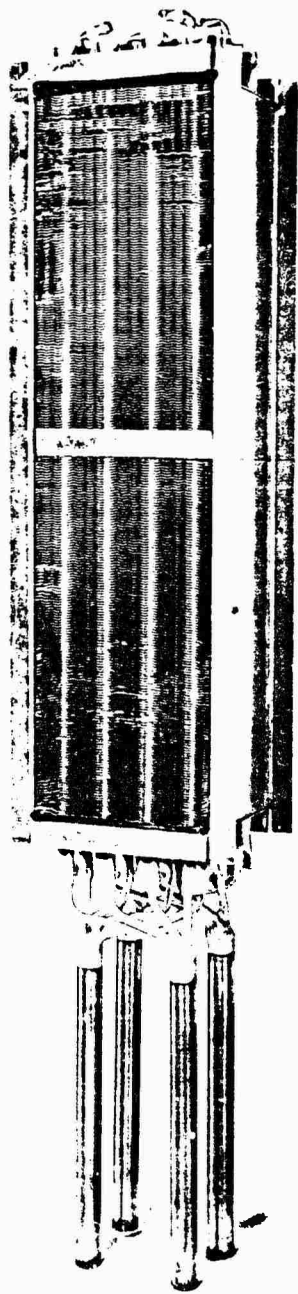


FIG. 5 - THE PROTOTYPE HEAT EXCHANGER

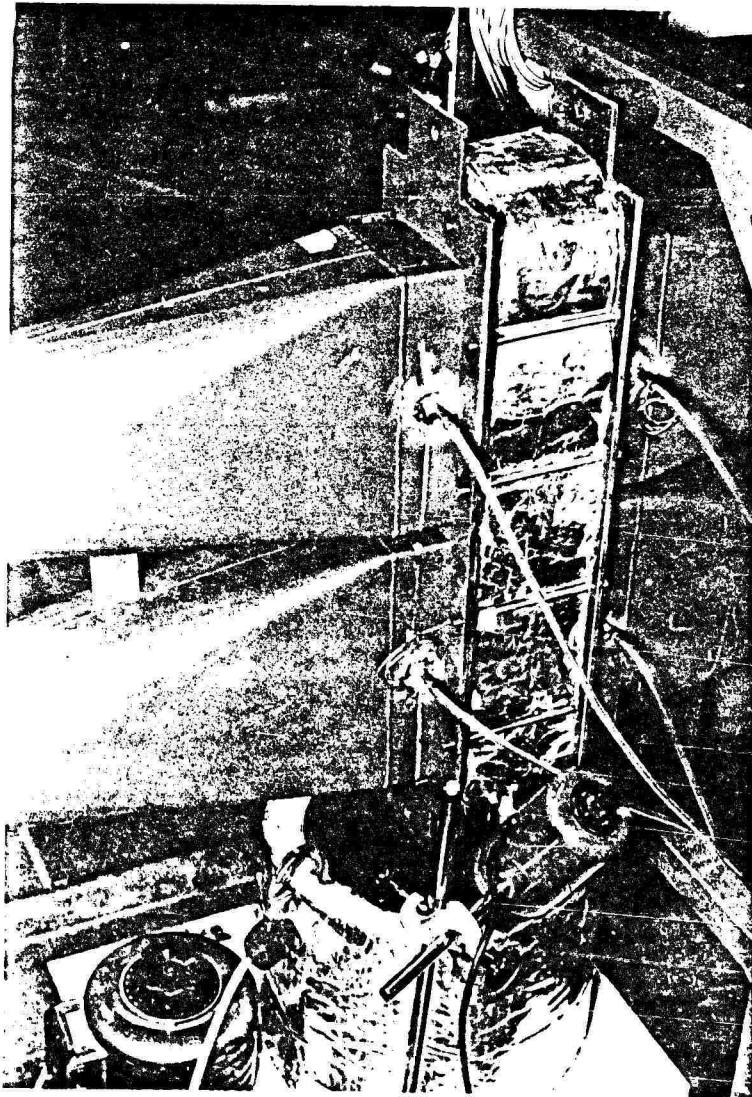


FIG. 7 - REAR VIEW OF HEAT EXCHANGER SHOWING PRESSURE TAPS, OIL BATH AND ELECTRICAL RESISTANCE HEATER

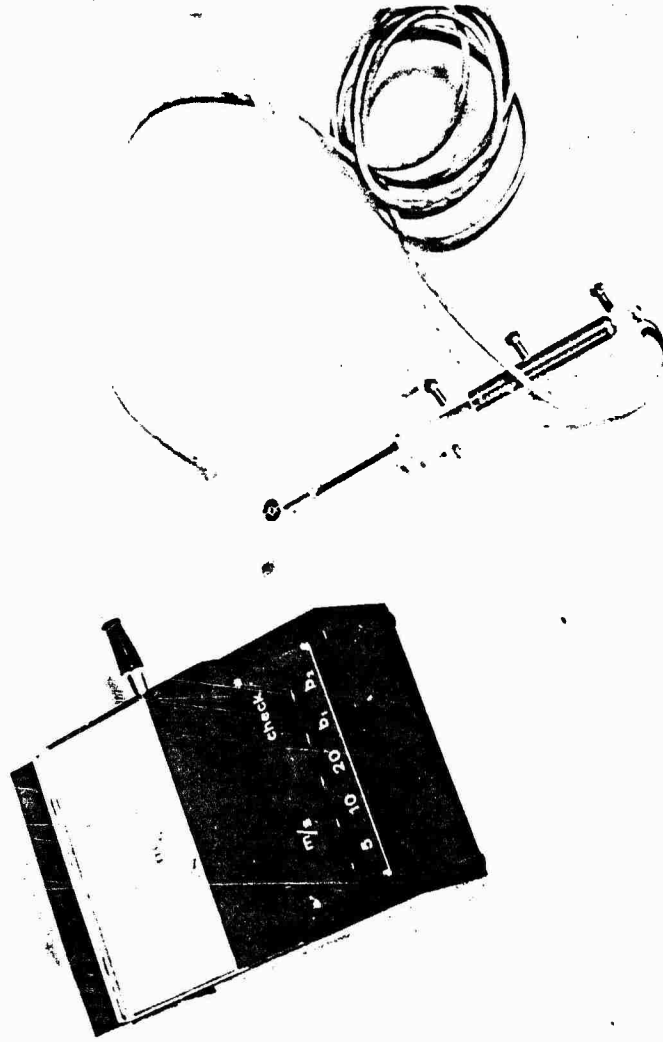


