NASA CONTRACTOR REPORT

NASA CR-1400

HEAT PIPE DEVICES FOR SPACE SUIT TEMPERATURE CONTROL

by A. P. Shlosinger

Prepared by
TRW SYSTEMS GROUP
Redondo Beach, Calif.
for Ames Research Center

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • OCTOBER 1969

NASA CR-1400

HEAT PIPE DEVICES FOR SPACE SUIT TEMPERATURE CONTROL

By A. P. Shlosinger

Issued by Originator as Report No. 06462-6005-R0-00

Prepared under Contract No. NAS 2-3817 by
TRW SYSTEMS GROUP
Redondo Beach, Calif.

for Ames Research Center

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

For sale by the Clearinghouse for Federal Scientific and Technical Information
Springfield, Virginia 22151 – CFSTI price $3.00

FOREWORD

This report summarizes the results of technological research activities of general interest which were generated in the performance of a program with the title "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits". The specific application to space suits and related materials research performed are covered by separate reports and referenced in this report.

The program was sponsored by the Ames Research Center of the National Aeronautics and Space Administration at Moffett Field, California, under Contract No. NAS 2-3817. The period of performance covered in this report is 30 June 1966 to 1 September 1968. Mr. James Blackaby of the Environmental Control Research Branch directed this effort for the NASA/Ames Research Center.

The program was performed by the Environmental Sciences and Engineering (ESE) Department of the Systems Engineering and Integration Division of the TRW Systems Group. Mr. A. P. Shlosinger, Head of the ESE Research Section was the Principal Investigator and Program Manager. Valuable contributions in the performance of this program were made by W. Woo and E. W. Bentilla, Members of the Technical Staff; G. Hwa, Engineer and Research Specialist; G. B. Patchell and J. G. Morgan, Laboratory Technicians.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Cont'd)

HEAT PIPE DEVICES FOR SPACE SUIT TEMPERATURE CONTROL

RESEARCH REPORT

by

Arnold P. Shlosinger

SUMMARY

Investigations performed to develop techniques for control of temperature in extravehicular space suits have provided technological knowledge of general applicability to temperature control systems. This report summarizes this knowledge. Other reports, referenced in this report, deal with the specific application to space suits and the related materials research performed. The following techniques have been investigated and are reported in this report:

A heat pipe "thermal switching" device is described and test results presented. Techniques for bonding of capillary structures to solid substrates are discussed. Recommendations for suitable bonding techniques are provided. Heat pipes of flexible materials are described and data resulting from experimentation with flexible heat pipes presented. Concepts for preventing freezing or for restart of frozen heat pipes are discussed and experiments, demonstrating feasibility of one of these concepts, are described and data presented.

INTRODUCTION

Background

A study program was performed for purpose of improvement and simplification of the techniques presently used for body temperature control of astronauts during extravehicular activities. The results of this study program, in its specific application to space suits, are summarized in another, separate report (ref. 1). The research activities required to modify the basic concept of the heat pipe to application in a space suit have, however, generated knowledge which has applications outside and beyond this specific program.

This report summarizes this knowledge. It deals with modifications to heat pipes and techniques for the fabrication of modified heat pipes in terms of general technical information rather than with regard to the specific application to space suit temperature control.

Heat Pipes

The first publication on the subject of heat pipes was probably an article which appeared in the Journal for Applied Physics in June, 1964 entitled "Structures of Very High Thermal Conductance" by G. M. Grover, T. P. Cotter and G. F. Erickson from the Los Alamos Scientific Laboratory. This article described the construction and the results of several experiments with tubular heat pipes and also introduced the term "Heat Pipe".

A heat pipe was defined as a pipe or tube, internally lined with a capillary structure, closed on both ends and evacuated of noncondensable gases. The capillary structure, such as a wire mesh wick or porous ceramic tube, would be soaked with a suitable working fluid, for example, water or a liquid metal. Functioning as closed evaporating-condensing cycles, heat pipes demonstrate very high apparent rates of heat transfer. Essentially they perform like rods of extremely high thermal conductance. Apparent thermal conductances, orders of magnitude higher than those of the best metallic thermal conductors, were observed.

The method of heat transport in a heat pipe is the well-known process of heat transport by mass transfer and liquid-vapor phase change. When heat is added to any part of the heat pipe, a small local temperature rise results in an increase of vapor pressure of the liquid retained in the capillaries of the wick. Liquid will evaporate and the vapor flow to the cooler areas of the tube cavity due to vapor pressure differences. There the vapor will condense and the liquid condensate will return by capillary action of the wick to the areas of the wick which are deficient in liquid as a result of the evaporation which had taken place.

The process thereby involves phase change, first from liquid to vapor with absorption of the latent heat of vaporization and then from vapor to liquid with release of the latent heat of condensation. Because latent heats of phase change of many substances are high, the process shows the above described high rates of heat transfer. Also, because condensate return results from capillary action and is not dependent on gravity, the device is applicable under zero "g" conditions; in fact, it is one of the few devices known which will work better in a zero "g" field than on earth.

It is quite obvious that the process is in no way limited to the geometry of a pipe or tube. Any cavity, lined internally with a capillary structure, free of noncondensible gases and with the capillary structure soaked with a suitable liquid will transport heat by the described process. Heat flow will take place in any direction, from areas of heat addition to areas of heat removal and only a very small temperature gradient is required as driving force.

The magnitude of the temperature gradient existing between the evaporating and the condensing liquid-vapor interface is obviously a function of the resistance to vapor flow. The fluid in the heat pipe is in a saturated condition due to the simultaneous presence of the liquid and vapor phase. Temperature and pressure are therefore related by the saturation curve of the fluid. The temperature of the liquid phase will rise until the increase in vapor pressure provides an adequate driving force for the vapor to flow to the cooler areas. Where vapor flow passages are relatively large, as in a tubular heat pipe, the pressure drop of the vapor is very small, hence, the temperature gradient will be often too small to be measured. The heat pipe will then be essentially isothermal.

In order to be able to apply this interesting device to application in a space suit, a number of modifications to the basic heat pipe concept were required. The reasons for these modifications are discussed in ref. 1 and 2. Suffice to say that they include:

- o Techniques to make the device a controllable heat flow device responsive to externally applied control signals.
- o Techniques to attach capillary structures such as wicks to flat or curved internal surfaces of the heat pipe cavities.
- o Techniques to fabricate heat pipes of flexible materials.
- o Techniques to deal with the problem of restarting heat pipes in which the working fluid has solidified as a result of low heat rejection temperature and/or temporary reduction of heat input.

THE CONTROLLABLE HEAT PIPE

The necessity to make heat pipes controllable, such that heat flow could be modulated and stopped in response to externally applied control signals was established. In order to accomplish this the processes taking place in a heat pipe and the possibilities to influence these processes and thereby control heat flow were evaluated.

The operation of a heat pipe can be described in terms of six distinct simultaneous processes: (ref. 3)

(1) Conductive heat transfer from the outside face of the heat pipe to the liquid-vapor interface at the wick surface.

(2) Endothermic liquid to vapor phase change, i.e. evaporation, at the liquid-vapor interface at the wick surface.

(3) Flow of the vapor from the warmer to the colder regions of the heat pipe.

(4) Exothermic vapor to liquid phase change, i.e. condensation, at the vapor-liquid interface at the wick surface.

(5) Conductive heat transfer from the vapor-liquid interface to the outside surface of the heat pipe.

(6) Capillary flow of the liquid working fluid from the condenser to the evaporator regions of the heat pipe.

The first five of these six processes are heat and mass transfer processes in a series arrangement. A temperature gradient is required as driving force for each of these five processes.

Temperature gradients are obviously required for the conductive heat transfer in processes Nos. (1) and (5). Evaporation in process (2) and condensation in process (4) require superheating and subcooling respectively. Vapor flow in process (3) requires a pressure difference which, under the saturated fluid conditions within a heat pipe cavity, requires the existence of a temperature gradient. We can therefore state that processes (1) through (5) will cause a temperature gradient from the heat input surface to the heat output surface of the heat pipe and that this temperature gradient will be related to area and heat flow, very similar to the relationships in conductive heat transfer. Increase of this gradient will cause larger heat flow and inversely, diminishing heat flow will reduce this gradient.

Process No. (6), the capillary flow of the liquid working fluid from the condenser to the evaporator end of a heat pipe is an inherently different process. Capillary force rather than temperature gradient provides the driving force. Increasing temperature gradient will not increase, but counteract capillary flow. This because the capillary force must overcome the pressure differential, which exists between the evaporating and condensing end of the cavity, in addition to liquid flow resistance. In most heat pipe configurations, this pressure differential is small and does not significantly interfere with capillary pumping. The capability of the capillary structure to pump will therefore be either nearly independent of, or have an inverse relationship to the temperature gradient in the heat pipe.

Increasing temperature gradient between heat source and heat sink, with resulting increase in heat flow will, however, lead eventually to complete cessation of the heat transport function when the amount of fluid evaporated in the evaporator end of the device exceeds the capillary pumping capacity of the wick.

The first five processes impose resistances in a series arrangement on heat flow. Increase in heat flow will result from an increase in temperature gradient and vice versa. The sixth process, i.e. capillary flow in the wick will however behave in a different way and will not respond to a change in gradient across the device with change of transfer capability.

From considerations of controlling heat flow in a heat pipe, this distinction is important. Increase in resistance to heat or mass transfer in any one of the first five transfer processes will in effect increase the apparent thermal resistance. An increased driving potential in the form of a larger temperature gradient will be required to overcome this increase in effective resistance.

One widely suggested technique of heat flow control in heat pipes is based on the effect the presence of gases, which are noncondensable in the range of operating temperatures of a heat pipe, have on heat transport capability. In a heat pipe there is continuous flow of vapor from areas of heat addition to areas of heat removal. Noncondensable gases are therefore flushed by the vapor towards the cooler end of the device. In a tubular heat pipe, this results in the formation of a pocket of noncondensable gas at the cool end. Heat transfer through a noncondensable gas, especially in the absence of gravity convection effects, is small. The presence of a noncondensable gas, therefore, in effect shortens the active length of the heat pipe. Rate of heat rejection at the cool end of a heat pipe is dependent on effective area. The presence of a noncondensable gas will reduce this area and therefore the amount of heat which can leave the external surface of the device.

Introduction or removal of a noncondensable gas into a heat pipe could thereby be used to control the rate of heat which will be transferred from the wick face where condensation occurs to the external heat pipe surface. This technique can be considered as control of process No. (5). It is control of conductive heat transfer from the condensing vapor-liquid interface to the external heat pipe surface by variation of the effective heat transfer area.

This technique was not suitable to application to space suit temperature control. In a tubular heat pipe, noncondensable gas can be vented to a vacuum without a significant loss of working fluid vapor, if the venting is from the cooled end of the heat pipe. The noncondensable gas would, however, either be lost to space or would have to be recovered and recompressed, to be available as a control medium when reduction of heat flow is again desired. Either alternative was considered undesirable for the specific application.

In the application under question, the shape of the heat pipe is furthermore that of a flat, thin panel with major direction of heat flow occurring perpendicular to the flat surfaces. These heat pipes could therefore be described as only a few inches in length and very wide, with one of the panel surfaces being cooled. A distinct "pocket" of noncondensable gases would therefore not be formed. Rather, there would be a thin layer of noncondensable gas distributed over a relatively large, flat surface and any local venting from such a surface would unavoidably incur significant losses of working fluid vapor.

Of several other methods conceived for control of heat transmission in a heat pipe a technique, schematically shown in Figure 1, was selected. It is based on control of process No. 3, i.e. the flow of vapor from evaporator to condenser. It requires modification of the basic heat pipe into a two-chamber heat pipe with separate evaporating and condensing chambers and separate capillary and vapor flow passages. This modification permits control of the vapor flow by a valve. The vapor passage may be completely closed or flow may be throttled, which would result in a purposely induced pressure differential between the evaporating and condensing chamber. As in a heat pipe the working fluid is in a saturated condition, pressure controls the evaporating and condensing temperatures. Introduction of a throttling valve into the vapor passage will therefore permit control of the temperature gradient between the evaporating and the condensing chamber. This technique therefore provides complete shutoff or control of the temperature gradient in response to an externally generated control signal.

Application of this technique requires that the vapor flow passage, which is controlled by the valve, is the only path for vapor flow from the evaporating to the condensing chamber.

Evaluation of the feasibility of this requirement leads to the following considerations: As Figure 1 shows, the chambers are internally lined with a wick and the wick linings are interconnected by a transfer wick, fitted into a tube. A wick will be impermeable to vapor provided all its capillaries are filled with liquid. Therefore, the wick shown in Figure 1, when tightly fitted into a tube, would not permit vapor to pass and the device would be responsive to the vapor passage control valve.

The force which retains the liquid within the wick is, however, the capillary pressure. Given enough pressure difference between the evaporating and the condensing chamber, the liquid will be driven out of the capillaries and vapor flow through the pores of the wick will take place.

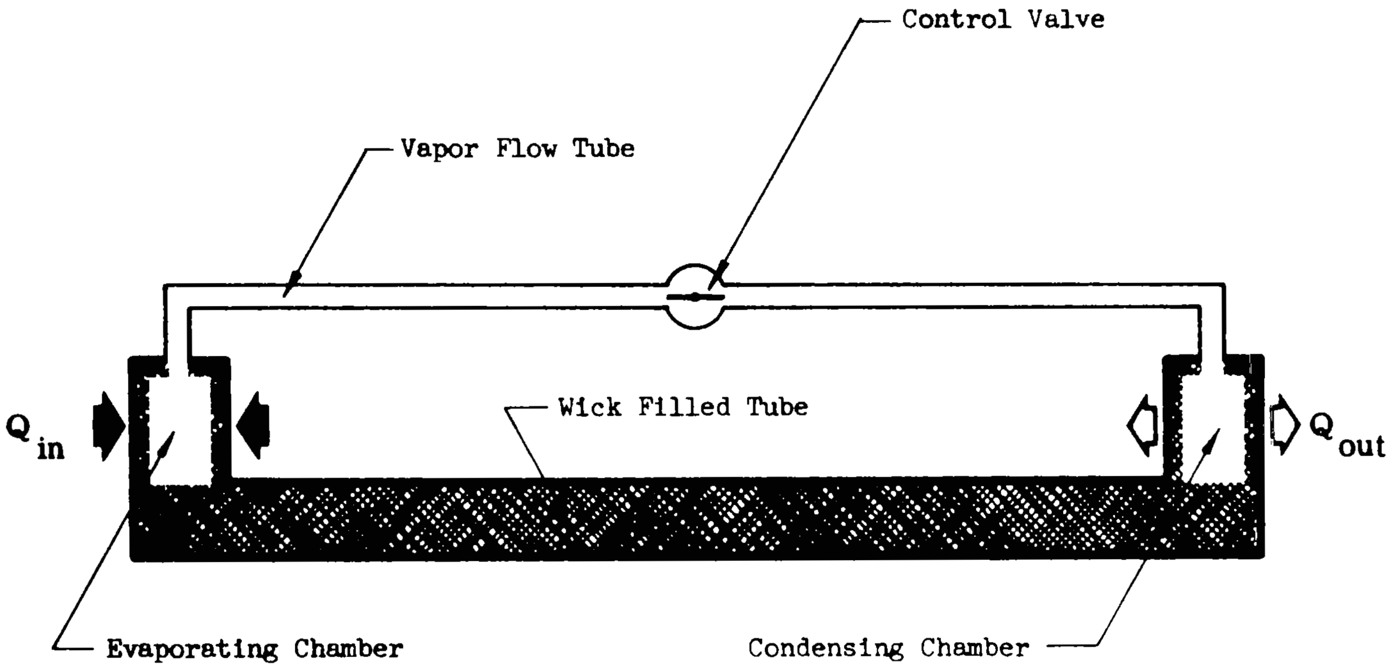

FIG.1 VARIABLE HEAT PIPE (VAPOR FLOW CONTROL CONCEPT)

This will not defeat the purpose of complete closing of the vapor transfer valve. If the vapor transfer valve is completely closed, the intent is to stop heat transmission in the heat pipe completely. Forcing the liquid working fluid out of the wick and driving it into the condensing chamber would by itself have the same effect and would not cause heat transmission in the heat pipe.