FIG. 8 - VANE ANEMOMETER MOUNTED IN SLIDING SUPPORT

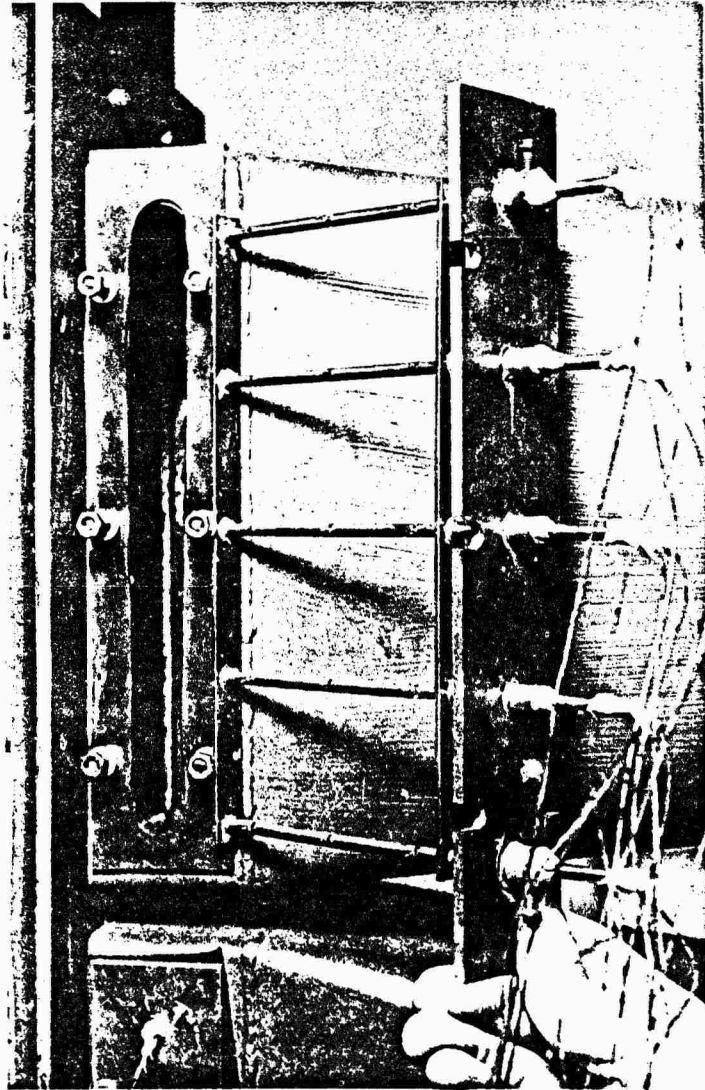


FIG. 9 - THERMOCOUPLE GRID AND TEST SECTION

FIG. 10 - \dot{m} vs. $\sqrt{\Delta P_{HX}}$ FOR THE
EVAPORATOR SECTION

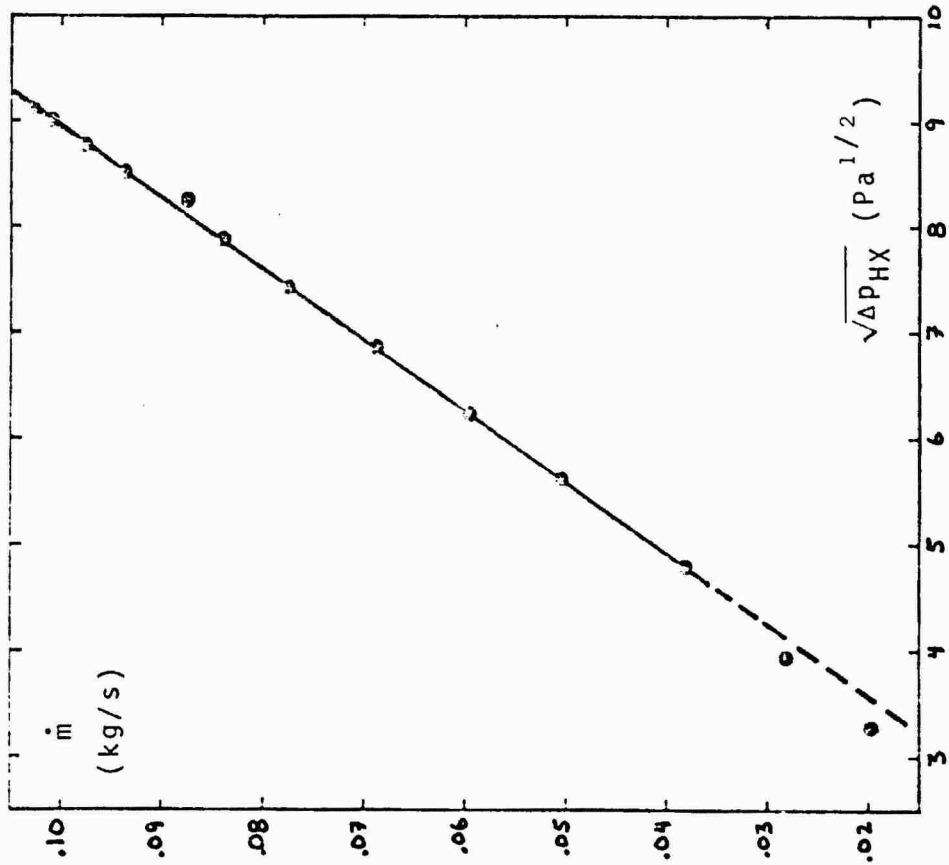
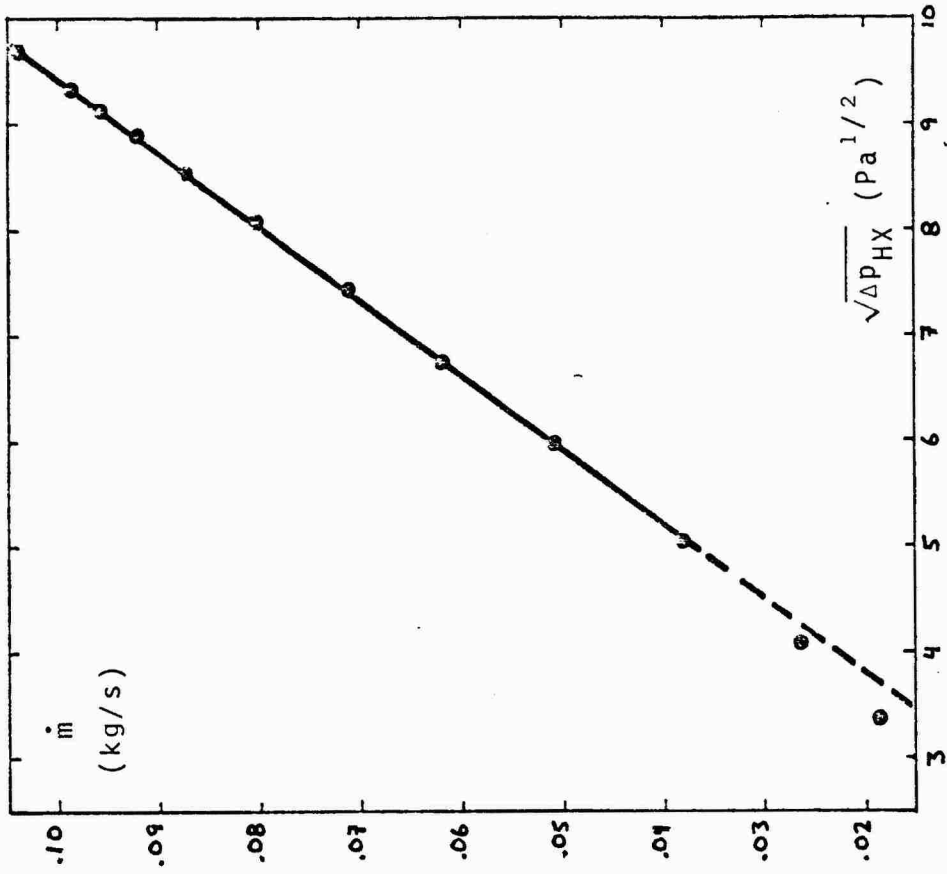


FIG. 11 - \dot{m} vs. $\sqrt{\Delta P_{HX}}$ FOR THE
CONDENSER SECTION



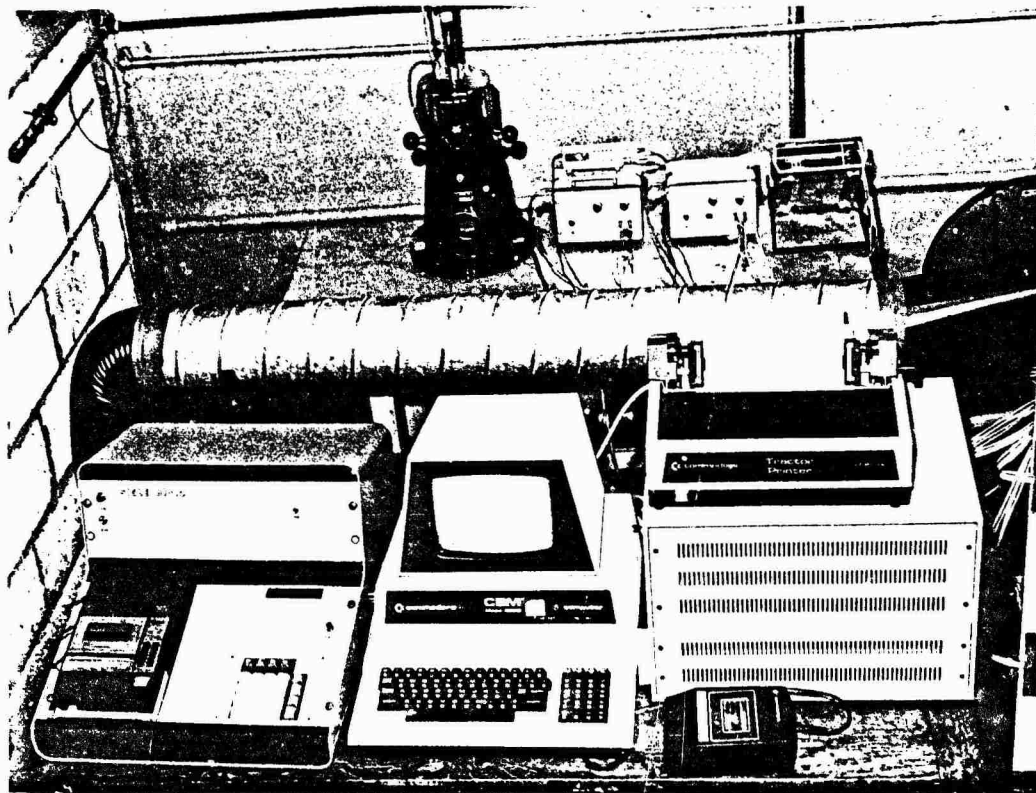


FIG. 12 - CLOSE-UP VIEW OF THE DATA ACQUISITION SYSTEM

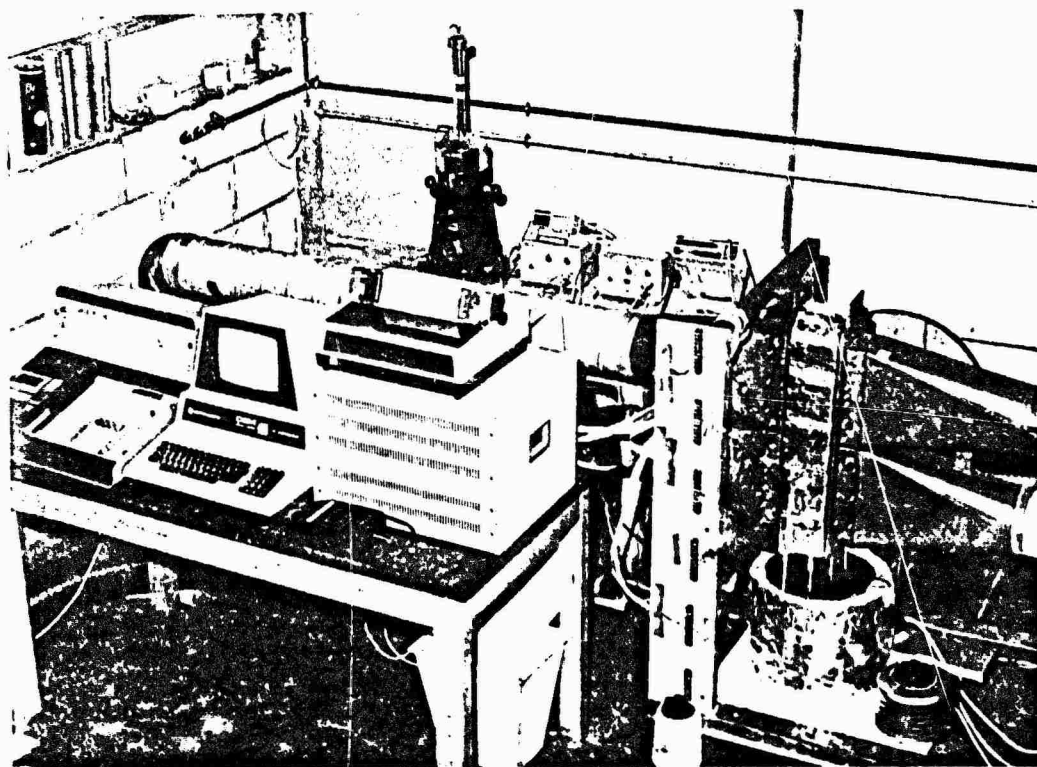


FIG. 13 - VIEW OF THE INSTRUMENTED HEAT EXCHANGER AND THE DATA ACQUISITION SYSTEM

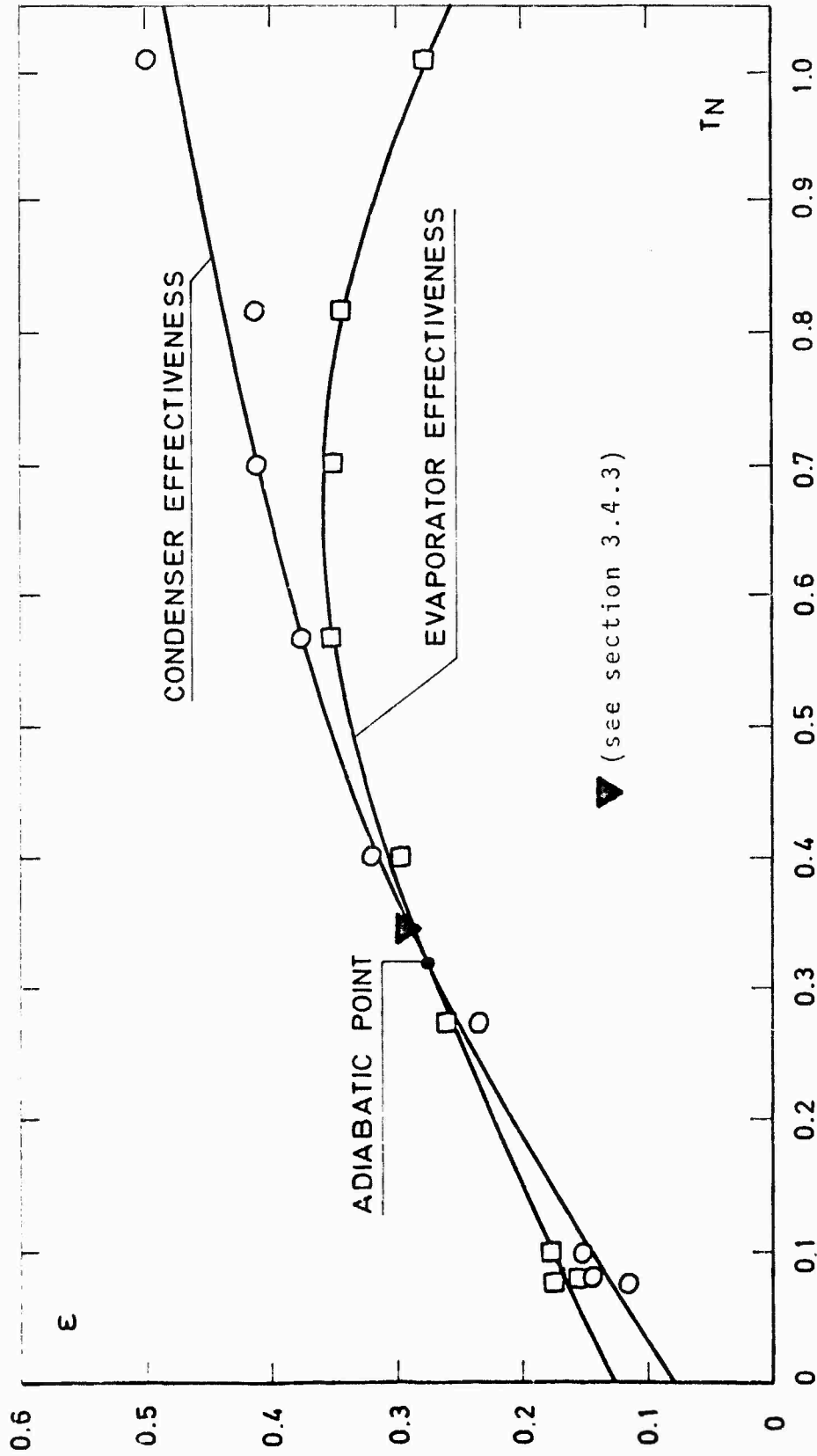


FIG. 14 - EVAPORATOR AND CONDENSER EFFECTIVENESS vs. NORMALIZED RESERVOIR TEMPERATURE FOR MASS FLOW RATIO = 1.0

FIG. 15 - TYPICAL OUTLET AIR TEMPERATURE PROFILE IN THE SUB-ADIABATIC REGIME

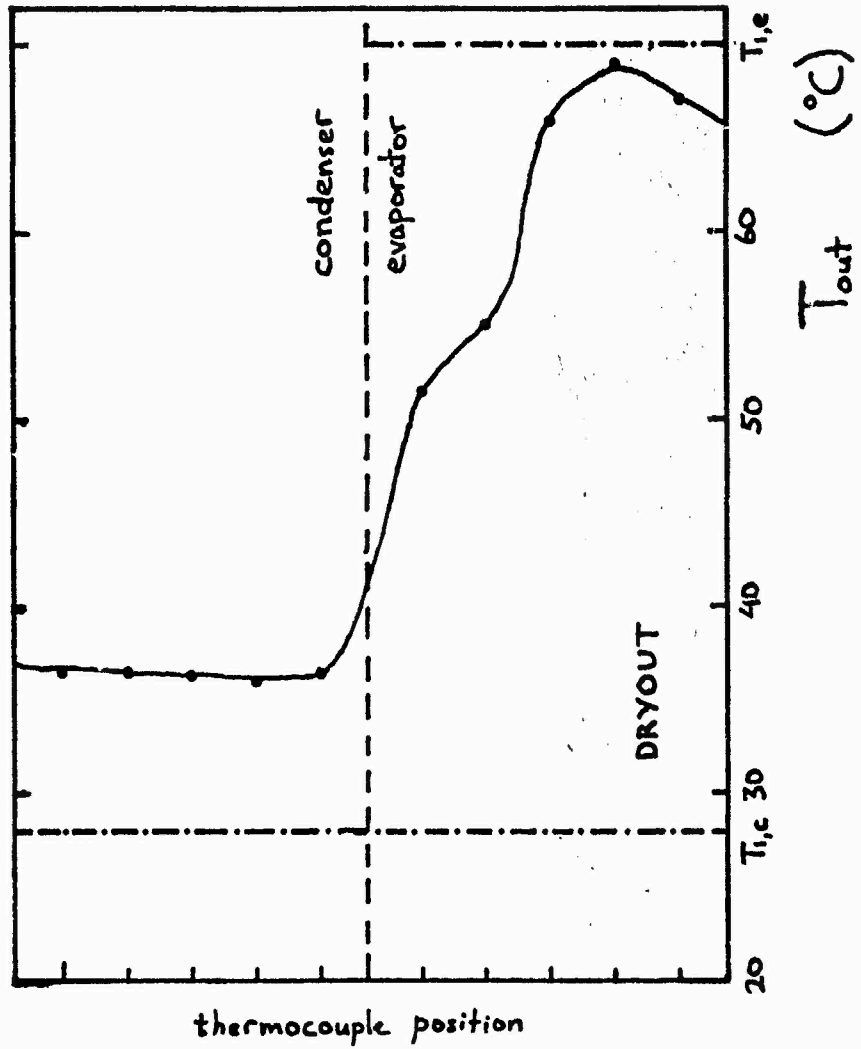
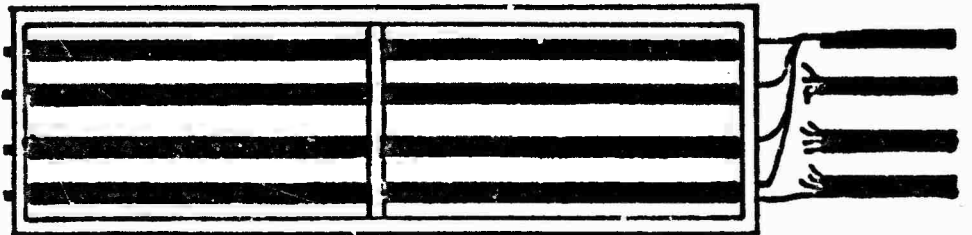
Test Conditions