The situation is different in regard to throttling flow of heat in a heat pipe. By imposing a resistance to vapor flow by throttling of the vapor transfer valve we want to increase the temperature difference between the evaporating and the condensing chambers. For a given rate of heat flow, we want to establish a specific temperature difference, or in other terms, we want our heat pipe essentially to act as a controllable resistor to heat flow. The range over which we can achieve this and thereby achieve modulating control is, however, limited by the pressure differential capillary force can maintain between the evaporating and the condensing chamber. At the level of temperature difference between the two chambers which corresponds to a pressure difference capable of forcing the liquid out of the transfer wick, modulating capability of the device will break down in favor of complete shutoff of heat transmission.

We can therefore expect that, as we throttle vapor flow we will increase the temperature gradient across the device up to a point at which then, quite suddenly, complete shutoff of heat transmission will occur.

It is, of course, desirable to make the modulating range prior to complete shutoff of heat transmission as large as possible. This requires careful selection of the transfer wick. Largest capillary pressure will be obtained by the smallest possible capillaries. The smallest capillaries will, however, not provide maximum capillary pumping because capillary pumping rate is dependent on resistance to liquid flow in the capillaries as well as on available capillary pressure.

A further requirement results from the above considerations. Many wick materials show a nonuniformity of capillary passages. Typically, wicks woven from fibrous materials will consist of fibers spun into yarn and yarn woven into a fabric. Two obvious classes of capillary porosity which will exist in such a fabric are the capillary passages between the fibers of the yarn and those between the strands of yarn which are woven into the fabric.

Significant nonuniformity of the capillary passages will permit vapor breakthrough through the larger capillaries while the smaller ones would still pump liquid in counter flow with the vapor. Separation of the vapor flow passage and the wick passage would thereby be defeated. The transfer wick must therefore have, as far as possible, uniformly-sized capillaries. Only if this can be accomplished will vapor breakthrough drive the liquid out of all capillaries and shutoff of heat transmission be achieved.

Given development of a wick which will provide reasonably high capillary pressure and acceptable uniformity of capillary sizes, it is further required that the wick be able to rapidly pick up liquid and pump it from the condensing to the evaporating chamber when, after an extended period of operation with the vapor transfer valve closed, this valve is opened again. As stated above, closing of the vapor transfer valve will result in the liquid working fluid being driven out of the wick by the vapor. Vapor will flow through the wick passage and condense in the condensing chamber. Given enough time, all of the working fluid will be transferred to the condensing chamber. Dry out of the wick in the evaporating chamber will result.

If the vapor transfer valve is opened, pressure between the two chambers will equalize through the vapor passage. This will result in re-initiation of capillary pumping from the condensing chamber to the evaporating chamber. In order to reestablish heat transmission by the heat pipe, it is desirable that this capillary pumping rate be high and that the wick in the evaporating chamber be wetted with liquid working fluid in the shortest possible time after the vapor transfer valve is opened.

From these considerations, it was apparent that the development of the transfer wick is, in itself, a significant research task. In order to progress with the development of a variable effective thermal resistance heat pipe device (thermal switch), it was decided to pursue development of the transfer wick as a parallel effort with the development of the design of the thermal switch. Full understanding of the transfer wick problem would permit evaluation of the performance of experimental devices with a less than perfect wick. This wick would be selected to provide at least reasonable uniformity of capillary size. On-off operation of the thermal switch could be evaluated even with an imperfect wick. Although modulating capability would be desirable, satisfactory on-off switching performance would, at worst, satisfy the requirements. Development of a more nearly perfect transfer wick would permit progression towards modulating operation of the thermal switch.

For the application to a space suit, thermal switch configurations somewhat different from the one shown in the schematic of fig. 1 were developed. Instead of devices much longer than wide, as common for heat pipes, flat panels functioning as thermal insulating panels, whose effective thermal conductance can be varied over a wide range, were required. These panels are two-chamber heat pipes with the vapor and wick passages penetrating a layer of highly effective thermal insulation. A device used for experimental investigations is shown in fig. 2. It consists of circular evaporating and condensing chambers. The chambers are machined from stainless steel and closed by blank copper flanges as shown. A wick and a vapor passage are provided by bonding fiberglass tubes into the walls of the evaporating and condensing chambers.

A pneumatically operable valve is used for control of the vapor passage. The transfer wick between the two chambers was initially a strip of woven quartz fiber cloth, rolled into a cylinder and fitted into the wick transfer tube. It was found to be unsatisfactory. A different type wick had meanwhile been developed which consists of heat resistant glass fiber filtering wool packed to a density of 0.95 g per cc. The glass wool is retained in the tube by perforated pressure plates, held together by a centrally located bolt (fig. 3). Braided quartz fiber wicks are used to transfer fluid through the pressure plate perforations from the packed glass wool to the wick linings of the copper plates of the evaporating and condensing chamber. Advantage is taken of the fact that wicks, brought into intimate contact with each other, readily transfer fluid between their capillaries.

An experimental program was performed with the objective to obtain data on heat transmission performance of the device with the valve open or closed, at several power input and heat sink temperature levels and to observe the transient response of the device to opening and closing of the vapor transfer valve. Water was used as the working fluid. The experiment arrangement is schematically shown in fig. 4.

The device was instrumented by locating a thermocouple each on the external surfaces of the copper plates and the internal faces of the wicks in the evaporating and the condensing chamber.

The tests were performed inside a vacuum bell jar, using reflective shield-type vacuum insulation to simulate an adiabatic condition. Heat was supplied to the top surface of the device by an

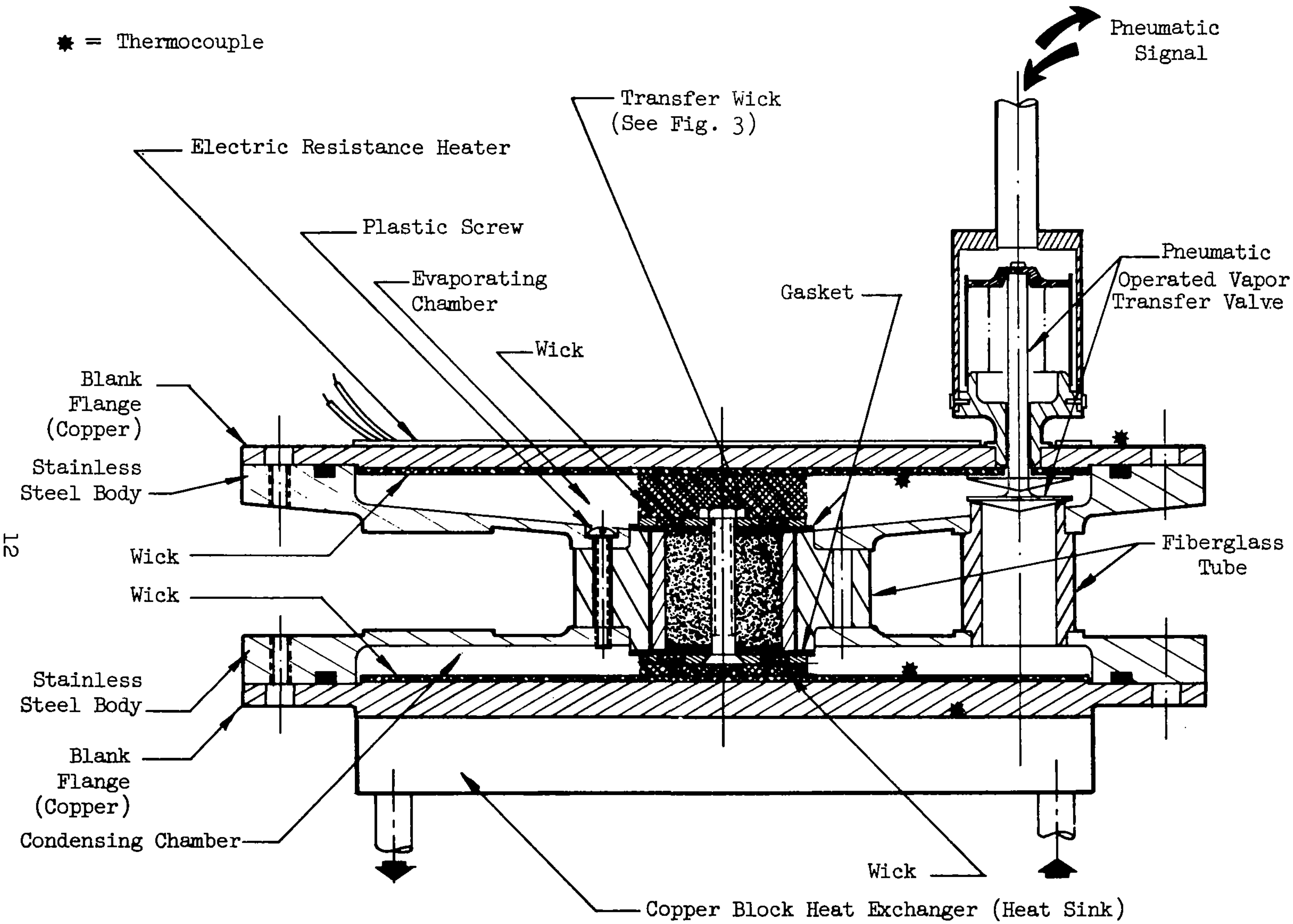

FIGURE 2 - THERMAL SWITCH (CROSS SECTION)

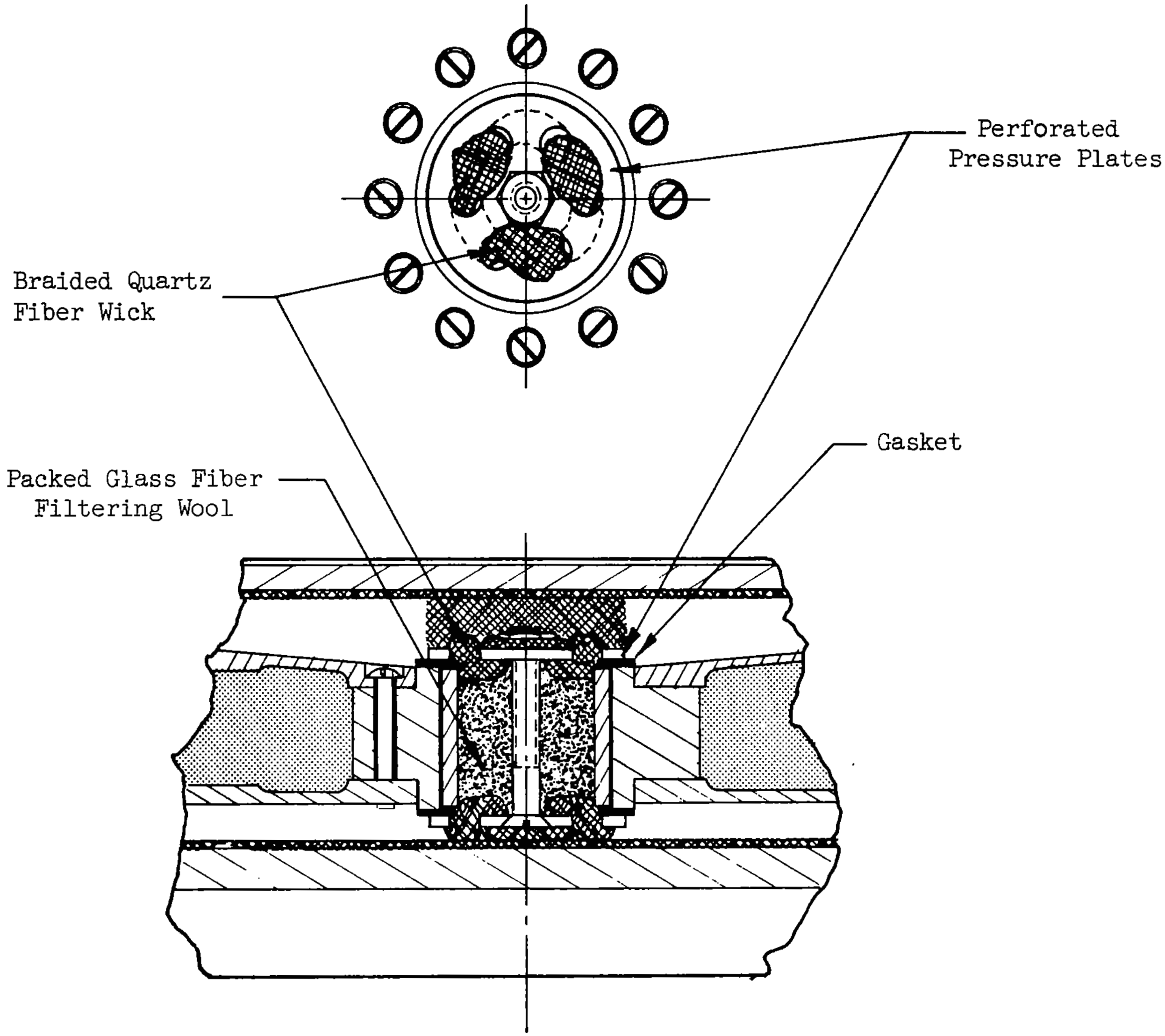

FIG. 3 TRANSFER WICK

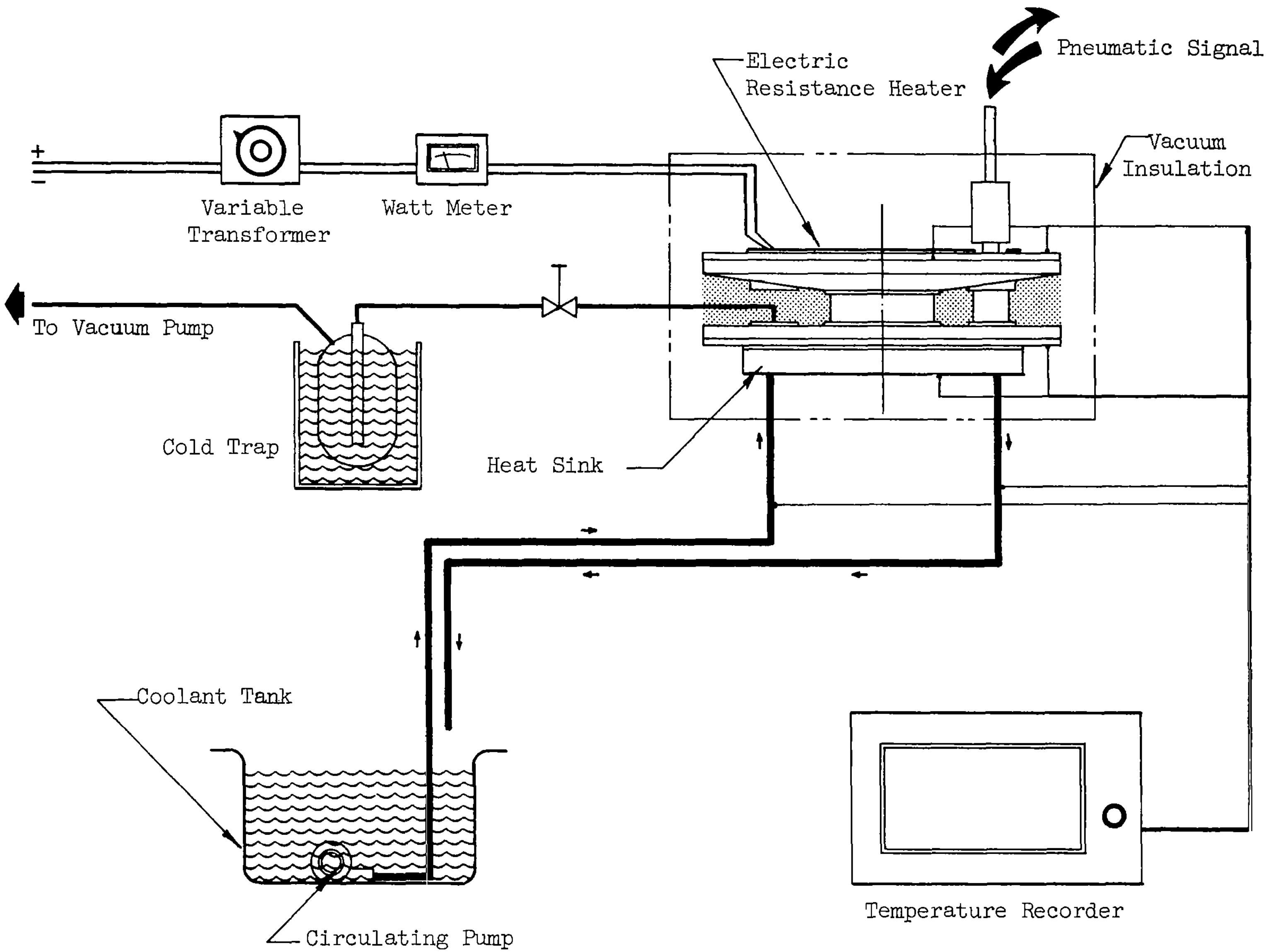

FIG. 4 SCHEMATIC OF THERMAL SWITCH EXPERIMENT ARRANGEMENT

electric resistance heater. The bottom plate of the device was in intimate thermal contact with a copper block-heat exchanger through which cooling water at various temperatures was circulated. This arrangement provided heat flow in a downward direction, hence "negative g" operation.