heater power = 6 kW

MR = 1.0

T_n = 0.19

ϵ_{HX} = 0.19



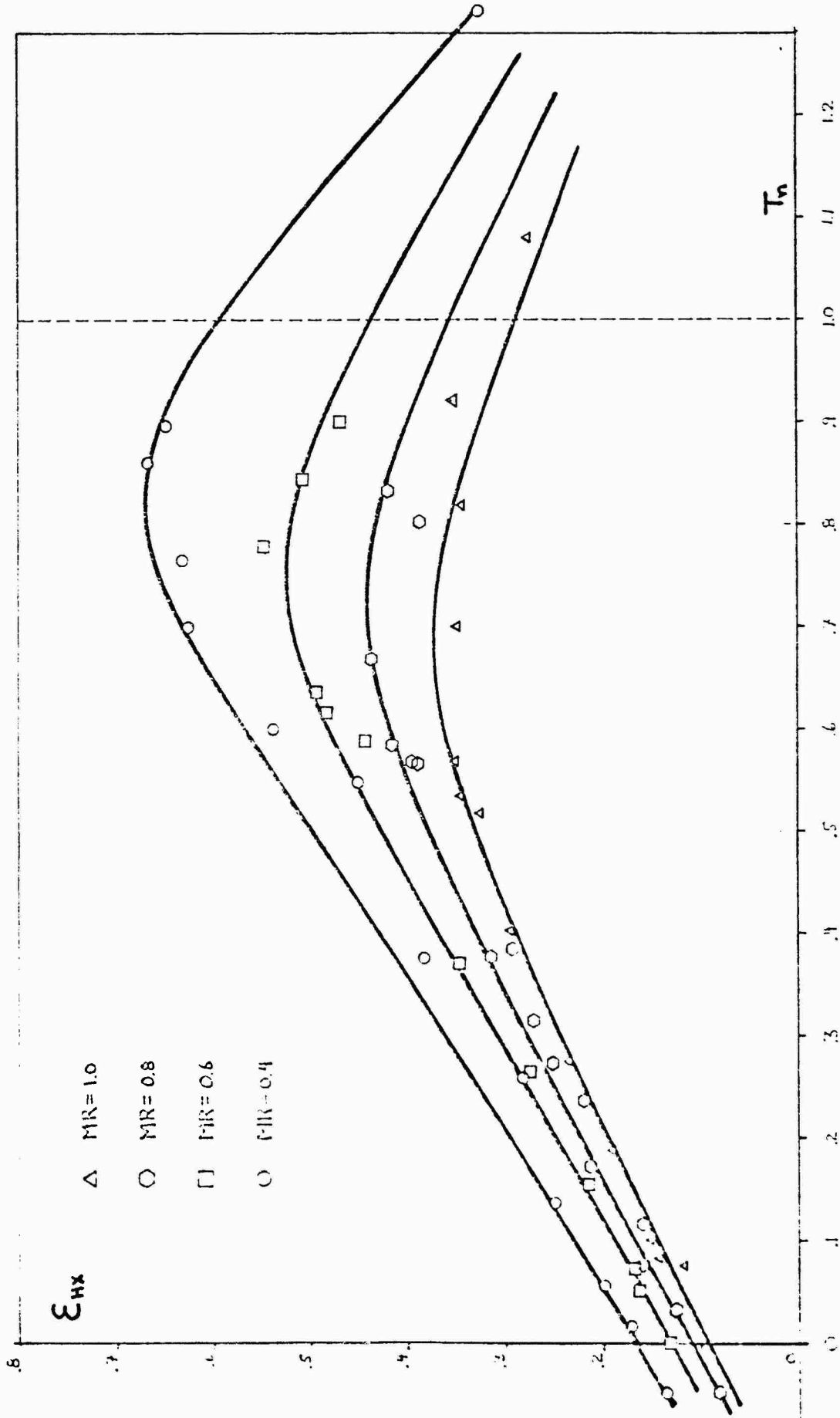
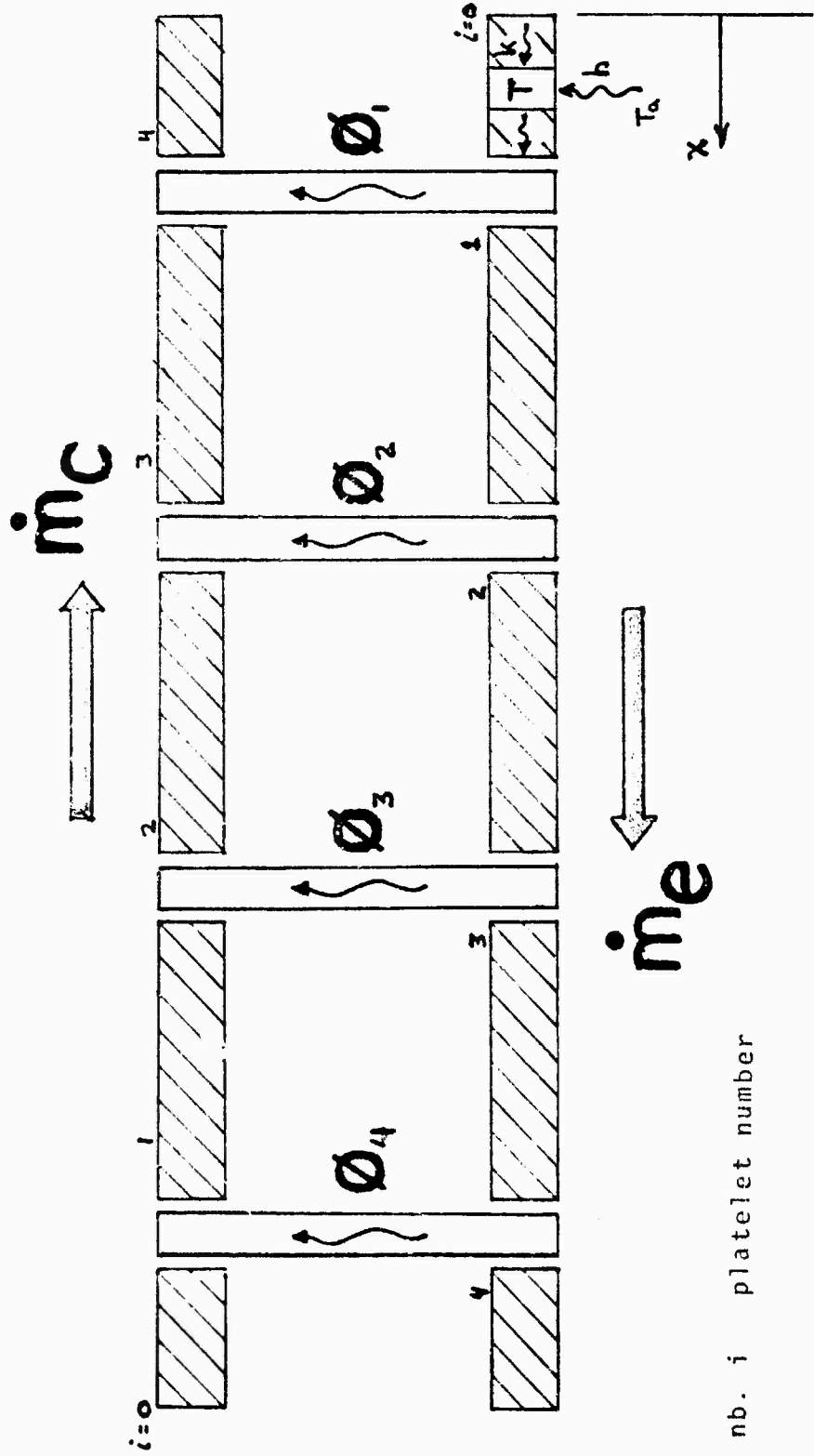


FIG. 16 - TRUE HEAT EXCHANGER EFFECTIVENESS vs. NORMALIZED RESERVOIR TEMPERATURE FOR VARIOUS MASS FLOW RATIOS

FIG. 17 - DISCRETIZATION OF THE HEAT EXCHANGER INTO PLATELETS AND HEAT SOURCE/SINKS



nb. i platelet number

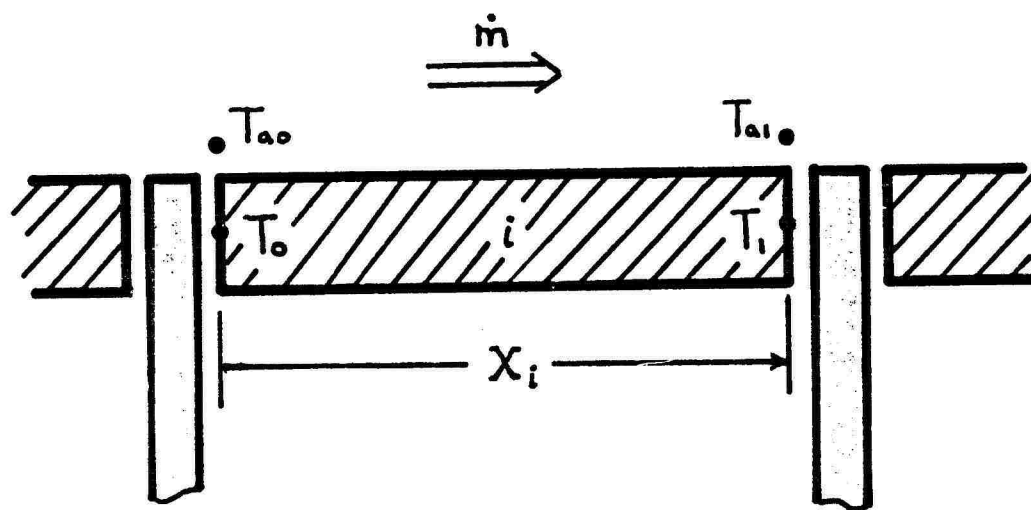
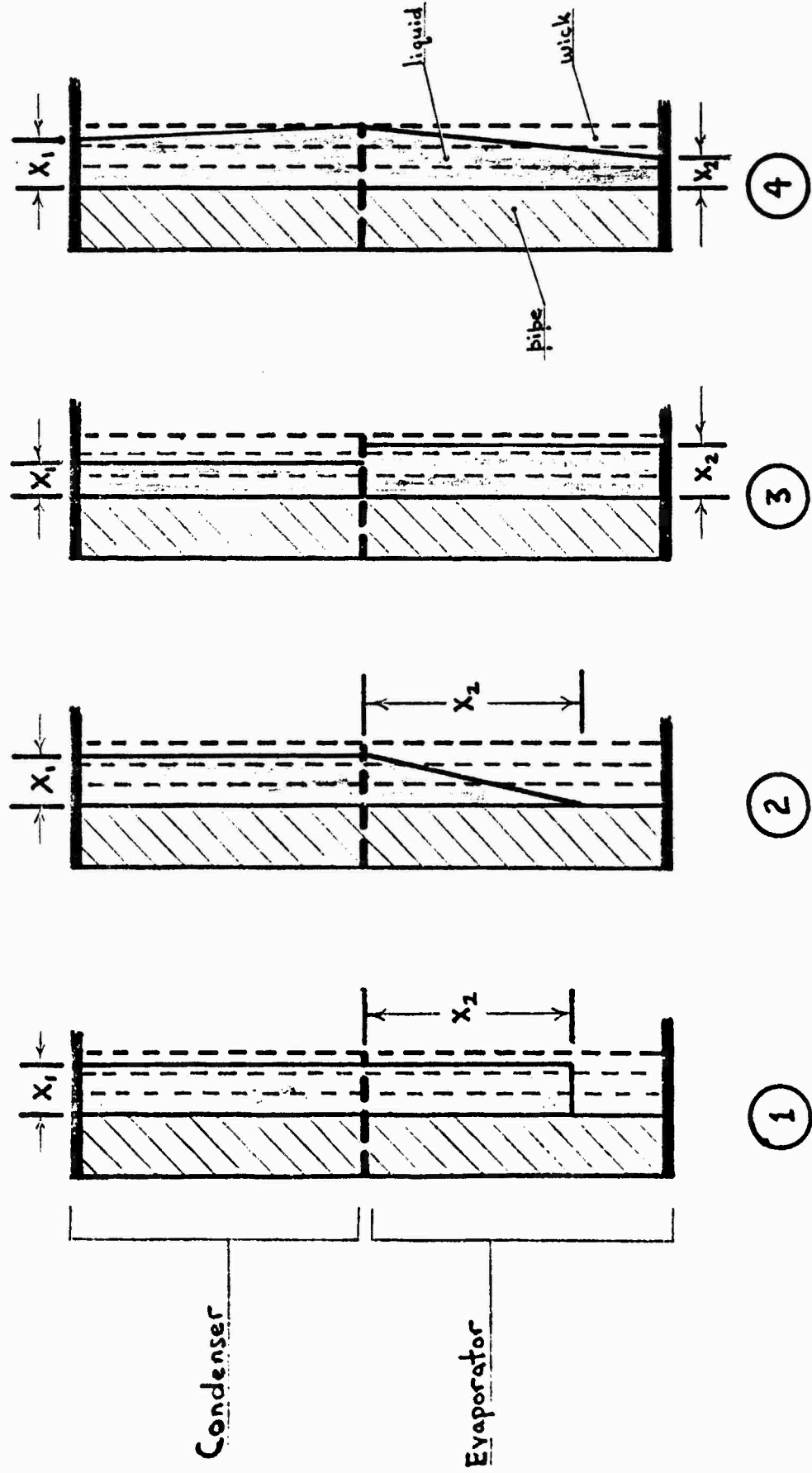


FIG. 18 - NOMENCLATURE FOR A SINGLE PLATELET (§ 4.4)

FIG. 20 - SEVERAL IDEALIZED LIQUID PROFILES IN THE WICK
 (CROSS-SECTIONAL VIEW OF THE HEAT PIPE WALL)



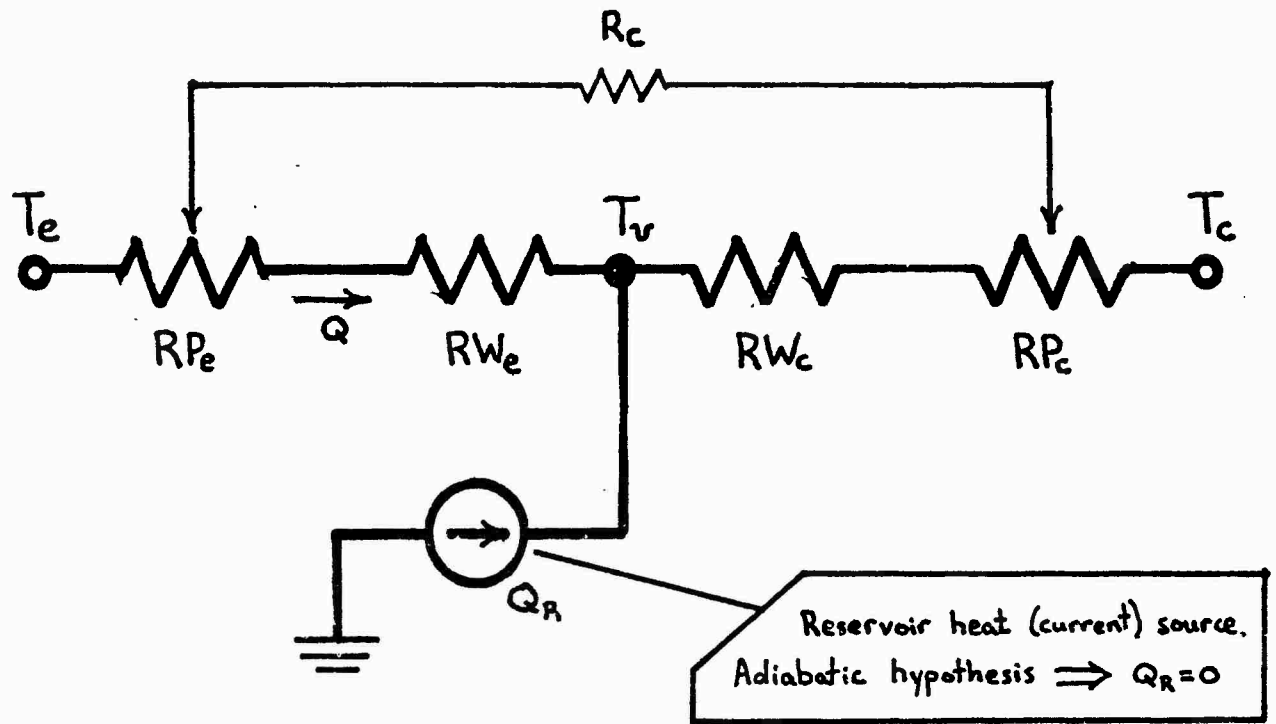


FIG. 21 - THERMAL RESISTANCE NETWORK MODEL OF THE HEAT PIPE
(see section 4.2.2)