The following test procedure was used: Cooling water at a selected temperature was circulated through the copper block-heat exchanger. The vapor transfer valve was maintained open and power at a selected level was supplied to the electric resistance heater on top of the device. With the vapor transfer valve open, the device operated as a regular heat pipe. After steady state operation of the heat pipe was attained, the vapor transfer valve was closed. This stopped operation of the device as heat pipe. Only a small amount of heat was transferred by conduction and radiation, and most of the heat input provided by the electric heater was stored in the thermal mass of the evaporator chamber, resulting in a temperature rise. The temperature of the evaporator chamber was permitted to rise to a specified level at which time the vapor transfer valve was opened. Heat pipe function was reestablished. During the tests the temperatures of the four (4) thermocouples were recorded.

From a number of test runs which were performed at various heater power levels and heat sink temperatures, figures 5, 6, 7 and 8 were selected as typical examples of thermal switch operation. Figure 5 shows heat pipe performance with the vapor transfer valve in an open position up to time "zero". Heater power input was 20 W. The slight temperature rise of both evaporator and condensing chamber temperature from time "-20" to time "0" resulted from a temperature rise of the coolant, circulating through the heat sink. At time "zero", the vapor transfer valve was closed. This resulted in a temperature rise of the evaporator chamber up to 60° C over a period of approximately 22 minutes. For this experiment, 60° C had been selected as maximum evaporator chamber temperature and at time point "22 minutes", the valve was opened. It is interesting to note the temporary drastic drop from 60 to 50° C of the evaporator chamber wick temperature indicated by the dash line. This drop is believed to be due to the fact that, with a temperature rise to 60° C over a time period of 22 minutes, the evaporator chamber wick had not dried out. Therefore, as soon as the vapor transfer valve was opened the sudden pressure drop in the evaporator chamber resulted in rapid evaporation of the working fluid from the evaporator wick and an initial temperature drop.

In the condensing chamber the temperature approached, after closing of the valve, the temperature of the cooling water. After

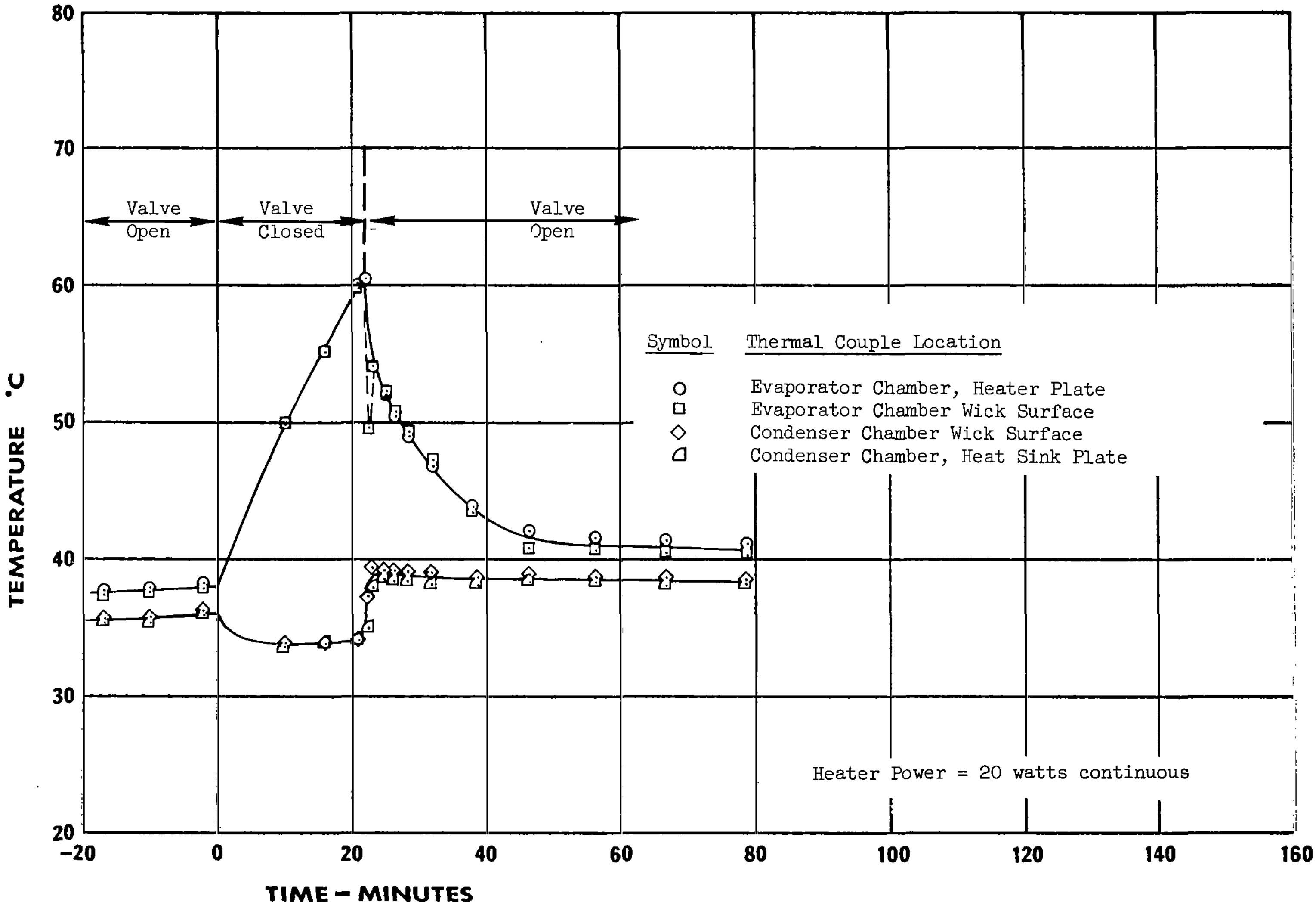

Figure 5 - TWO CHAMBER HEAT PIPE WITH VAPOR FLOW CONTROL VALVE, RUN NO. 23

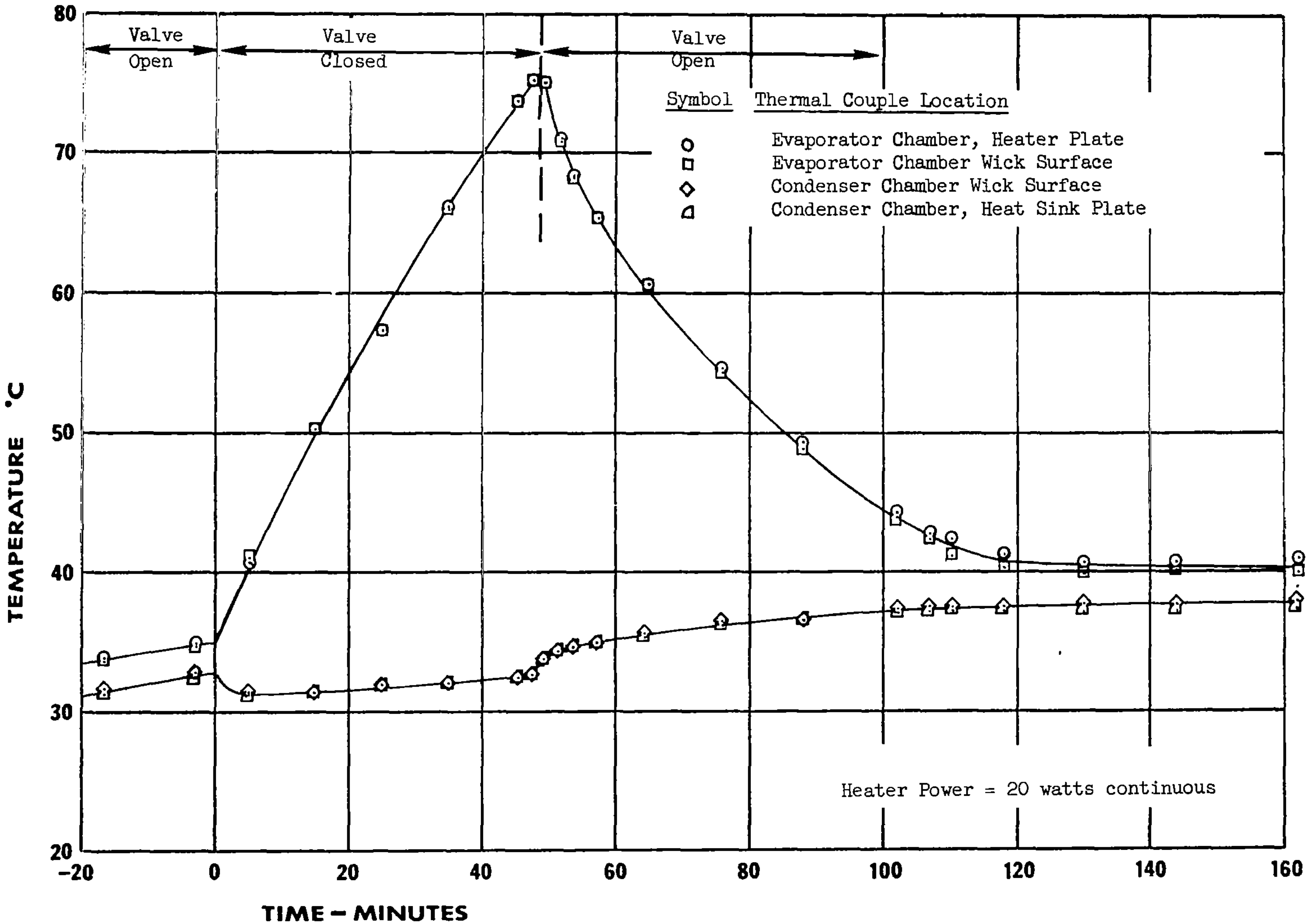

Figure 6 - TWO CHAMBER HEAT PIPE WITH VAPOR FLOW CONTROL VALVE, RUN NO. 20

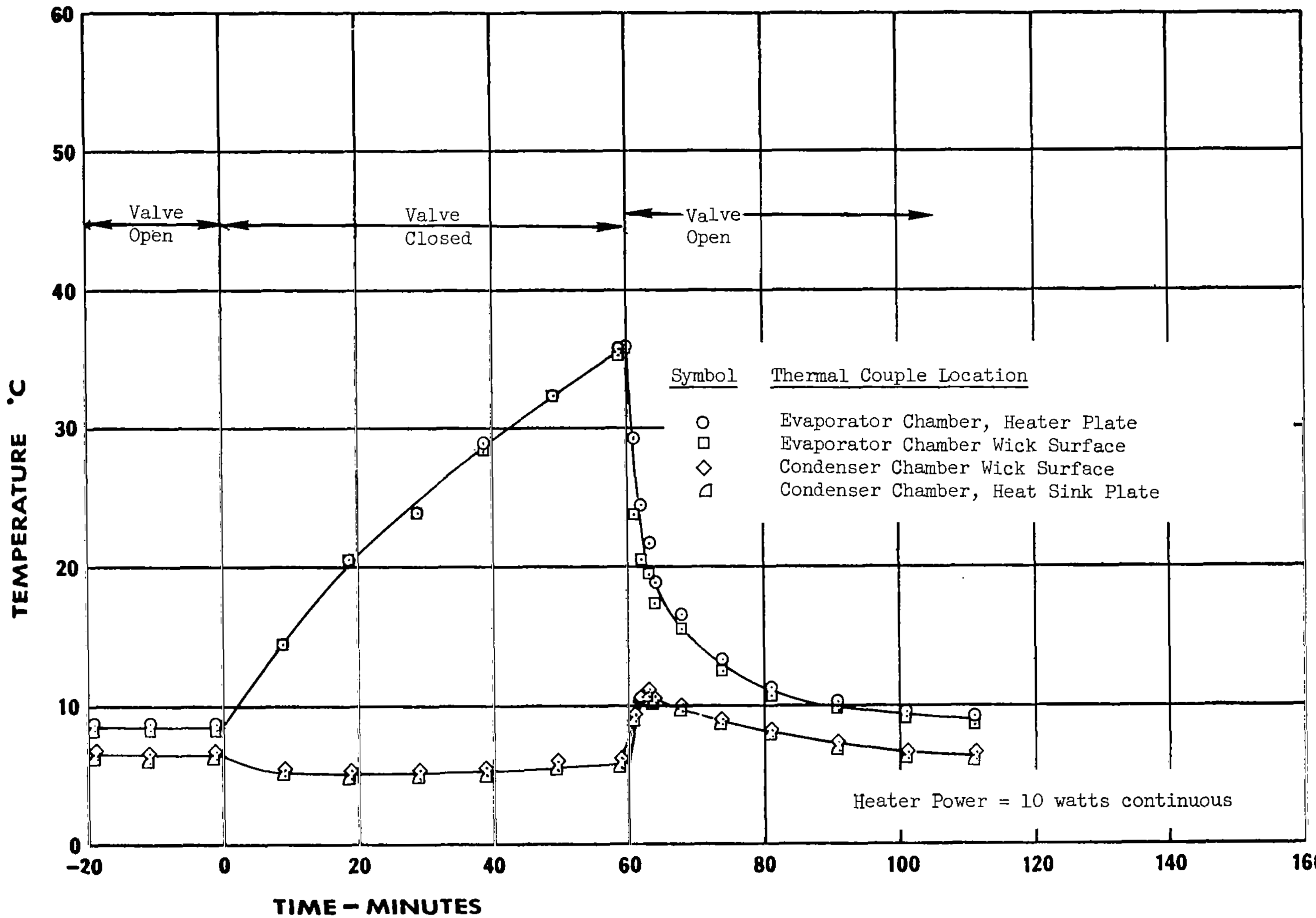

Figure 7 - TWO CHAMBER HEAT PIPE WITH VAPOR FLOW CONTROL VALVE, RUN NO. 1

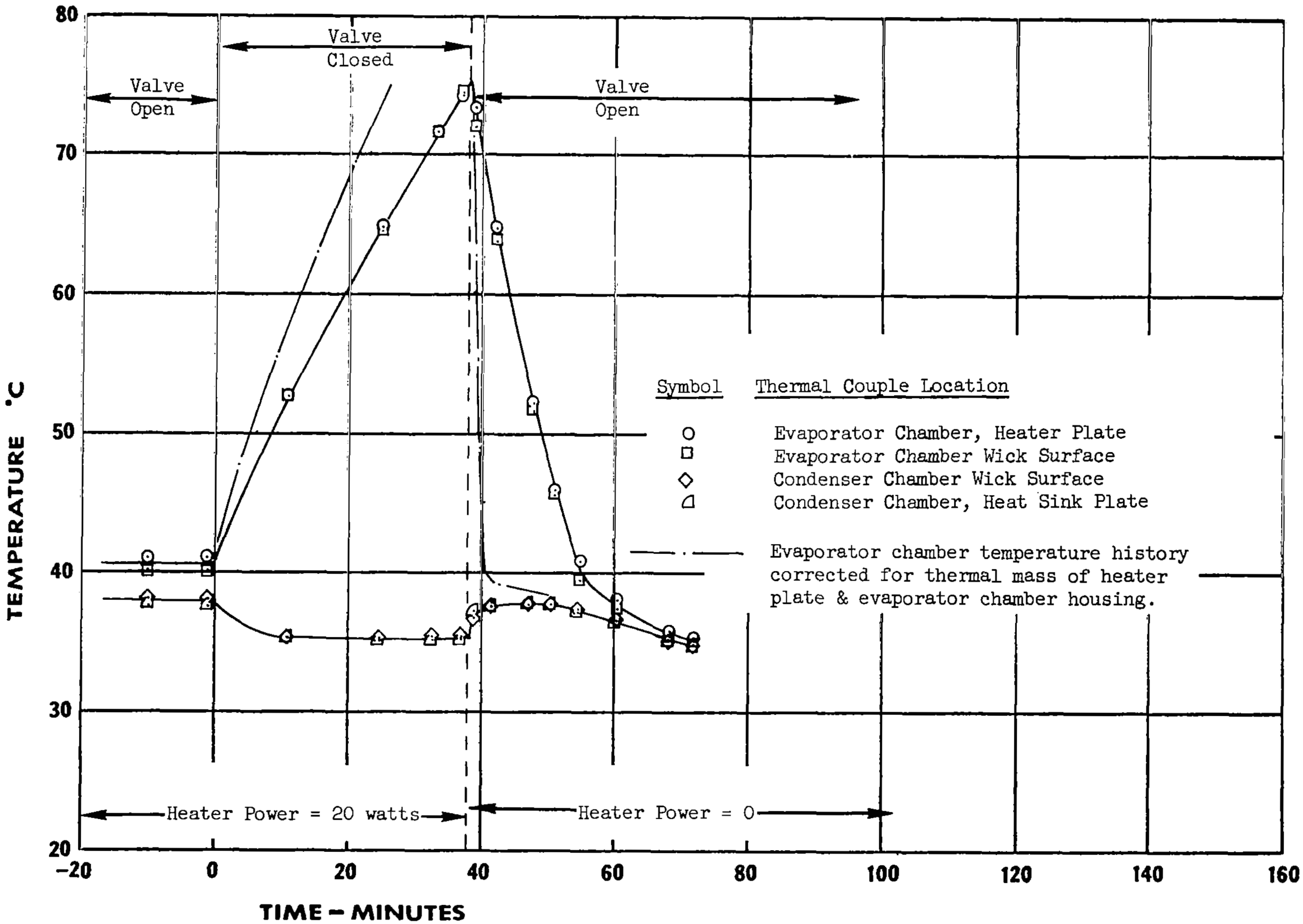

Figure 8 - TWO CHAMBER HEAT PIPE WITH VAPOR FLOW CONTROL VALVE, RUN NO. 21

reopening of the vapor transfer valve, the temperature rose very rapidly to a value higher than observed prior to closing of the valve. This temperature showed, from this point on, a steady slight decline. A temperature difference between the evaporating and condensing chamber, essentially identical to the temperature difference prior to closing of the vapor transfer valve, was reestablished. The temperatures after reopening of the valve were, however, several degrees higher than before the valve was closed.