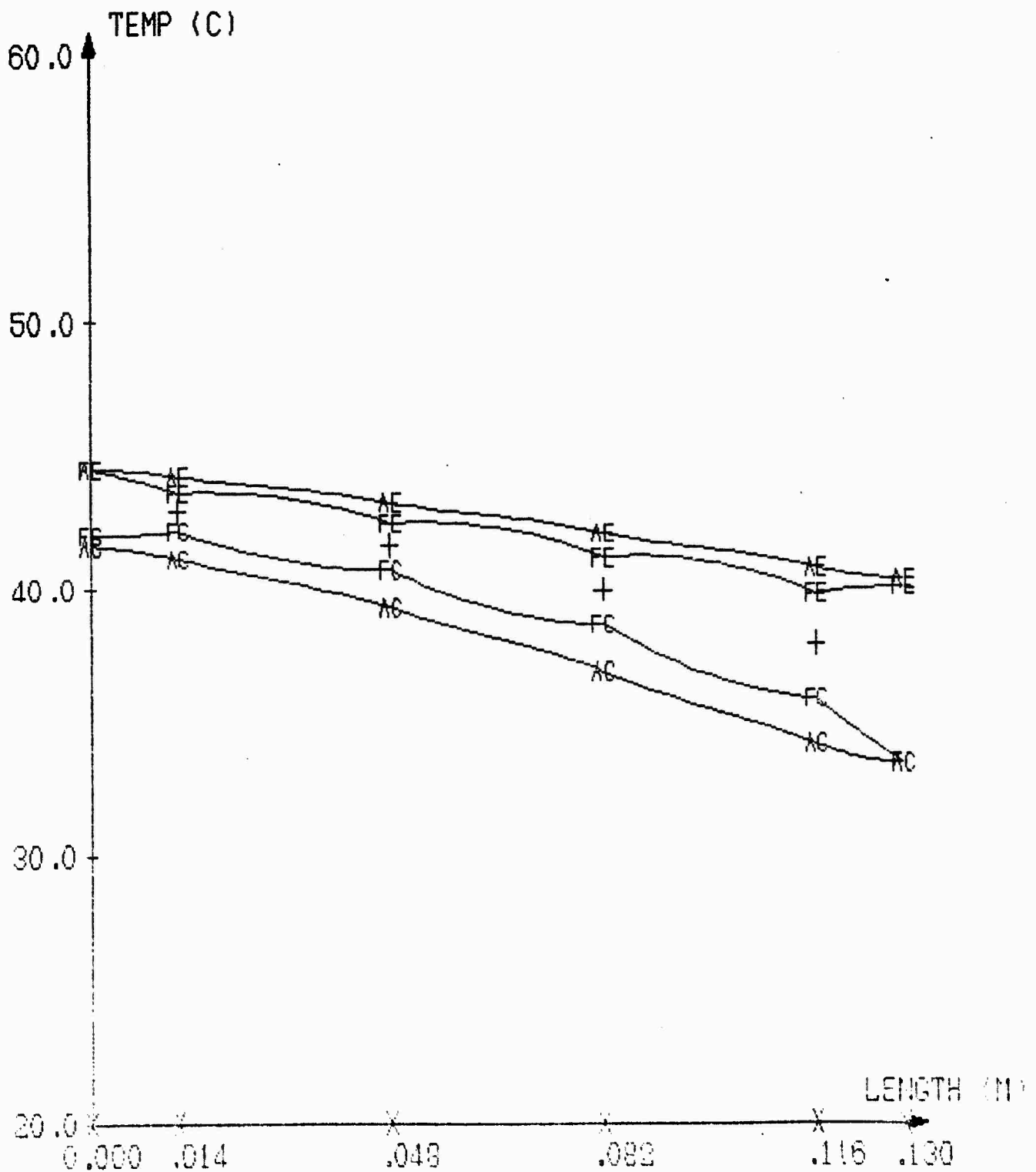


FIG. 22 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

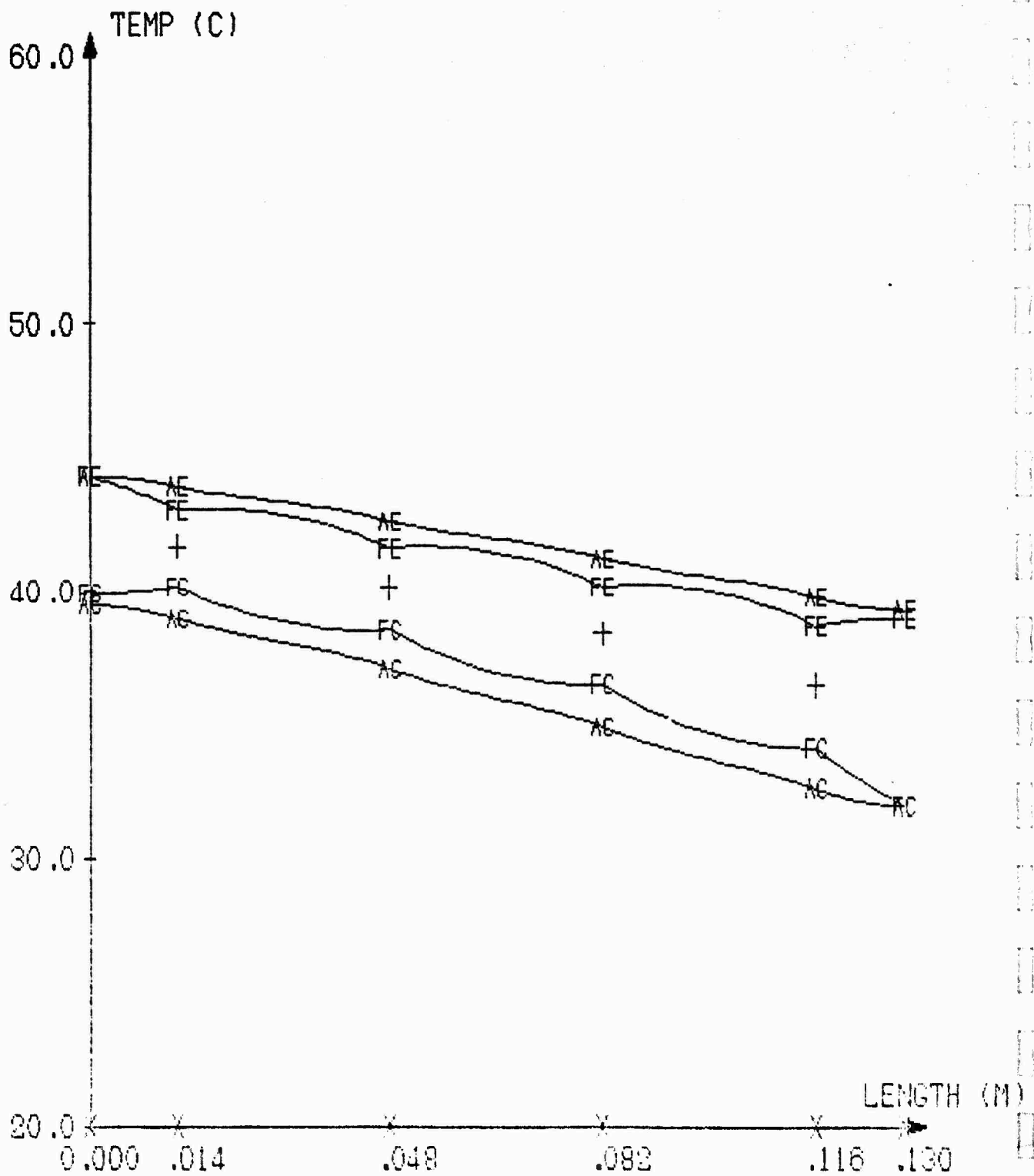


FIG. 23 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

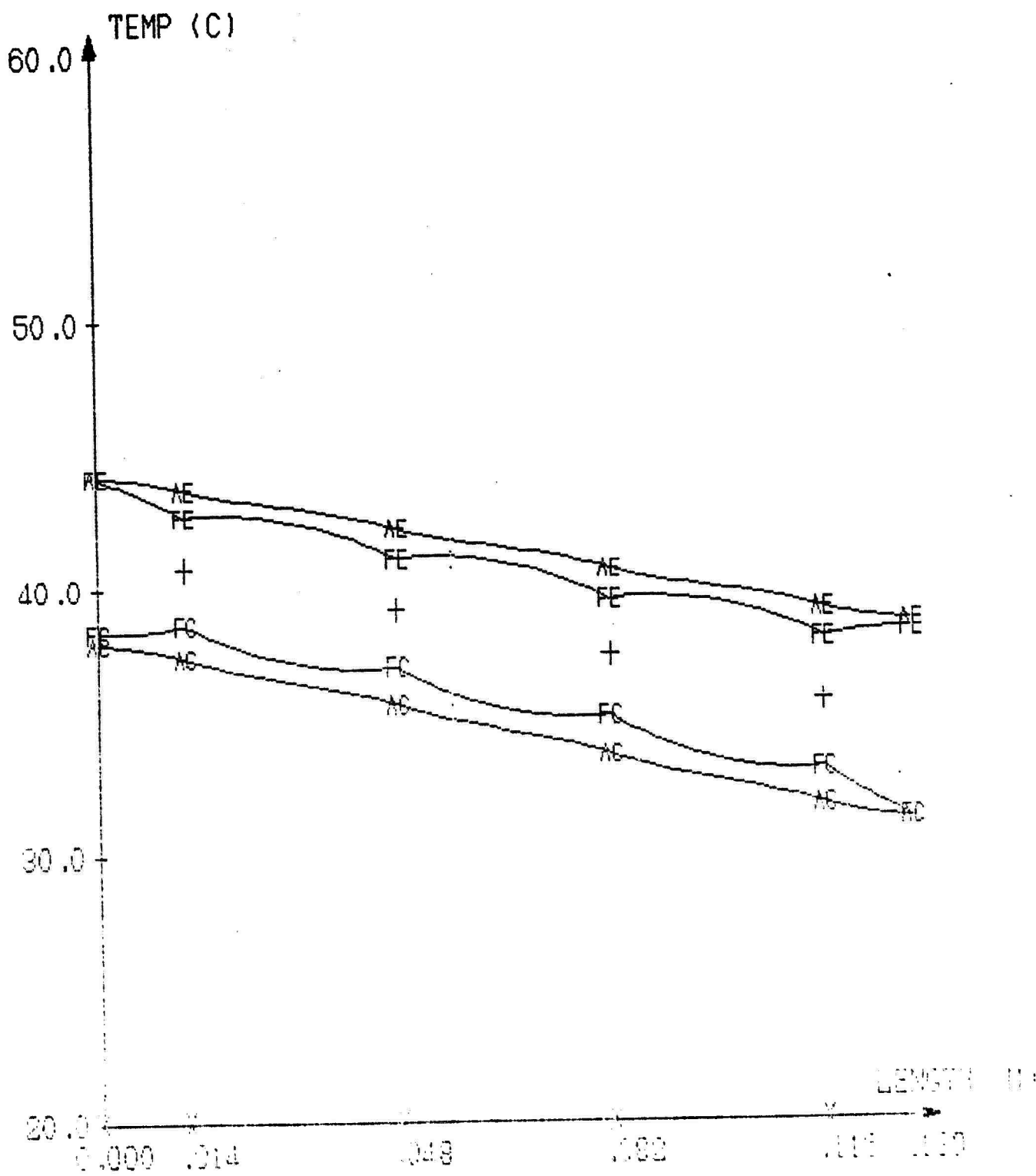


FIG. 24 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

AIR INLET TEMPERATURES 45 C, 30 C, MASS FLOW RATIO .8

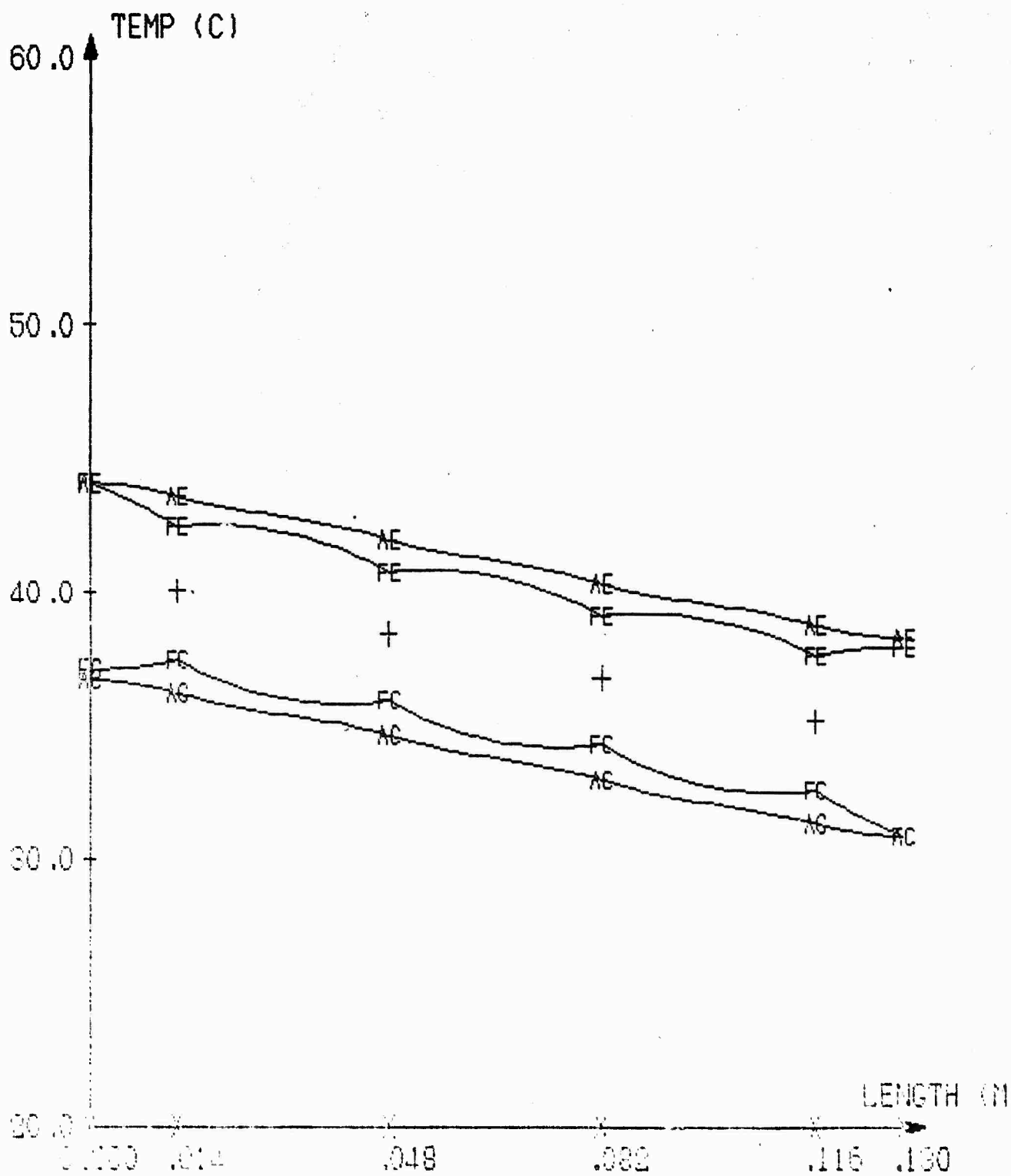


FIG. 25 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

AIR INLET TEMPERATURES 15 C, 20 C, MASS FLOW RATIO 1

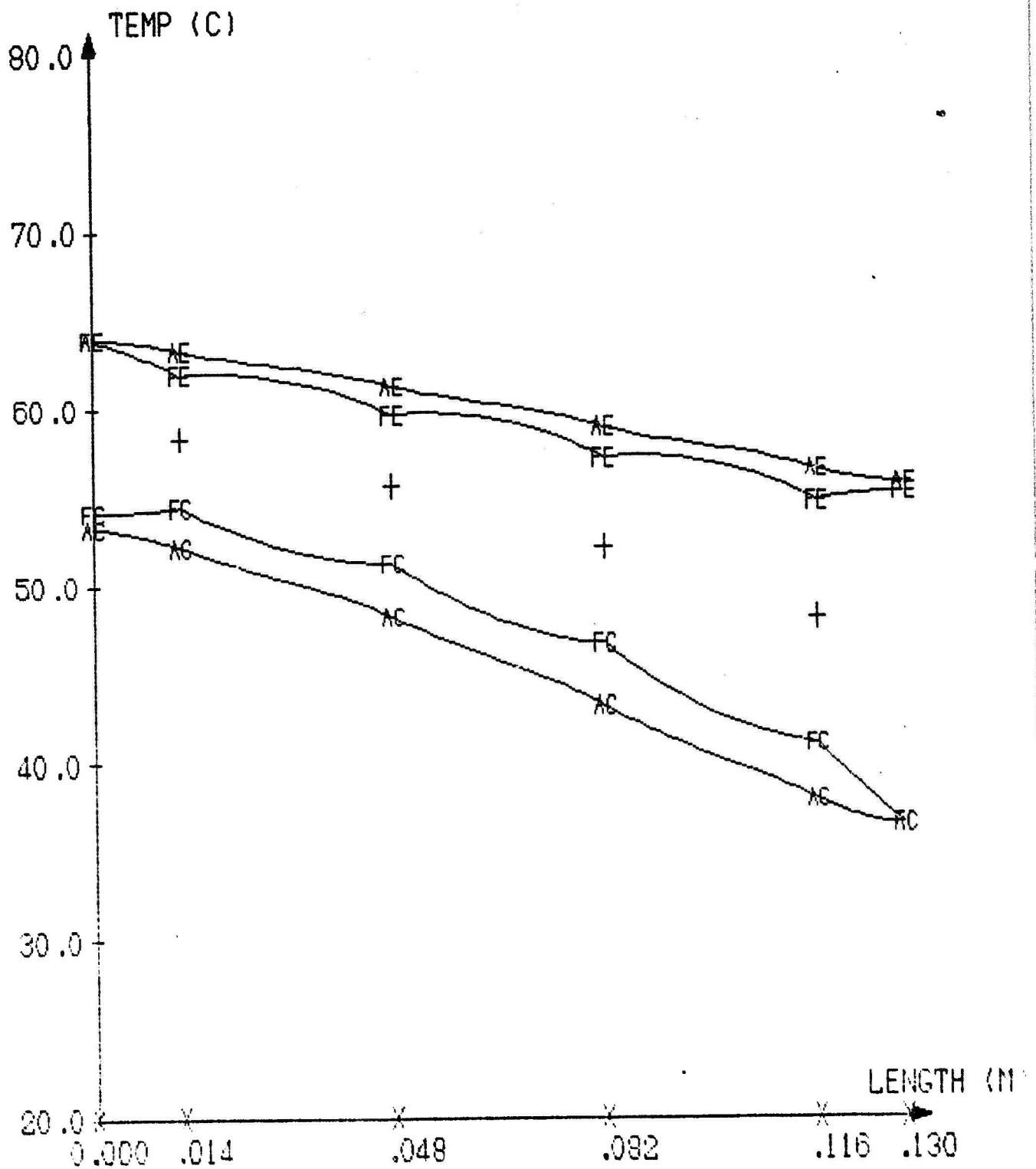


FIG. 26 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

AIR INLET TEMPERATURES 65 C. 30 C ; MASS FLOW RATIO .4

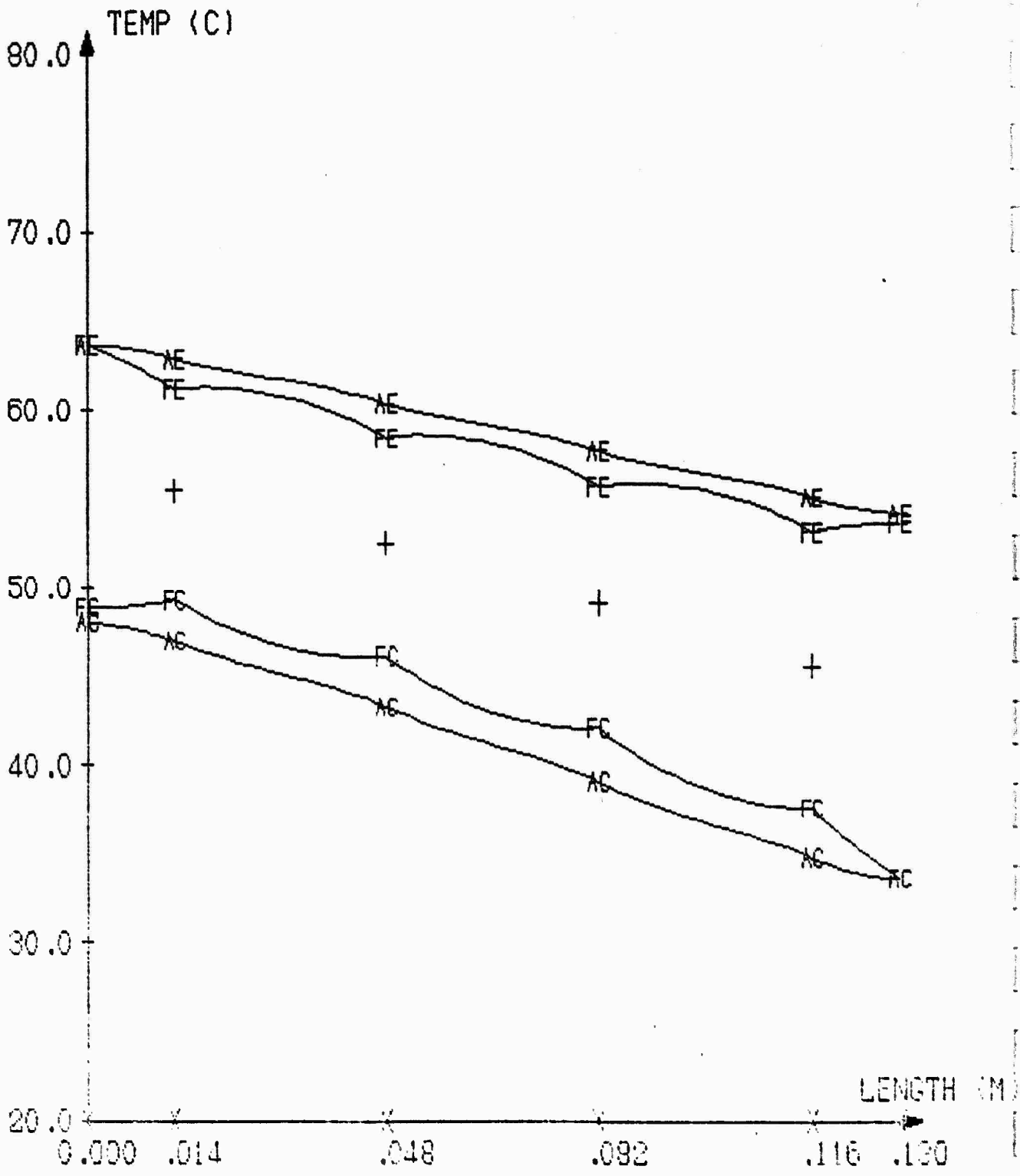


FIG. 27 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.
 (+) HEAT PIPE VAPOR TEMPERATURE