Figure 6 shows the result of a similar test run during which the temperature of the evaporator chamber was permitted to rise to 75° C. At this higher temperature and with the identical power input of 20 W, wick dry out apparently occured in the evaporator chamber. The sudden drop in temperature in the evaporator chamber when the vapor valve was opened was not observed. A steady, slight rise of the temperature in the condensing chamber with the valve closed is attributed to conductive and radiative heat transfer within the structure of the device, which could be expected to be more significant at the higher temperature difference. The response to reopening of the vapor transfer valve was significantly slower than in figure 5. The temperature difference between evaporating and condensing chamber observed prior to closing of the valve was reestablished, but the temperatures were again higher than prior to closing of the valve.

The slower response after opening of the vapor transfer valve is believed to result from the relatively large thermal mass of the device. The design of this experimental thermal switch is such that both chambers can be opened for purpose of changes of internal instrumentation, changes of wicks, etc. As figure 2 shows, this was accomplished by the use of flanged "O ring" seals. Relatively massive flanges were required to provide the necessary rigidity to obtain a reliable seal.

If the temperature of the evaporator chamber is permitted to rise, significant amounts of heat are stored in the mass of the chamber housing. After reopening of the vapor transfer valve, this amount of stored heat must flow to the condensing chamber and be dissipated to the heat sink in addition to the heat input provided by the electric resistance heater.

The higher level of the four (4) temperature points, without apparent change in temperature gradients after reopening of the valve is attributed to a shortcoming of the experimental arrangement. During experiments where the condenser side of the device was maintained between 30 and 40° C, the source of coolant fluid was a large tank filled with water at the desired temperature. The large volume and heat exchange with the room air were expected to result in an essentially constant temperature. The heat input from the resistance heater via the heat pipe into the water resulted, however, in a steady increase in coolant water temperature and the temperature of the evaporator side, as especially apparent in figure 6, prior to closing of the vapor transfer valve.

The test run shown in figure 7 used an ice bath as source of coolant fluid and the temperature of the coolant water was maintained constant by the melting ice. No increase in the temperature level of the heat pipe occurred while the vapor transfer valve was open.

During this test run the power input into the heater was 10 W. At this lower power level and with the capacity of the heat sink increased by the use of the lower temperature coolant, a better response to reopening of the vapor transfer valve was observed. After reopening the valve, the temperature of the evaporator chamber dropped sharply and the temperatures after the valve closing-and-opening cycle matched those observed prior to closing of the valve.

Figure 8 shows the temperature history of a 20 W test run with essentially the same heat sink temperature of figures 5 and 6. During this test, the heater input power was turned off when the vapor valve was reopened. After a temperature rise, with the valve maintained closed, to 75° C, opening of the vapor transfer valve resulted in a drastic temperature drop and removal of heat stored in the evaporator chamber in a period of less than 20 minutes. Temperature equalization between the evaporator and the condensing chamber to a temperature difference of only a few degrees and approach to heat sink temperature occurred quite rapidly. An analytical correction for the thermal mass of the evaporator chamber was made for this test run and plotted in figure 8 in phantom line. With "zero thermal mass" temperature drops almost vertically from the 75° C point to below 40° C after the vapor transfer valve is opened.

In evaluation of the data presented above, it should be kept in mind that these experiments represent a more severe condition than would occur during an application of the thermal switch to temperature control of, for example, electric/electronic equipment or the human body.

In such an application, the imposed variable would not be opening and closing of the vapor transfer valve. Rather the position of the vapor transfer valve would be controlled such as to maintain the temperature at the heat input side (evaporator) essentially constant. A typical example of such an application may be the temperature control of a battery compartment aboard a satellite, passing through the shadow of the earth. It could be assumed that heat generation would be constant, but heat removal rate would increase during the eclipse period. As heat removal rates increase and temperature of the battery drops, the vapor transfer valve would be closed in response to a temperature control signal in order to prevent excessive heat loss from the battery compartment. When, after passing through the earth's shadow, solar irradiation tends to increase satellite temperature, the vapor transfer valve would open. During the whole cycle, the temperature of the evaporator chamber would be nearly constant.

This distinction is important. In an application of the Heat Pipe Thermal Switch to temperature control, with the evaporating chamber maintained at an essentially constant temperature, less tendency to wick dry out and faster response can be anticipated. Thermal mass, with its tendency to maintain temperatures constant would help rather than impair the desired performance.

WICK BONDING TECHNIQUES

The wick most commonly used in tubular heat pipes is a stainless steel wire screen rolled into a cylinder and pushed inside the tube. As a result of the inherent springiness of the wire screen or with the help of spiral springs which are wound up tight, pushed into the tube and then permitted to unwind, the wick is held by mechanical means against the inside wall of the heat pipe. The heat pipe devices developed under this program are flat panels. Techniques inherently limited to application in tubular heat pipes could therefore not be applied.

High thermal conductance between the external heat pipe surface and the wick is one of the requirements for a low temperature gradient in a heat pipe. Good thermal contact between the wick and the internal heat pipe wall is therefore required. Obviously good heat transfer would be obtained by bonding of the capillary structures to the interior surfaces of the heat pipe cavities.

For water as heat pipe working fluid, braided and woven fabrics of glass or quartz fibers were selected as wicks. Testing as well as review of earlier work on wicks (reference 2 and 4) had shown the superior capillary pumping capability of these fabrics relative to other potential capillary materials.

The bonding of a capillary material to a solid substrate does, however, represent a problem. Bonding requires that the adhesive "wets" the surfaces to be bonded. A liquid adhesive, brought in contact with a capillary material which can be wetted by the adhesive will, due to capillary effects, penetrate into the capillaries. If this penetration proceeds unchecked, the capillary passages of the material will fill with adhesive. After setting of the adhesive, capillary capability will have been destroyed.

A requirement for the bonding of wicks to substrates is, therefore, that penetration into the fibrous structure of a wick material be limited to a small fraction of the total thickness of the wick, leaving the major part of the capillaries free to provide capillary pumping of the working fluid. Accomplishing this is made more difficult by the fact that, for reasons of good conductive heat transfer across the wick, it is necessary to make the wick as thin as a millimeter or less.

A potential solution of this problem, expressed in theoretical terms,is to dose the available adhesive such that an amount only sufficient for limited penetration is available at any area element of the wick-substrate interface.

Adhesives which dry to a tacky state prior to bringing the wick in contact with them are probably the simplest way of attaching a wick to a substrate. A typical example of such adhesives is ordinary rubber cement of the type used for repairing bicycle innertubes. In application of such a cement, it is required to let the solvent completely evaporate and then press the surfaces to be bonded against each other. In application to wick bonding, the rubber cement is applied to the solid substrate only and the solvent permitted to evaporate. Pressing a wick against the so prepared substrate provides a fairly good bond which is suitable for limited experimentation. However, the temperature up to which ordinary rubber cement is suitable is limited to approximately 50° C and the bond is not strong (reference 5).

Attempts to use a silicon rubber compound commercially available in tubes and of a similar consistency as ordinary rubber cement for this application, with the hope that the silicon rubber would permit increase in maximum temperature for this bond, failed. This was due to "poisoning" of the wick, resulting from the curing process of the silicon rubber compound. Apparently low molecular weight silicons are released in the process of curing. These silicons penetrate into the capillaries and effectively waterproof the wick. Several days after application, wicks attached by this technique had become so perfectly waterproofed that water droplets would run off an inclined wick surface (reference 5).

A good approach to accomplish minimum and controlled penetration into the wick capillaries is heat sealing of the wick to a substrate with the help of heat sealable film materials. Such materials include thermosetting and thermoplastic sheets of polyester resin, polyethylene-type heat sealable films and heat sealable forms of FEP-fluorocarbon film.

Heat sealing and bonding techniques are described in detail in references 5 and 6. A recommended technique is heat sealing with 0.05mm (2 mils) thick FEP-fluorocarbon film at 290° C (550° F), 2.11kg per cm^2 (30 psi) pressure and with 0.5 to 1.0 second dwell time.

The techniques for heat sealing of glass and quartz fiber fabrics to solid substrates could be extrapolated to applications of metal fiber wicks, including screens, metal felts and sintered metal sheets. Instead of a thermoplastic or thermosetting film, the substrate could be coated with a thin layer of solder. In order to provide a layer of uniform thickness, electroplating of the solder to the substrate is suggested. The capillary structure would then essentially be "heat sealed" to the solder-coated substrate by application of heat and pressure. This technique had no application to subject contract and has not been experimentally investigated. It is believed that it could be successfully applied where a requirement exists.

FLEXIBLE HEAT PIPES

Body temperature control in a space suit requires removal of heat from the human skin. In order to accomplish this, it was necessary to develop heat pipe devices which would be flexible to permit body motion and muscular, chest and abdominal expansion. These heat pipes would be in contact with the human skin and therefore located within the internal atmosphere environment of the space suit.

The enclosures of flexible heat pipes would unavoidably be somewhat permeable to the vapor of the working fluid. It was, therefore, required to select a working fluid which would be safe, not flammable and not be a source of atmosphere contamination. Reviewing potential working fluids, it became apparent that water would be the most desirable fluid for this application.

The use of water as a working fluid in a flexible heat pipe device, located within the pressurized internal environment of a space suit presents, however, a problem. At the range of temperatures applicable to space suit operation, i.e. at or below human body skin temperature (around 33° C), the pressure inside the heat pipe would be far below the pressure of the internal space suit environment.

Internal pressure in present space suits using an oxygen atmosphere is in the order of 0.25 kg per cm^2 (3.5 psia). Advanced space suit concepts will use gas mixtures at higher pressures. As the working fluid in a heat pipe is in a saturated condition, the internal pressure in a heat pipe at 33° C would be in the order of only 0.05 kg per cm^2 (0.7 psia). A flexible device with water as working fluid operating in this temperature range would, therefore, collapse as a result of external pressure.

The key problem to development of heat pipes for this application was, therefore, development of flexible, hollow devices which would not collapse when evacuated and exposed to external pressure.

A number of potential solutions for this problem were considered and experimentally investigated. They included internal support of flexible tubes by helical springs or flexible, plastic extrusions of cross-or star-shaped cross section and support of flat panels with a commercially available three-dimensional fabric. This fabric is made by weaving two different fibers along the length of the fabric and another across the width. One of the longitudinal fibers shrinks more when temperature-processed than the other. As a result, the fabric "puckers" and gets its third dimension. The resulting fabric is essentially a three-dimensional net or screen with large transversal pores and parallel longitudinal channels, like corrugated cardboard made of screen material. Various polymers, dimensions, degrees of flexibility and resistance against collapsing are available.

Application of the concept of a flexible, external shell, internally supported against collapsing required consideration of a second problem,presented by the location of the capillary structure. The wicks which are suitable as flexible capillary lining inside the flexible heat pipes are, in their capillary performance, sensitive to applied pressure which can actually "choke" a wick.

In the concept of supporting a flexible heat pipe by insertion of a helical spring into a flexible tube, the most desirable location for the wick would be, as always in a heat pipe, in contact with the tube wall. The wick would thereby be located between the tube and the helical spring. Under the effect of external atmosphere pressure, the tube would squeeze the wick hard against the spiral spring and "choking" of the wick will occur along the area of contact between the spring and the wick. Liquid would then have to travel by capillary action in a much longer spiral path from the condenser to the evaporator end of the tube, which would impair the pumping performance of the wick.

Heat pipes of this type were assembled from heat-shrinkable, irradiated polyolefin tubing, a braided tubular quartz fiber wick and a helical spring (figure 9). The braided quartz fiber tubing was put over the spring, then pushed inside the heat-shrinkable tubing. A rigid tube for evacuation and filling was fitted into one end and the assembly heated in order to shrink the shrinkable tubing tightly around the wick, spring and fill tube. The free end of the tubing was squeezed flat and heat sealed. Heat pipes of this type were tested, but performance was unsatisfactory, probably because of wick "choking" or because of high permeability of the heat-shrinkable polyolefin tubing to atmospheric air, or for both reasons.

The use of longitudinal supports consisting of a plastic rod with a cross-shaped cross section was an attempt to circumvent the wick choking effect of the helical spring. It was reasoned that in such a heat pipe, the tube-wick-internal support contact area would be lengthwise, essentially dividing the wick into parallel longitudinal strips and that this would not seriously interfere with the operation of the heat pipe (fig. 10).

Experimentation with these heat pipes was partially successful. Difficulties were experienced with inward leakage of atmospheric air through the polyolefin tubing.

An attempt to remedy this difficulty was made by use of irradiated FEP heat-shrinkable tubing. The irradiated FEP tubing performed better than the polyolefin tubing (fig. 11).