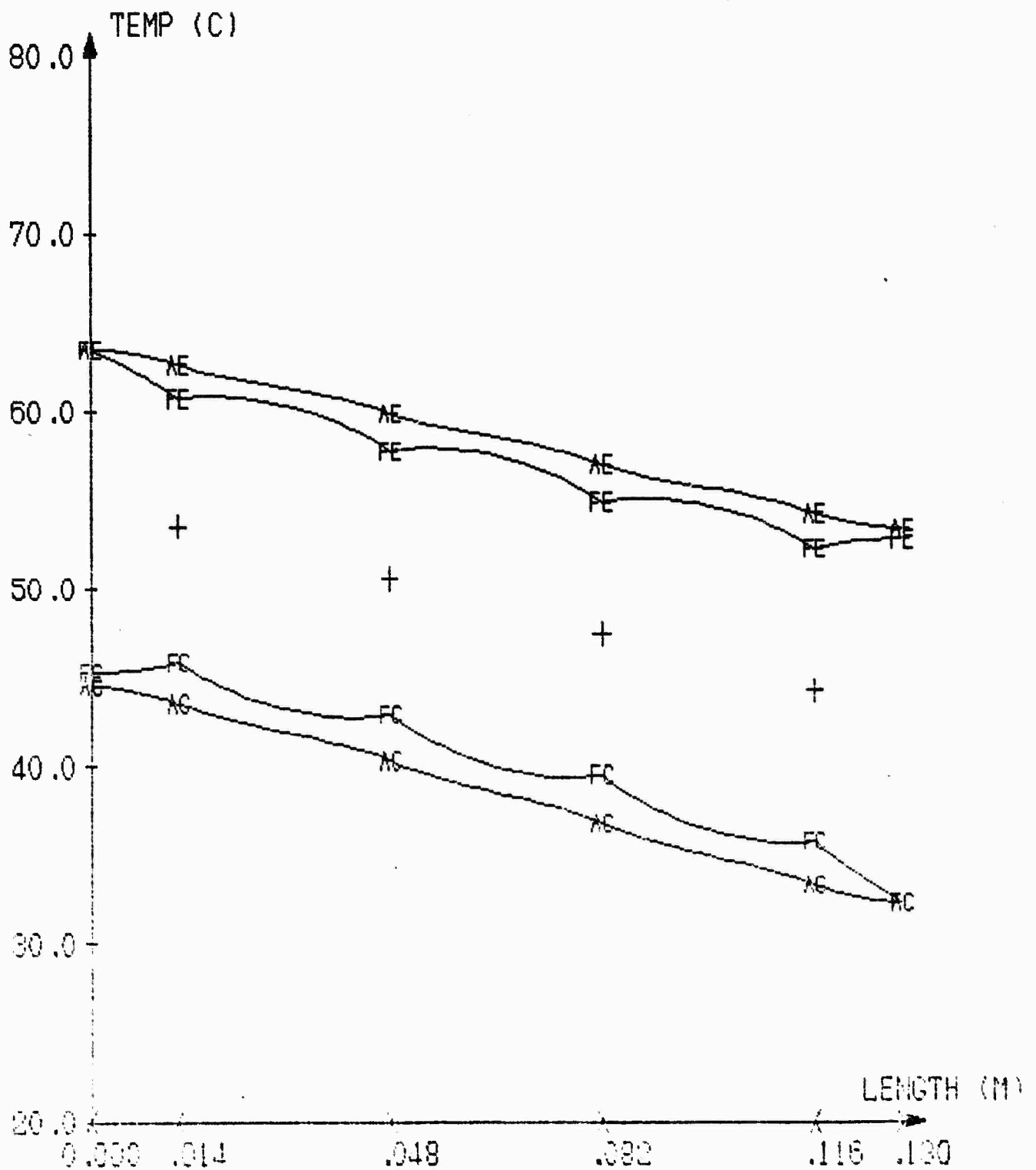


FIG. 28 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

AIR INLET TEMPERATURES 85.0, 90.0, MASS FLOW RATIO 0.0

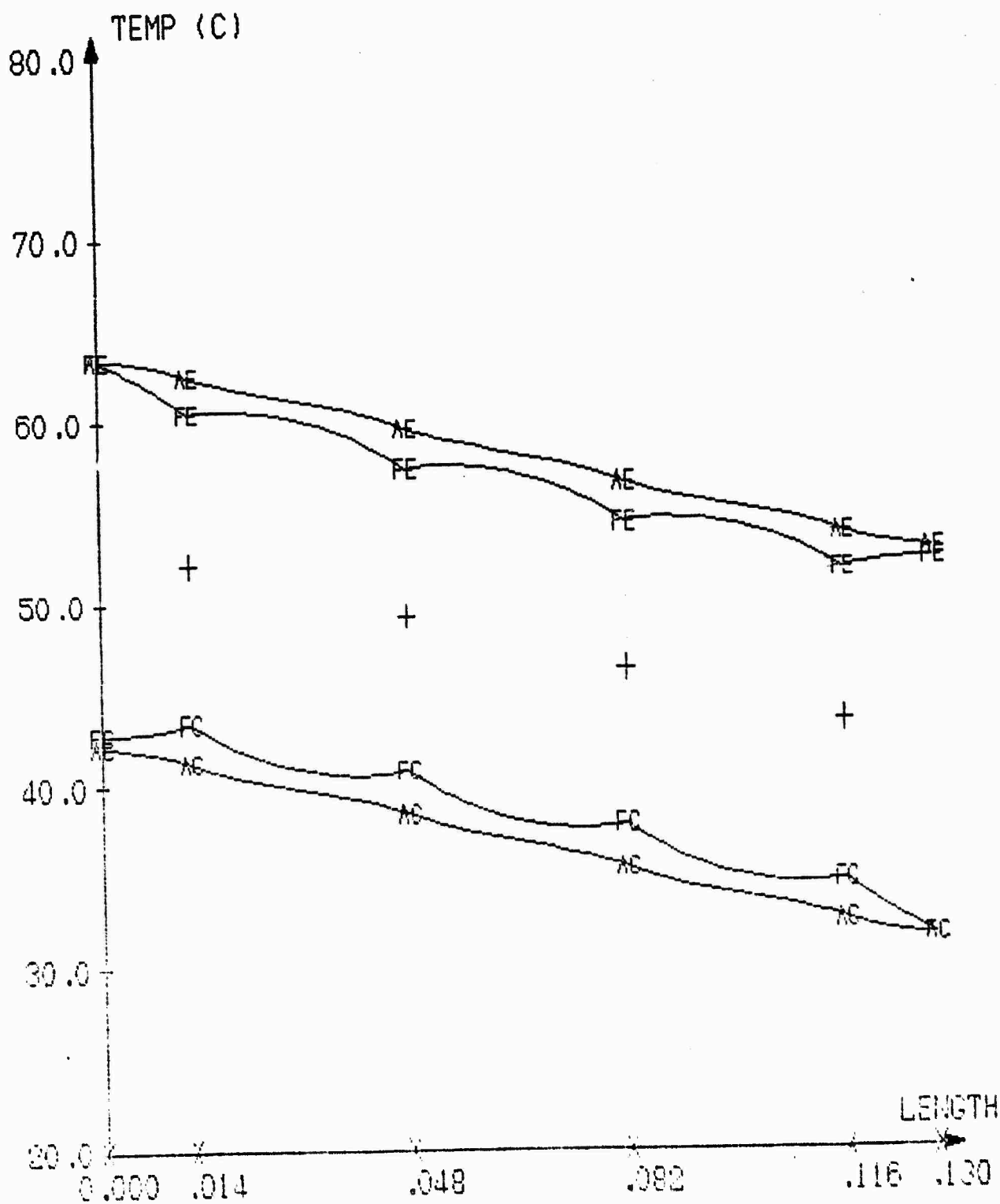


FIG. 29 AIR (A) AND FIN (F) TEMPERATURE PROFILES IN THE EVAPORATOR (E) AND CONDENSER (C) SECTIONS OF AN HEAT PIPE HEAT EXCHANGER.

(+) HEAT PIPE VAPOR TEMPERATURE

AIR INLET TEMPERATURES 65 C, 30 C ; MASS FLOW RATIO 1.

FIG. 30 - COMPARISON OF COMPUTATIONAL AND EXPERIMENTAL RESULTS : TRUE HEAT EXCHANGER EFFECTIVENESS VS. NORMALIZED RESERVOIR TEMPERATURE