A parallel effort to develop internally-supported, flat panels using three-dimensional fabric for internal support was relatively more successful and the efforts for development of flexible, tubular devices were abandoned. It has, however, not been established that the two approaches for tubular, internally-supported heat pipes

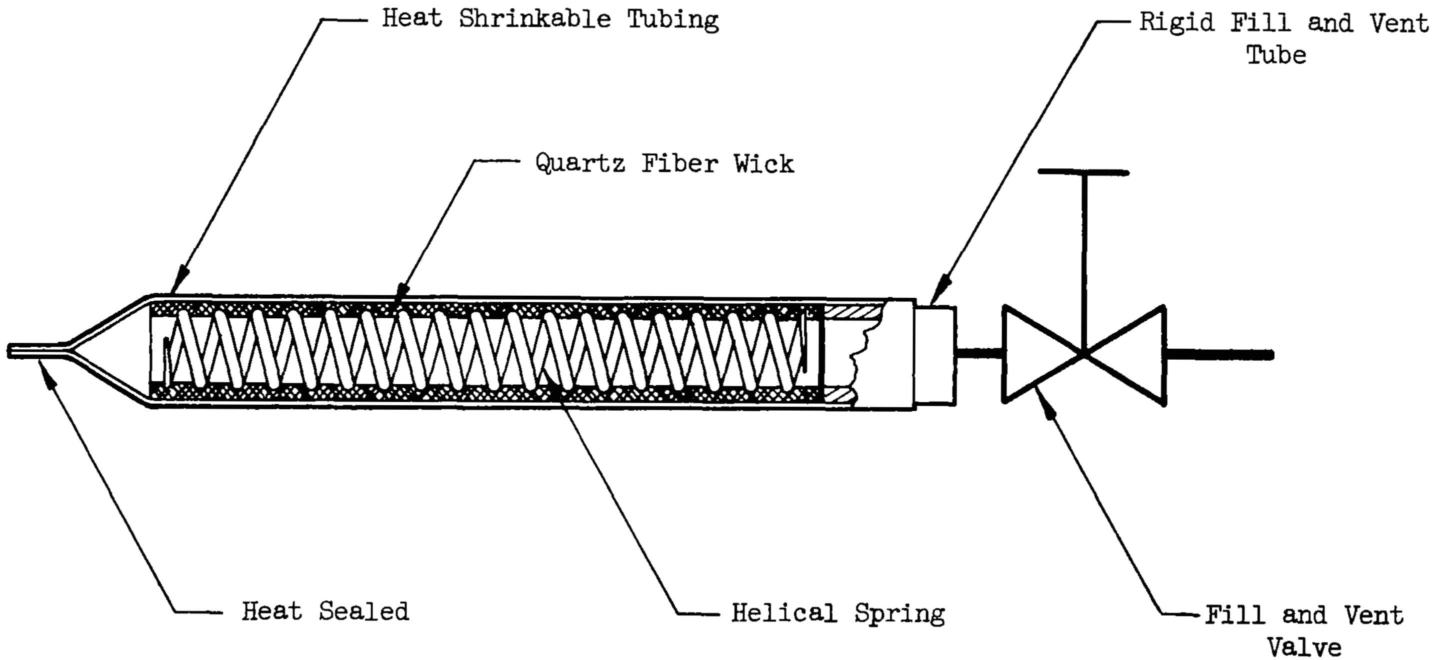

FIGURE 9 - FLEXIBLE HEAT PIPE - SPRING SUPPORTED

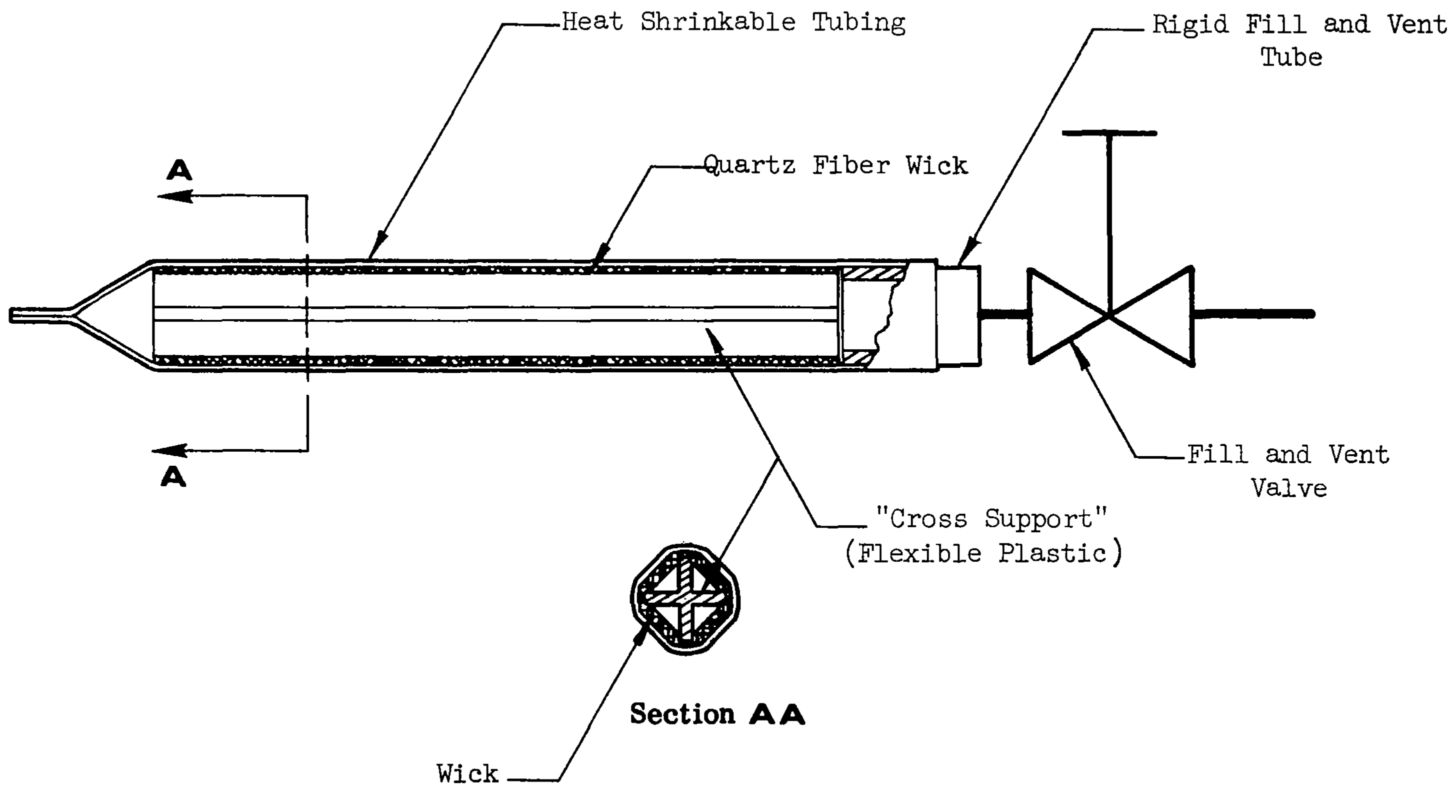

FIG. 10 - FLEXIBLE HEAT PIPE - CROSS-ROD SUPPORTED

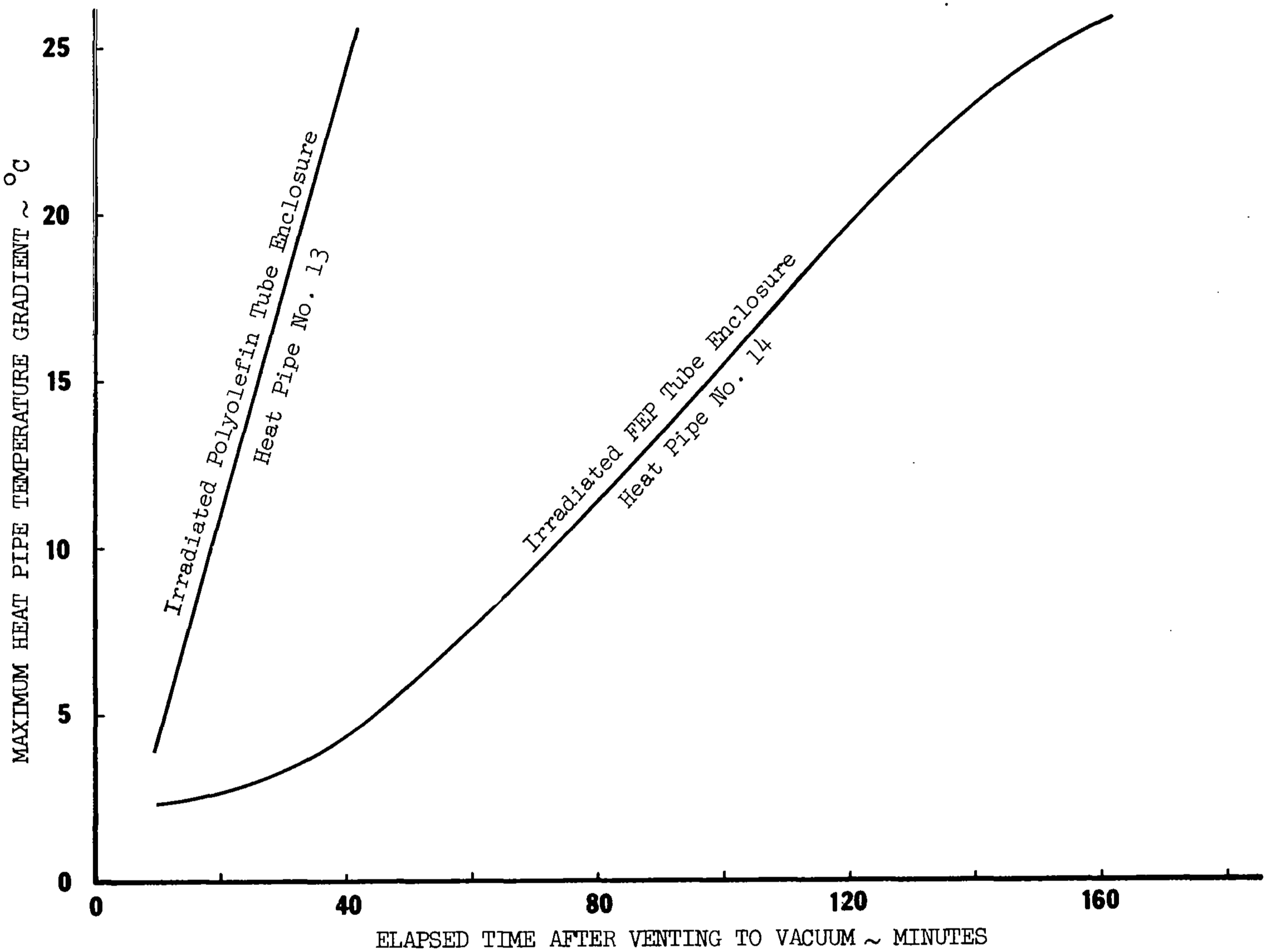

Figure 11 PERFORMANCE DETERIORATION OF FLEXIBLE HEAT PIPES
RESULTING FROM INWARD LEAKAGE OF AIR

described above are not feasible. Given selection or development of a less gas permeable tube for use as external enclosure, it is believed that the concept of longitudinal cross- or star-shaped support rods of flexible plastic would be successful for application to flexible, tubular heat pipes of small diameter.

Efforts were concentrated on developing flexible, flat panels with "three-dimensional" fabric support. The significance of inward leakage of atmospheric gas was at this time fully recognized and the fabricating techniques used for assembly of the flat panels permitted a wider choice of enclosure materials.

After preliminary experimentation with locations for the wick in flat heat pipe panels, it was decided that because the large flat area of panel devices would provide larger thermal contact areas with the human skin, a relaxation of the requirement to have the wicks in direct contact with the inside face of the external enclosure film would be acceptable.

The three-dimensional fabric used for internal support provided parallel, tubular channels. Strands of quartz fiber wick were inserted in alternating channels with in between channels left open as vapor passages (see fig. 12).

The wick and three-dimensional fabric assembly was then heat sealed inside a polymeric film enclosure. A small flanged tube machined from polyethylene rod was used as evacuation and fill connection for the heat pipe cavity and connected by heat sealing to the enclosure film. The polymer films applied included commercially available transparent films consisting of polyethylene-polyester, nylon-saran-polyester and aluminum-polymer laminates. With the exception of the aluminum-polymer laminates, all these films showed high inward leakage of atmospheric gases.

Identification of inward gas leakage as major difficulty and evaluation of relative merits of enclosure materials resulted from a series of test runs, using a test arrangement schematically shown in figure 13.

Heat was put into the evaporator end of the device by the electric resistance heaters and removed from the condenser end by the heat sink. Thermocouples were located along the heat pipe and temperature distribution was recorded. If after the initial evacuation and filling procedure, a significant gradient along the heat pipe appeared, the heat pipe was vented to vacuum for several

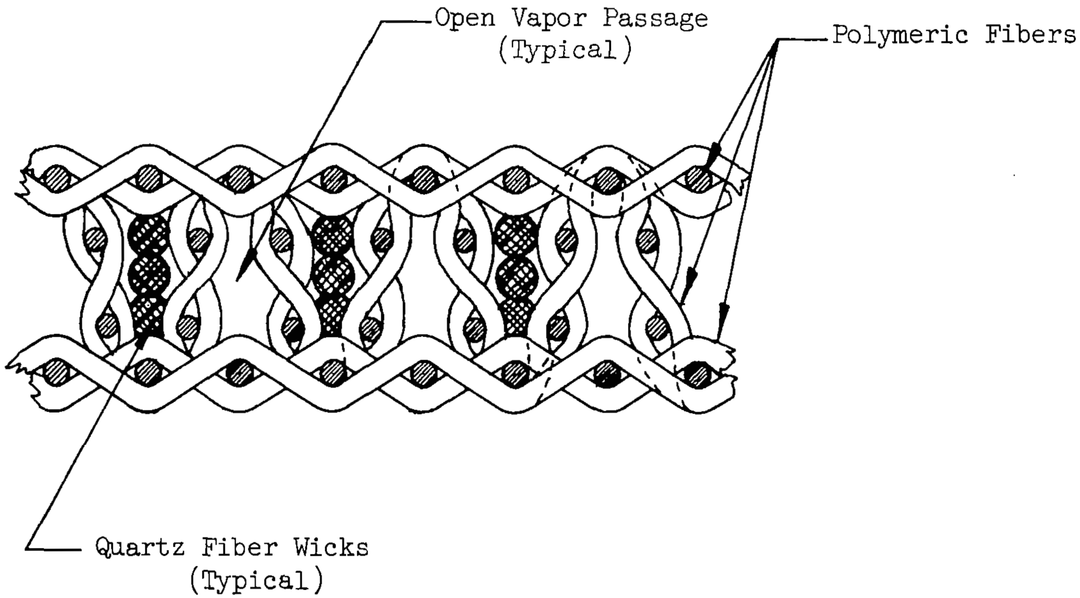

FIGURE 12 - SCHEMATIC CROSS SECTION THROUGH TYPICAL THREE-DIMENSIONAL FABRIC WITH WICKS IN ALTERNATING CHANNELS

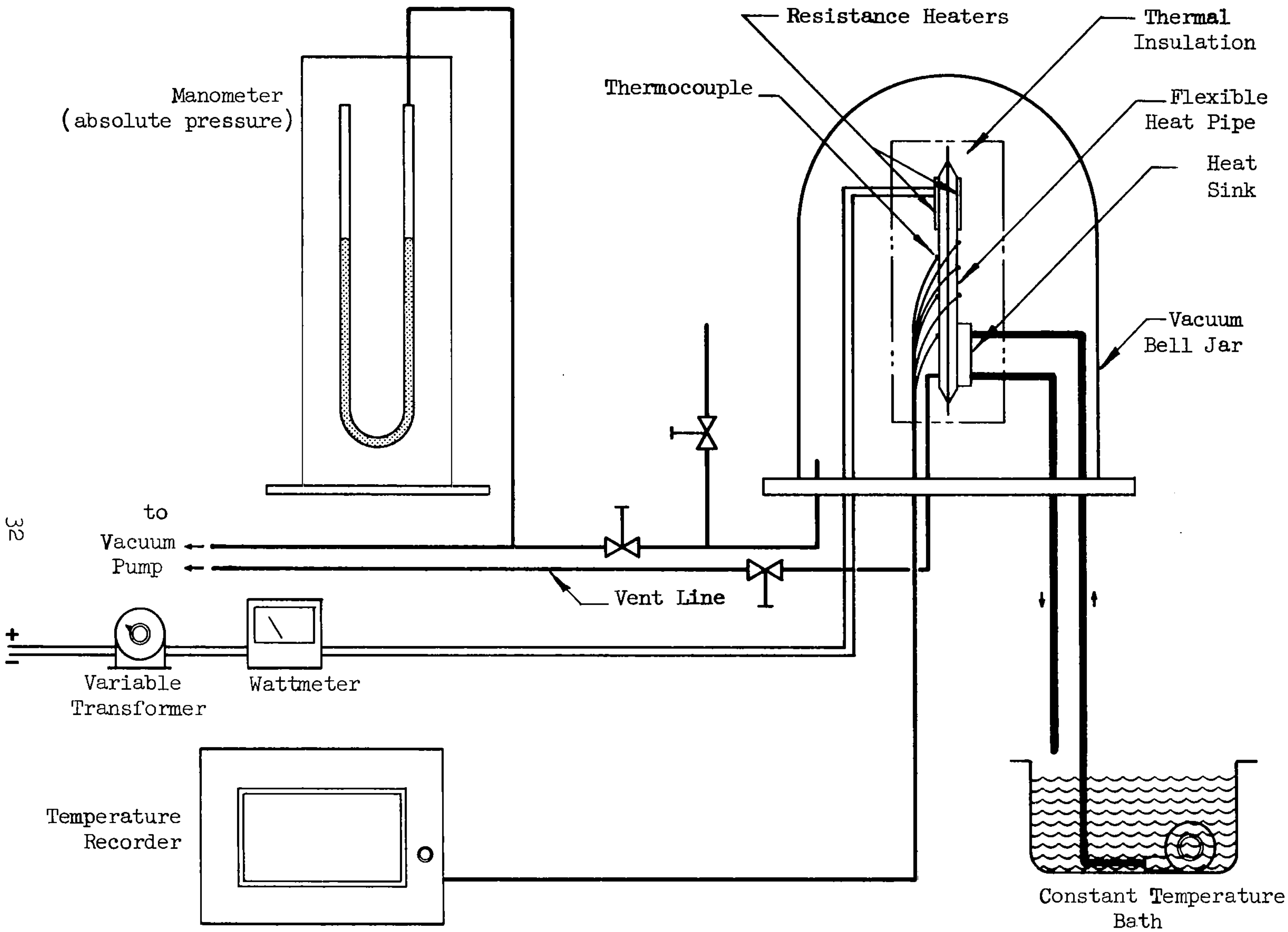

FIG. 13 SCHEMATIC OF EXPERIMENTAL ARRANGEMENT FOR PERFORMANCE TESTING OF FLEXIBLE HEAT PIPES

seconds. This removed noncondensable gas from the heat pipe and restored near isothermal operation of the device.

Tests were performed with the heat pipes exposed to the one atmosphere laboratory air environment or to an environment at an absolute air pressure close to internal heat pipe pressure.

The runs were performed for extended time periods. If deterioration in heat transport performance under exposure to one atmosphere occurred in a short time period and no or little deterioration was observed in the vacuum environment, it could be safely assumed that inward leakage of atmosphere gas was responsible for performance degradation.

A total of 18 flexible heat pipes were fabricated. Table I is a listing of the major characteristics of these heat pipes. Figures 14 and 15 show the typical performance of a flat heat pipe panel, No. 18, after exposure of 9 days to a reduced pressure environment and after an essentially equal time (10 days) to one atmosphere.

Up to the time of this writing, the best results were obtained with a commercially available aluminum laminate consisting of a layer of 0.025mm (1 mil) aluminum foil and a 0.076mm (3 mil) thick layer of a vendor proprietary heat sealable coating. Under exposure to one atmosphere, satisfactory operation in excess of two (2) days between venting was achieved. Obviously the problem of inward leakage of atmospheric gases has not been completely resolved. Two-day operation without measurable performance degradation, and restoration to normal operation by 15 seconds of venting to vacuum does however demonstrate essential feasibility of flexible heat pipe application.

An investigation into the probable causes of inward leakage, which included discussion with vendors and material specialists, led to the preliminary conclusion that this problem is due to the fact that the aluminum foil used in manufacturing of the laminate contains tiny pinholes which result from impurities in the metal. Such impurities will, when the material is rolled into a thin foil, in many places penetrate through the thickness of the foil and provide passages for gas molecules. The materials used in these activities were limited to materials commercially available. No material development effort was included in the present program. It is believed that better flexible laminates can be developed. They should be laminated from polymers of known low permeability to the specific gases in combination with metal foils rolled from high purity metals or of adequate thickness to reduce gas permeability.

Table I, Characteristics of Experimental Flexible Heat Pipes

Heat Pipe No.	Internal Support	Enclosure	Wick	Overall Dimensions, mm
1	Three-dimensional fabric	polyethylene-polyester film, heat sealable 0.11 mm thick	quartz fiber thread, 4 threads per channel in every channel	70 x 40.6 x 2.5
2	Three-dimensional fabric	polyethylene-polyester film, heat sealable 0.11 mm thick	quartz fiber thread, 4 threads per channel, 2 out of every 3 channels filled	72.5 x 34 x 2.5
3	Three-dimensional fabric	nylon-saran-polyethylene film, 0.09 mm thick, heat sealable	quartz fiber thread, 4 threads per channel, every other channel filled	68.5 x 37 x 2.5
4	Three-dimensional fabric	polyethylene-polyester film, 0.11 mm thick, heat sealable	quartz fiber thread, 4 threads per channel in every other channel. Quartz fiber batt used to fill voids between support and enclosure film	152 x 38 x 2.5
5	Three-dimensional fabric	polyethylene-polyester film, 0.11 mm thick, heat sealable	quartz fiber thread, 2 threads per channel, 2 out of every 3 channels	152 x 38 x 2.5

Table I, Characteristics of Experimental Flexible Heat Pipes (Cont'd)

Heat Pipe No.	Internal Support	Enclosure	Wick	Overall Dimensions, mm
6	Triple Layer of three-dimensional fabric of different pore sizes	polyethylene-polyester film, 0.11 mm thick, heat sealable	braided quartz fiber tubing, 0.7 mm o.d. One strand in every channel of outer support layers	148 x 36.5 x 4.8
7	Triple layer of three-dimensional fabric of different pore sizes	polyethylene-polyester film, 0.11 mm thick, heat sealable	braided quartz fiber tubing, 0.7 mm o.d. One strand in every channel of outer support layers	148 x 36.5 x 4.8
8	TFE rod with cross-shaped cross section	heat shrinkable irradiated polyolefin tubing	braided quartz fiber tubing, 9.5 mm o.d.	203 x 8.7 x 8.7
9	Helical spring	heat-shrinkable irradiated polyolefin tubing	braided quartz fiber tubing, 9.5 mm o.d.	89 x 12 dia.
10	Helical spring inside stainless steel wire screen	heat-shrinkable irradiated polyolefin tubing	braided quartz fiber tubing, 9.5 mm o.d.	114 x 8 dia.
11	Helical spring inside braided oxidized copper wire "shielding" sleeve	heat-shrinkable irradiated polyolefin tubing	braided quartz fiber tubing, 9.5 mm o.d.	82.5 x 12 dia.

Table I, Characteristics of Experimental Flexible Heat Pipes (Cont'd)

Heat Pipe No.	Internal Support	Enclosure	Wick	Overall Dimensions, mm
12	Three-dimensional fabric	polyester-saran-polyester film, heat sealable	quartz fiber thread, 2 threads per channel, 2 out of every 3 channels filled	152.5 x 38 x 2.5
13	Stainless steel rod of cross-shaped cross section (not flexible, for performance evaluation only)	heat-shrinkable, irradiated polyolefin tubing	braided quartz fiber tubing, 9.5 mm o.d.	203 x 8.9 x 8.9
14	Stainless steel rod of cross-shaped cross section (not flexible, for performance evaluation only)	heat-shrinkable, irradiated FEP tubing	braided quartz fiber tubing, 9.5 mm o.d.	203 x 9.6 x 9.6
15	Stainless steel rod of cross-shaped cross section (not flexible, for performance evaluation only)	heat-shrinkable, irradiated FEP tubing	braided quartz fiber tubing, 9.5 mm o.d.	241 x 9.6 x 9.6
16	Three-dimensional fabric	polyethylene-saran-polyester film, heat sealable	braided quartz fiber tubing, 0.7 mm o.d. 2 strands per channel every other channel	152 x 38.1 x 2.5

Table I, Characteristics of Experimental Flexible Heat Pipes (Cont'd)

Heat Pipe No.	Internal Support	Enclosure	Wick	Overall Dimensions, mm
17	TFE rod with cross-shaped cross sections	heat-shrinkable, irradiated FEP tubing	braided quartz fiber tubing, 9.5 mm o.d.	203 x 8.4 x 8.4
18	Three-dimensional fabric	0.012 mm polyester-0.012 mm aluminum-0.073 mm vendor proprietory heat sealing layer	braided quartz fiber tubing, 0.7 mm o.d. 2 strands per channel, every other channel	152 x 38.1 x 2.5

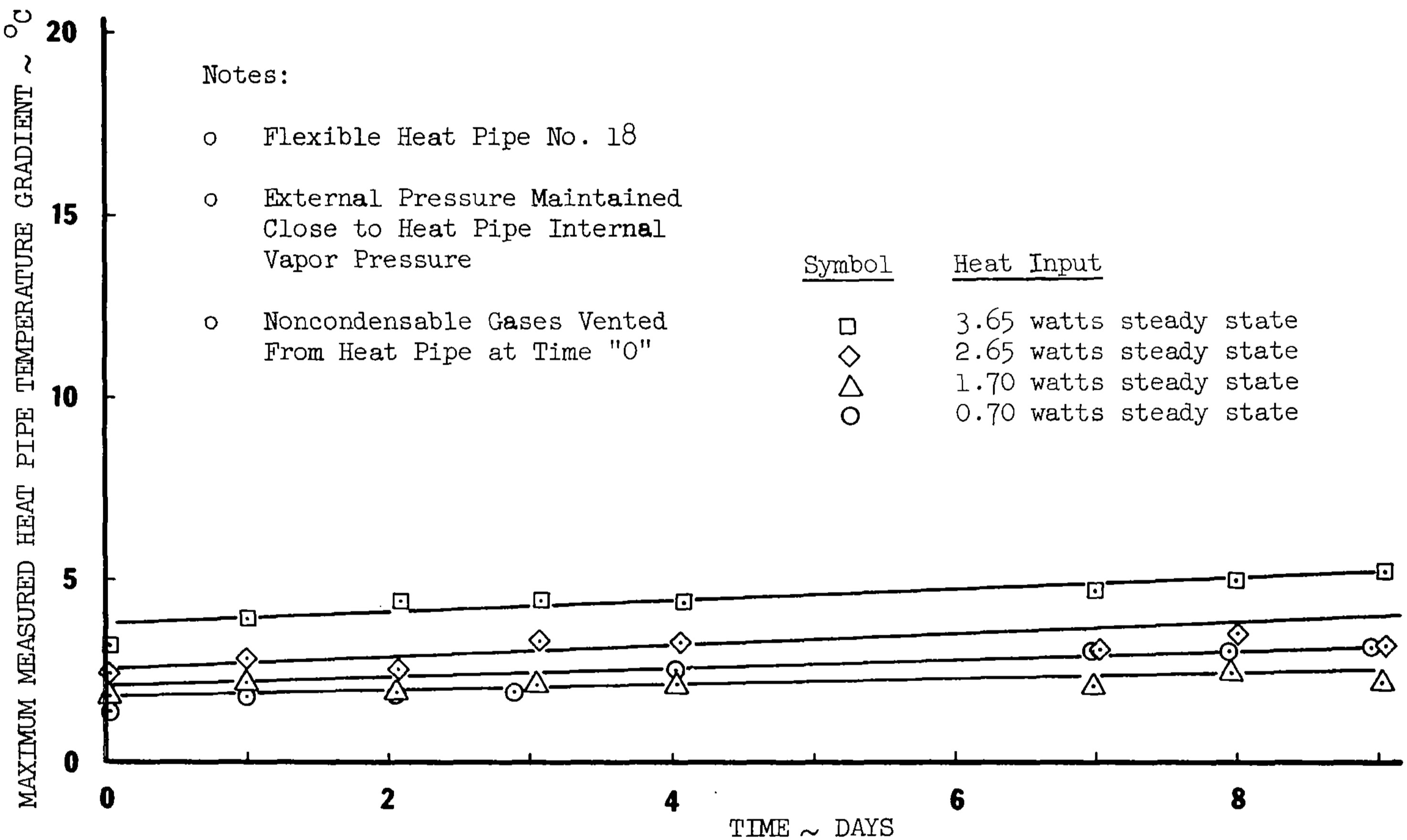

Figure 14 - FLEXIBLE HEAT PIPE PERFORMANCE vs TIME IN REDUCED PRESSURE ENVIRONMENT

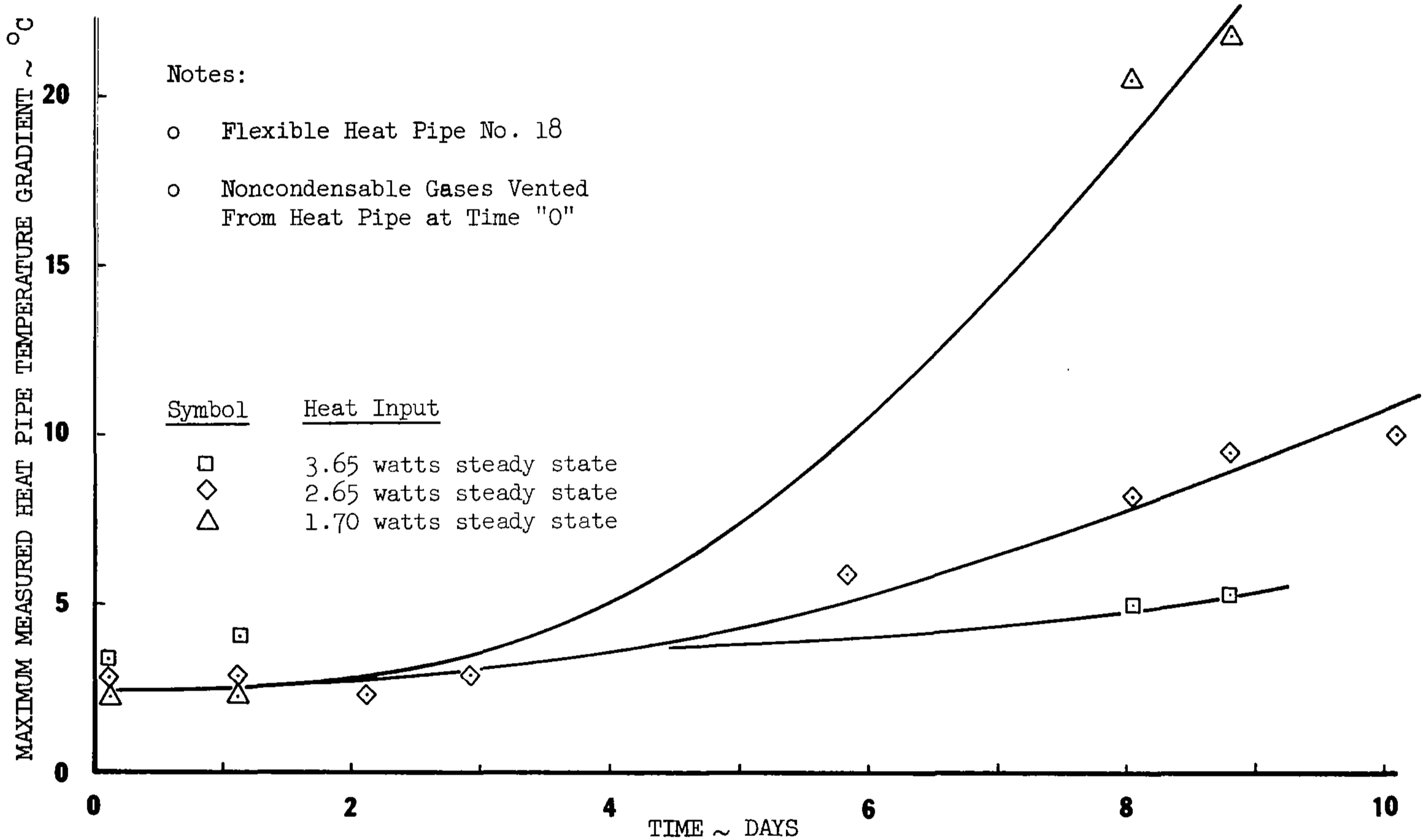

Figure 15 - FLEXIBLE HEAT PIPE PERFORMANCE vs TIME IN ONE ATMOSPHERE PRESSURE ENVIRONMENT

The flexible panels developed were combined with a flat, rigid heat pipe panel of stainless steel into a two chamber heat pipe-thermal switch assembly shown in figure 16.

This two chamber heat pipe-thermal switch demonstrated one concept of human body temperature control in a space suit. Under this concept, the two chambers would be located on both sides of the suit pressure shell and thermally bypass a layer of highly effective thermal insulation. The flexible chamber (on top in figure 16) would contact the skin and absorb heat, the rigid chamber would be at the outside surface of the space suit and emit heat by radiation. The thermal switching capability of the device would adjust heat flow, bypassing the thermal insulation, to the metabolic requirements of the astronaut.

The heat pipe assembly shown in figure 16 includes an electric resistance heater on top and a heat sink heat exchanger at the bottom of the device, which simulate the human body heat production and heat emission to space. The assembly is extensively instrumented with thermocouples. Its development, construction and performance are fully discussed in reference 1.

THE HEAT PIPE FREEZE UP PROBLEM

For a given working fluid, operation of a heat pipe is limited to the temperature band between the freezing point and the critical temperature of the working fluid. In the operation of most heat pipes, the operating temperature is well below the critical temperature. The possibility of freezing of the working fluid in the heat pipe is, on the other hand, a distinct possibility.

In the design of a heat pipe system, it would obviously be desirable to select a working fluid with a freezing point safely below the anticipated heat pipe operating temperature. This requirement, while desirable may, however, be difficult to satisfy. In the case of the application to a space suit, the selection of the working fluid, i.e. water, was dictated by considerations of safety and performance. In other applications, even if safety requirements are less stringent, heat transmission performance may still favor application of a working fluid with a solidification point higher than the lowest temperature which may be encountered over the full range of operating conditions of a heat pipe system.

The efficient heat pipe working fluids are those which combine high heat of vaporization, surface tension and liquid density with low viscosity (ref. 2). Evaluation of heat pipe working fluids

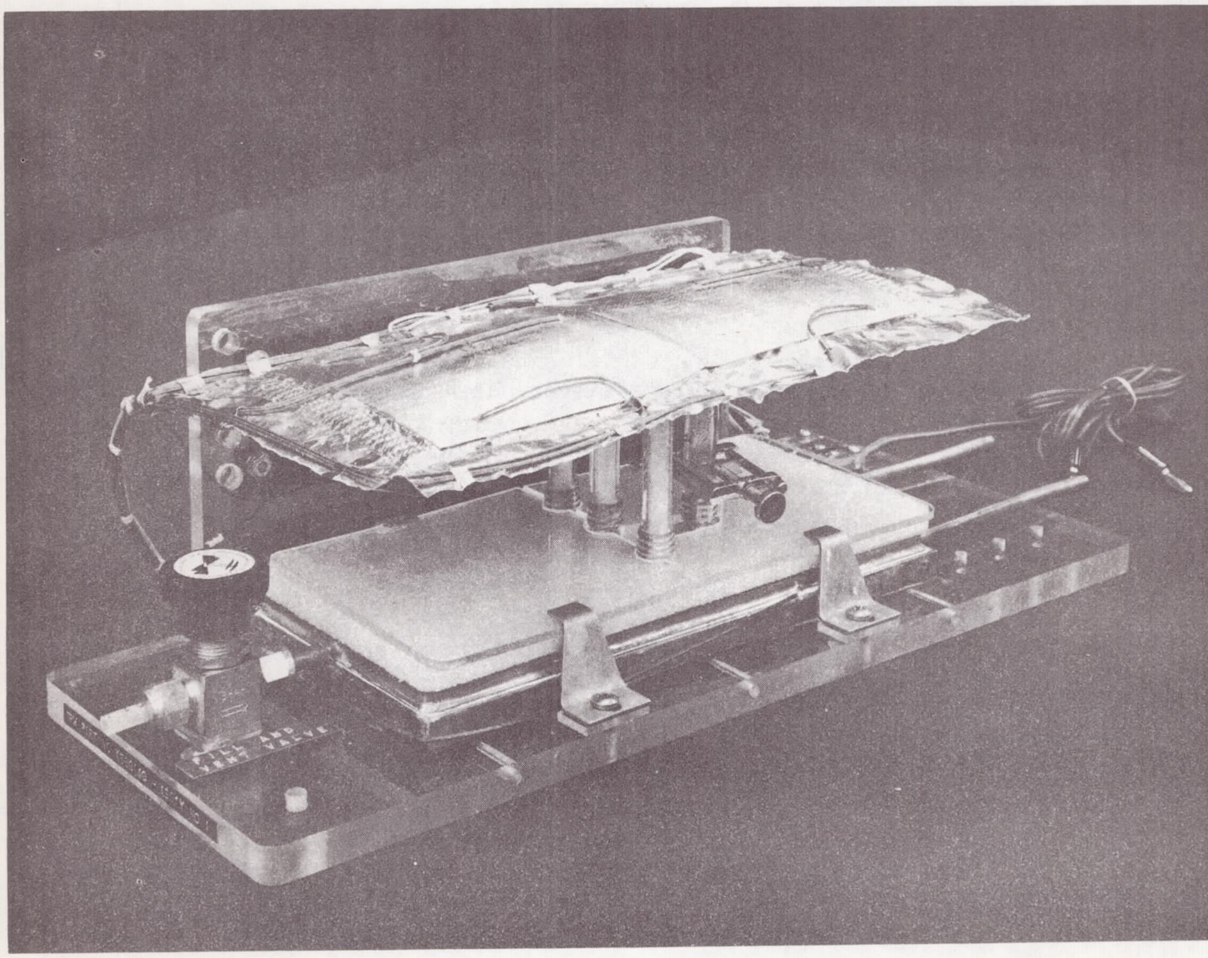

Figure 16 - THERMAL SWITCH INCLUDING FLEXIBLE PANEL (ON TOP)

on this basis shows that low melting point metals (sodium, potassium, etc.) and water top the list, while ammonia, the alcohols and freons are significantly less efficient. The efficient working fluids are fluids with relatively high freezing points.

The temperature at which a heat pipe operates is dependent on rate of heat flow into and out of the heat pipe. For a steady state condition, in-flow of heat must equal out-flow. The vapor temperature in most heat pipes is very nearly isothermal. Heat flow in and out of the heat pipe will be related to the thermal resistances between the heat source and the vapor, and the vapor and the heat sink. These resistances are very nearly constant and heat flow and vapor temperature will therefore change when heat sink and/or heat source temperature change.

A heat pipe could operate between a heat source at some relatively high temperature and a heat sink below the solidification temperature of the working fluid, with the operating temperature of the heat pipe maintained well above the solidification point by the heat input from the heat source. The heat source could be heat dissipating equipment, which is temperature controlled via the heat pipe by heat radiation to space. If the equipment would be temporarily shut down, its temperature would drop, no heat would be supplied to the heat pipe and heat pipe temperature would approach heat sink temperature. The working fluid would freeze and heat transmission, by the heat pipe evaporating-condensing cycle, cease. This, in itself would not be objectionable, as long as no heat is generated by the equipment. The freezing of the working fluid would actually inhibit excessive temperature drop of the equipment.

If after such a zero power period power is again supplied, the heat pipe will however not readily start to transmit heat.

A low temperature at the condenser of a heat pipe will be accompanied by a drop in vapor pressure. This will cause continuing evaporation of working fluid in the evaporator. The vapor will flow to and condense and freeze or sublimate at the coldest part of the heat pipe, i.e. the condenser. Wick dry out at the evaporator will occur, with all the working fluid accumulated and frozen at the condenser.

The heat pipe can not restart when heat input in the evaporator is increased. A temperature rise at the condenser is required in order to melt the frozen working fluid and to get the wick to pump the liquid to the evaporator before the heat pipe will start. No working fluid is, however, available at the evaporator to transmit heat to the condenser and melt the solidified working fluid. Figure 17 shows the temperature history and profile of a 60 cm long stainless steel, tubular heat pipe with water as working fluid, after freeze up had occurred. At time "zero" heat input into the evaporator was increased. The figure shows that the temperature at the evaporator rose over a period of 11 minutes to almost 200° C, while the temperature at the condenser did not measurably change and stayed at the freezing point of water.

Concepts were generated to either prevent freeze up, or enable a heat pipe to restart in a reasonably short time span after it has been frozen, when heat input into the evaporator is increased.

To prevent freezing of water, additives providing freezing point depression have been used in many applications. Anti-freeze additives including alcohols and ethylene glycol are commonly used in automobiles. In application to a heat pipe where phase change occurs in the working fluid, the applicability of mixtures is, however, questionable.

Assuming that the working fluid consists of a mixture of methyl alcohol and water, the higher vapor pressure of the methyl alcohol would cause fractioning of the mixture upon evaporation. The vapor phase of the fluid would be a mixture of vapors at the volume ratio of the saturation pressures at the temperature of the two (2) components of the mixture. Methyl alcohol with its higher vapor pressure would be the larger fraction of the vapor mixture. In the wick, liquid water would be the major component of the mixture.

We would therefore have a heat pipe in which most of the heat transfer would be performed by the methyl alcohol, which is not a very efficient working fluid, with a relatively large fraction of the liquid water remaining in the wick, where it would freeze at low temperatures making the wick ineffective for the pumping of either fluid.

The situation would be different with a freezing point depressant with a vapor pressure which is azeotropic with water. Such an additive would be present at a constant mixture ratio in the liquid and the vapor phase and the mixture would essentially function as one fluid.

A literature search for fluids with vapor pressure characteristics closely resembling those of water was initiated. This search uncovered the fact that normal propyl alcohol (propanol-1) has a vapor pressure curve almost identical to that of water above the freezing point of water, yet has a freezing point of -127° C. An

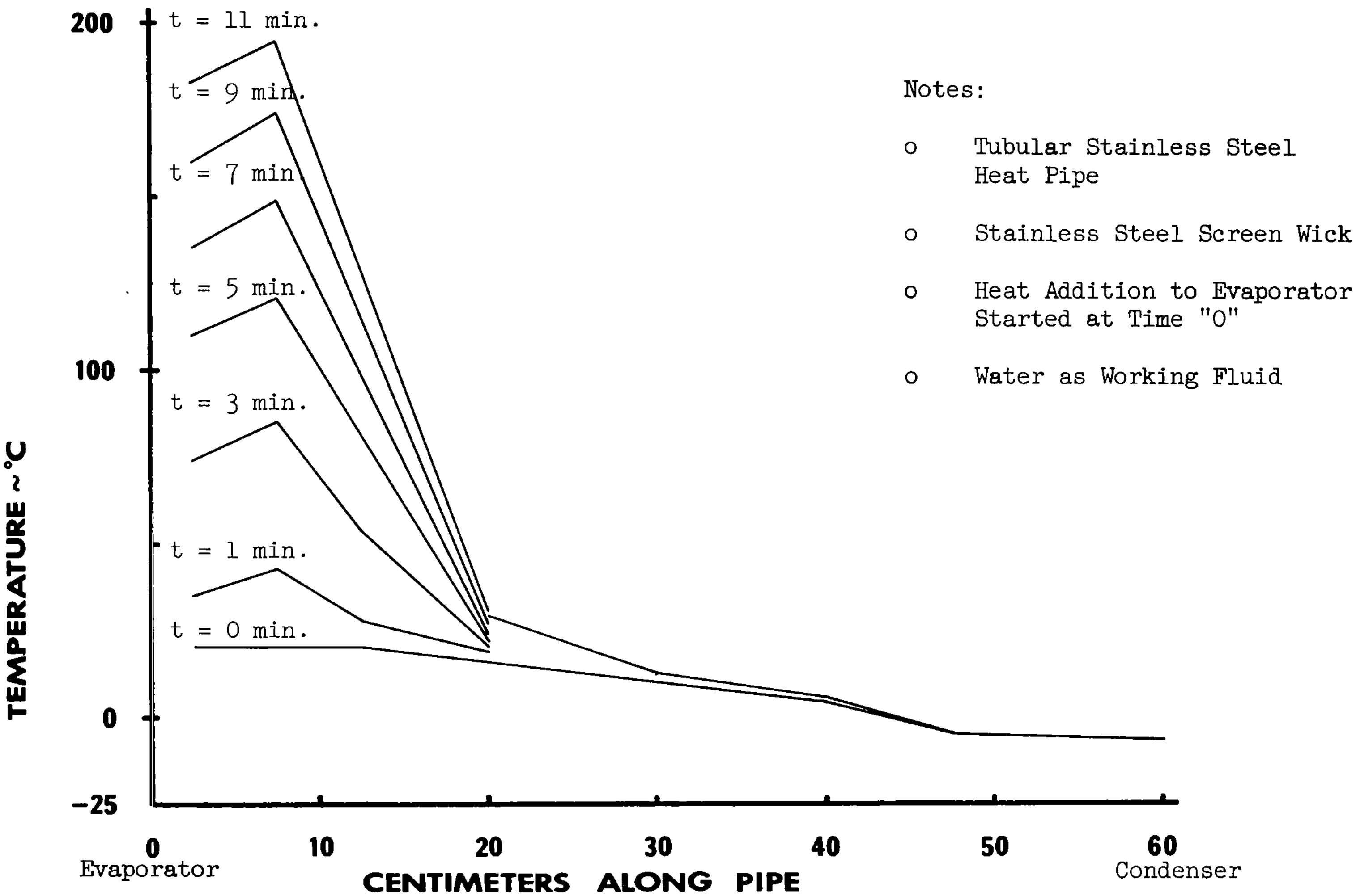

Figure 17 - TEMPERATURE PROFILES ALONG A FROZEN HEAT PIPE AT DIFFERENT TIME POINTS FOLLOWING INITIATION OF HEAT INPUT TO EVAPORATOR

investigation was therefore started to determine whether propanol-1 could be effectively used as a freezing point depressant for water. A literature search for the freezing points of propanol-1 and water mixtures provided data only to a weight fraction of 32.5 percent propanol-1, at which mixture ratio the freezing point is -10.5° C (ref. 7). In order to determine whether this approach is promising, it was necessary to extend the available data and determine existence of a eutectic ratio. For ease of testing, melting point rather than freezing point was measured. A simple test apparatus was developed. This apparatus and the test method are described in reference 6.

In figure 18 freezing points for mixtures of water and propanol-1 are plotted as a function of mixture ratio. No eutectic ratio was indicated and a significant freezing point depression is not achieved until the weight concentration of propanol-1 is in the range of about 90 percent. Propanol-1 is obviously not an effective freezing point depressant for low temperatures. It may have limited applicability to, maybe, -10° C.

A limited literature search for other fluids with the desired vapor pressure vs temperature characteristics did not lead to any promising freezing point depressants.

Another approach would not attempt to prevent freezing.

As stated above, the reason that a heat pipe after freezing will not readily restart is that all of the working fluid has evaporated in the warm regions and the vapor condensed and frozen or sublimated in the cold regions of the heat pipe. If the heat input into the warm regions is then increased, only heat conductance in the wick and structure of the heat pipe will transfer heat to the frozen working fluid. As figure 17 shows, this heat conductance is obviously inadequate.

Heat pipes are however, devices very capable to transfer heat at high rates. Supposing that water or a metal has been selected as working fluid for the heat pipe because of its superior heat transmission capability, an auxiliary heat pipe with a less effective but lower freezing point working fluid could be applied to transfer enough heat from the evaporator to the condensor to provide restart after freeze up in a reasonable time.

The auxiliary heat pipe would be in intimate thermal contact with the primary heat pipe. It could, for example, be inside the primary heat pipe and in direct contact with the wick over all its length. The primary heat pipe would be designed to carry the major

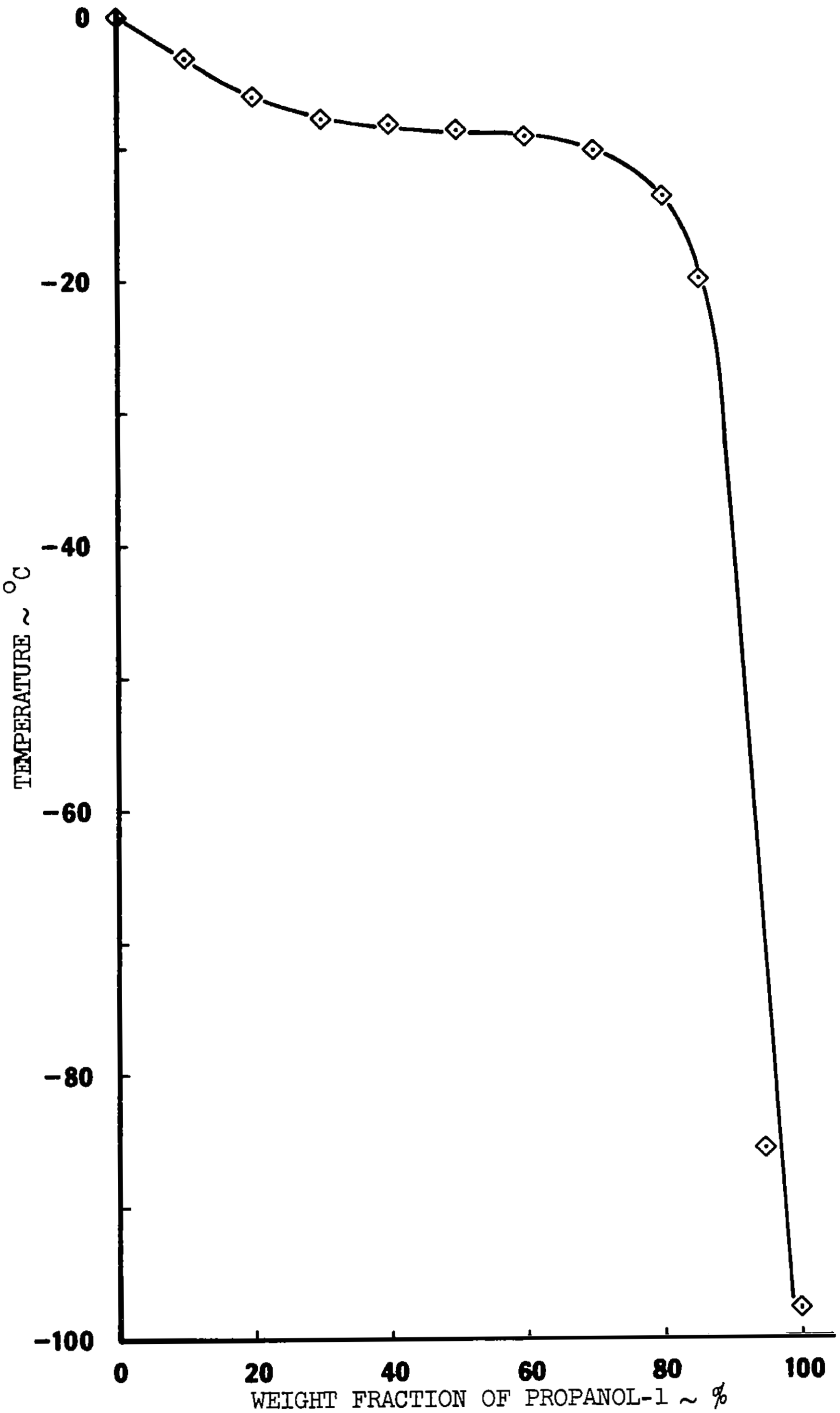

Figure 18 - FREEZING TEMPERATURES OF PROPANOL-1/WATER MIXTURES

heat load by use of an efficient working fluid. The secondary or auxiliary heat pipe would use a working fluid of low freezing point, and although its heat carrying capability would be quite limited, it would suffice to transfer enough heat to the condenser to melt the working fluid when a heat input into the evaporator is increased.

A device developed as multi-purpose test fixture for heat pipe experimentation was used to demonstrate feasibility of this approach. This device consists of a glass tube as shown in figure 19. It has a wick carrier and wick internally mounted. The wick carrier is equipped at one end with an electric resistance heater as heat source and at the other end with a heat sink - heat exchanger, through which a coolant fluid can be circulated. Thermocouples are provided in contact with the wick surface. A heat pipe using methyl alcohol as working fluid was used as auxiliary heat pipe. The auxiliary heat pipe was made of $\frac{1}{2}$" stainless steel tube with two (2) layers of 105 mesh stainless steel screen as wick.

Two identical test runs with water as the primary working fluid were performed. In one of these test runs, the auxiliary heat pipe was attached to the top of the water carrying wick, while in the other test run the auxiliary heat pipe was not used.

The tests were performed by circulating a coolant fluid at below freezing temperature through the heat exchanger, with a heat input of 2 W into the electric resistance heater. The heat pipe was operated until a large temperature gradient between evaporator and condenser appeared. This was an indication that the heat pipe had frozen and the working fluid, as anticipated had collected in the cold part of the device. The coolant fluid supply was then turned off and the power input into the heat pipe increased to 15 W. Figures 20 and 21 show the temperature vs time histories of these two runs with and without the auxiliary heat pipe. From the figures, it appears that the auxiliary heat pipe was capable to significantly accelerate the restart of the device.

This test is considered only as a preliminary indication of feasibility. As stated above, the basic heat pipe used for this experimentation is a "multi-purpose device" and was not specifically designed for this experiment. Also the auxiliary heat pipe was not designed to provide best heat transfer to the cold region.

The experiment demonstrated, however, the practicability of this concept of restarting of frozen heat pipes. Further research in this area is therefore recommended.

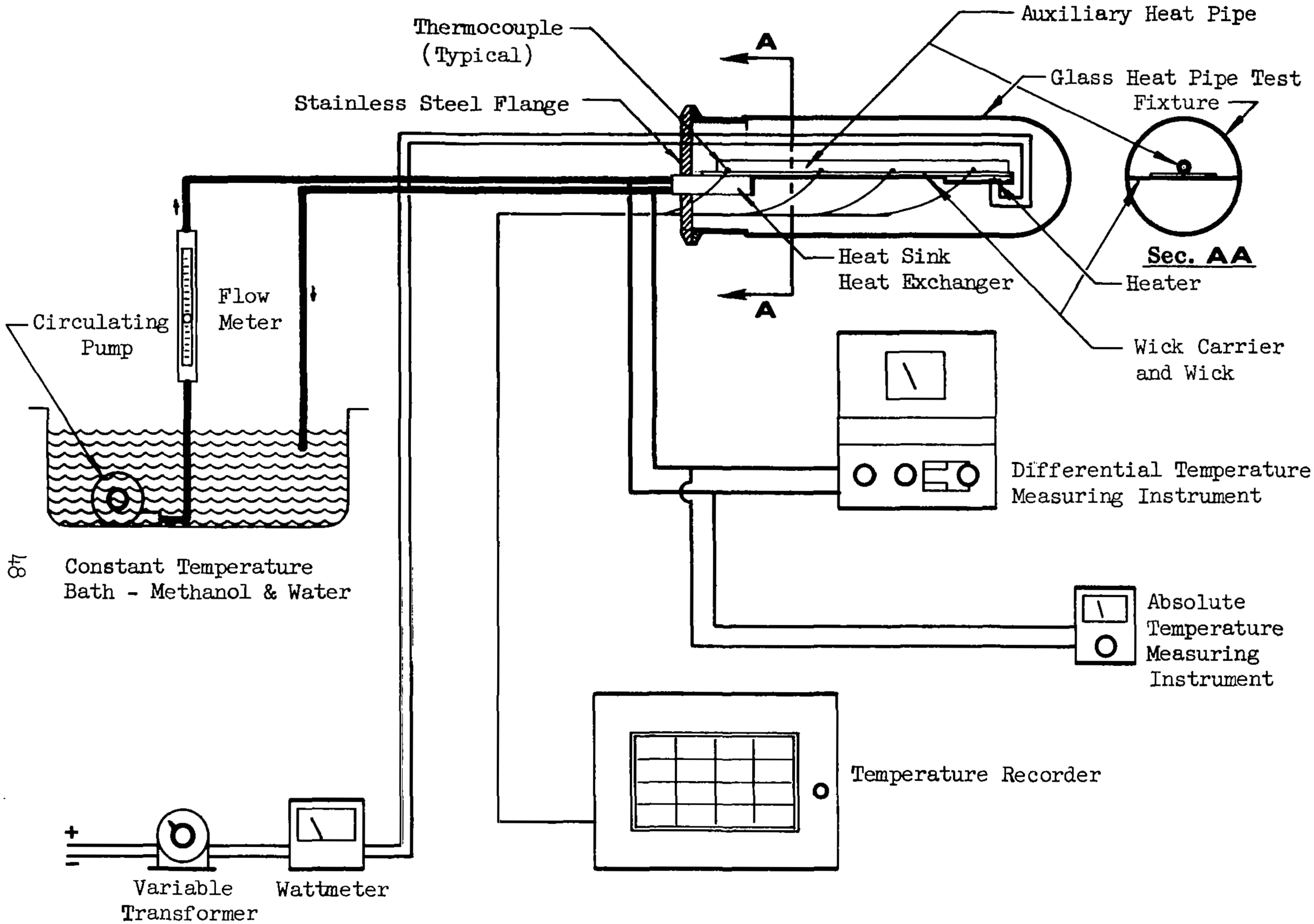

FIGURE 19 - SCHEMATIC OF EXPERIMENTAL ARRANGEMENT - RESTARTING OF FROZEN HEAT PIPE

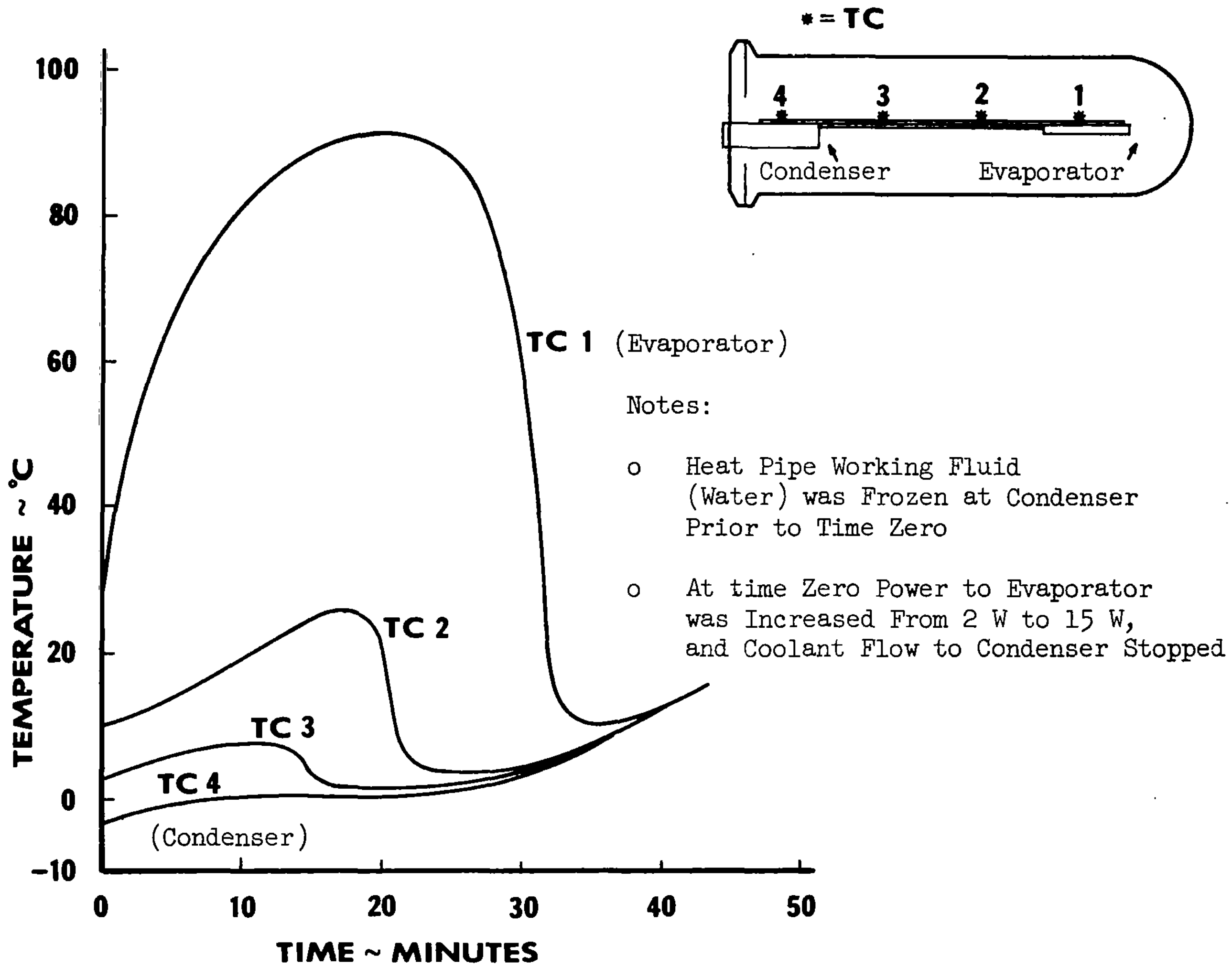

Figure 20 - RESTART OF FROZEN HEAT PIPE WITH AUXILIARY "START UP" HEAT PIPE

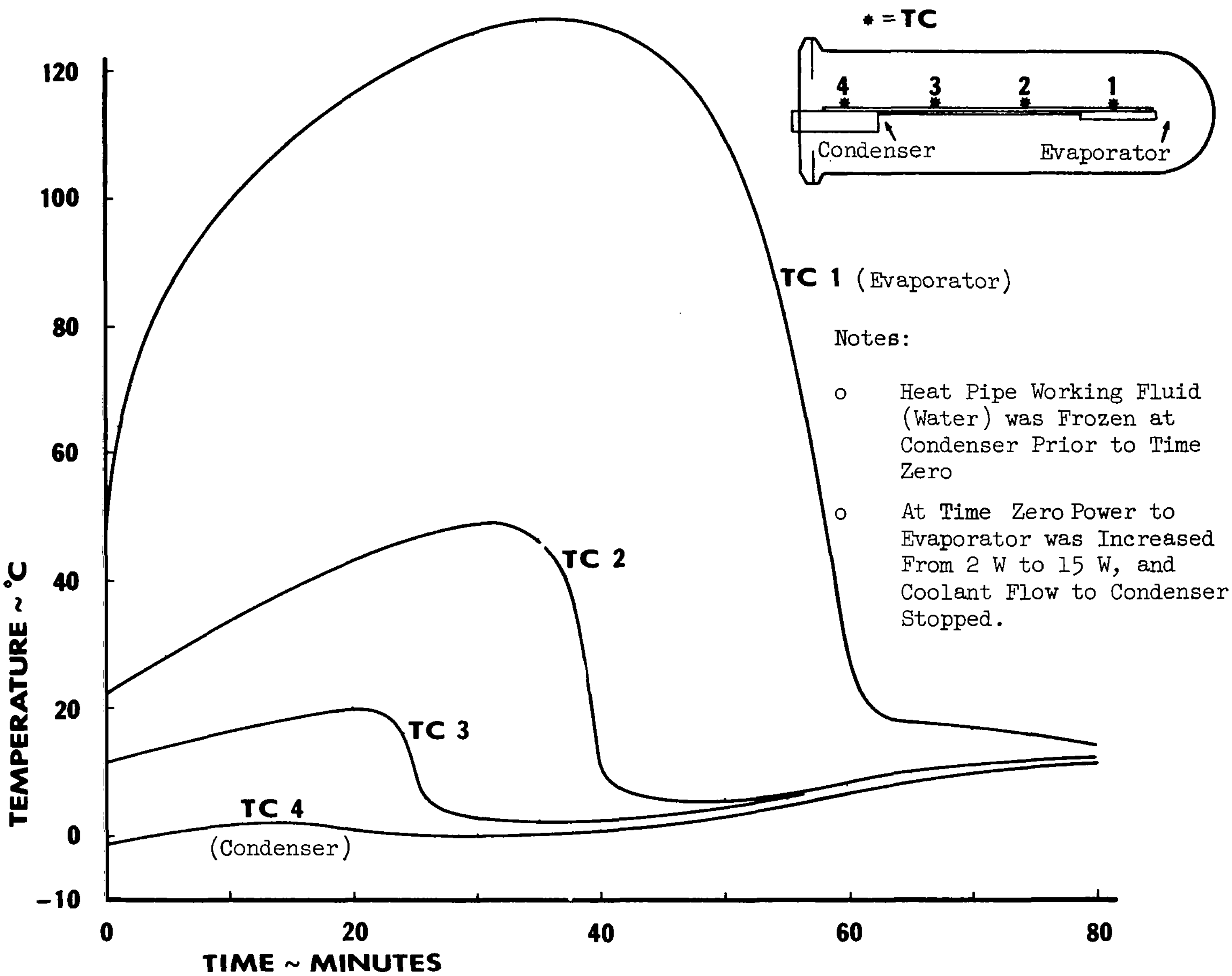

Figure 21 - RESTART OF FROZEN HEAT PIPE WITHOUT AUXILIARY "START UP" HEAT PIPE

REFERENCES

1. Shlosinger, A.P., "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits, Final Second Phase Report", NASA CR-73270, November, 1968.

2. Shlosinger, A.P., "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits", NASA CR-73168, September, 1967.

3. Marcus, B.D., "On the Operation of Heat Pipes", TRW Systems, Report No. 9895-6001-TU-000, May, 1965.

4. Shlosinger, A.P., Woo, W., Cafaro, C., Bentilla, E.W., "Technology Study of Passive Control of Humidity in Space Suits", NASA CR-69098, September, 1965.

5. Shlosinger, A.P., Jacobi, E.F., "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits, Material Research Report - First Year", NASA CR-73169, September, 1967.

6. Woo, W., "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits, Material Research Report - Second Year", NASA CR-73271, November, 1968.

7. International Critical Tables of Numerical Data, Physics, Chemistry and Technology, Volume IV, p. 262, McGraw-Hill Book Company, New York, 1933.