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INVESTIGATION OF A VARIABLE CONDUCTANCE HEAT PIPE

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Naval Postgraduate School
Monterey, California

March 1975

NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

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by

Theodore Phillip Naydan

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Thesis Advisor: M. Kelleher

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Investigation of a
Variable Conductance Heat Pipe

by

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ABSTRACT

A variable conductance, two inch diameter, sixty inch long heat pipe was designed and constructed. The performance characteristics of the heat pipe while operating in both the conventional and variable conductance modes were studied. In particular, the ramifications of utilizing a non-condensible gas which was more than twice as heavy as the working fluid were closely observed with the heat pipe being oriented both horizontally and vertically. Power inputs to the heat pipe were varied from twenty-five to one hundred fifty watts. Methanol was selected for use as the working fluid and krypton was used as the non-condensible gas. Condenser temperature profiles and liquid crystal photographs are presented for the various operating modes.

TABLE OF CONTENTS

LIST OF FIGURES

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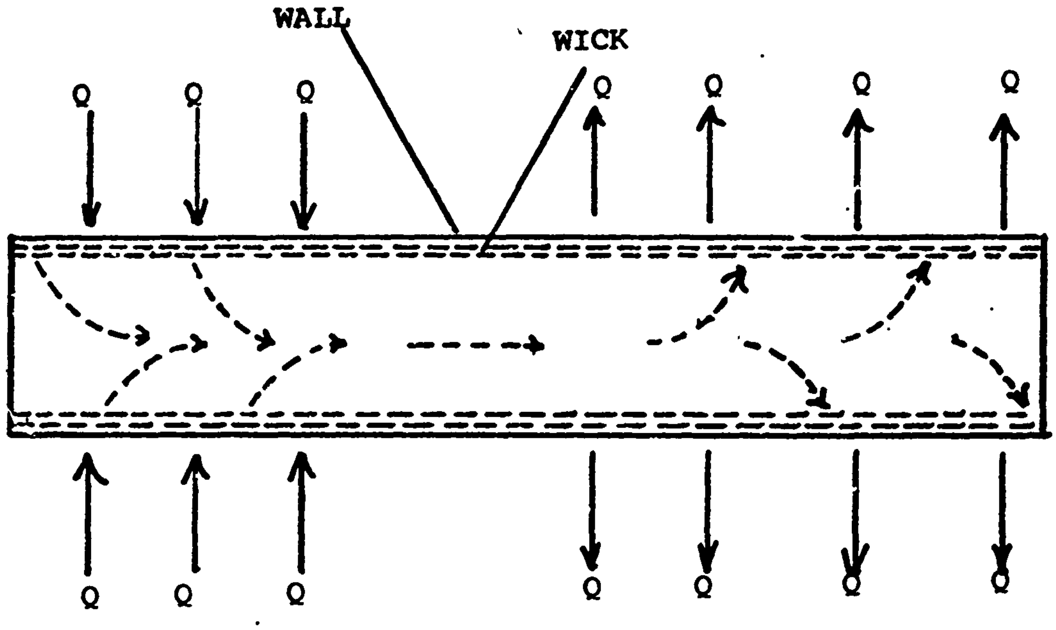
I. INTRODUCTION

A. BACKGROUND

A heat pipe is a device capable of transferring large quantities of heat under nearly isothermal operating conditions. The mechanism which enables this efficient transfer of heat is the evaporation of a working fluid at one end of the heat pipe and the condensation of the working fluid at the other end. The condensate is transported back to the evaporative section by means of capillary action through some sort of a wick inside the pipe. Previous experiments have demonstrated the heat pipe's capability to conduct heat one thousand to ten thousand times more effectively than the same size solid rod [Ref. 1].

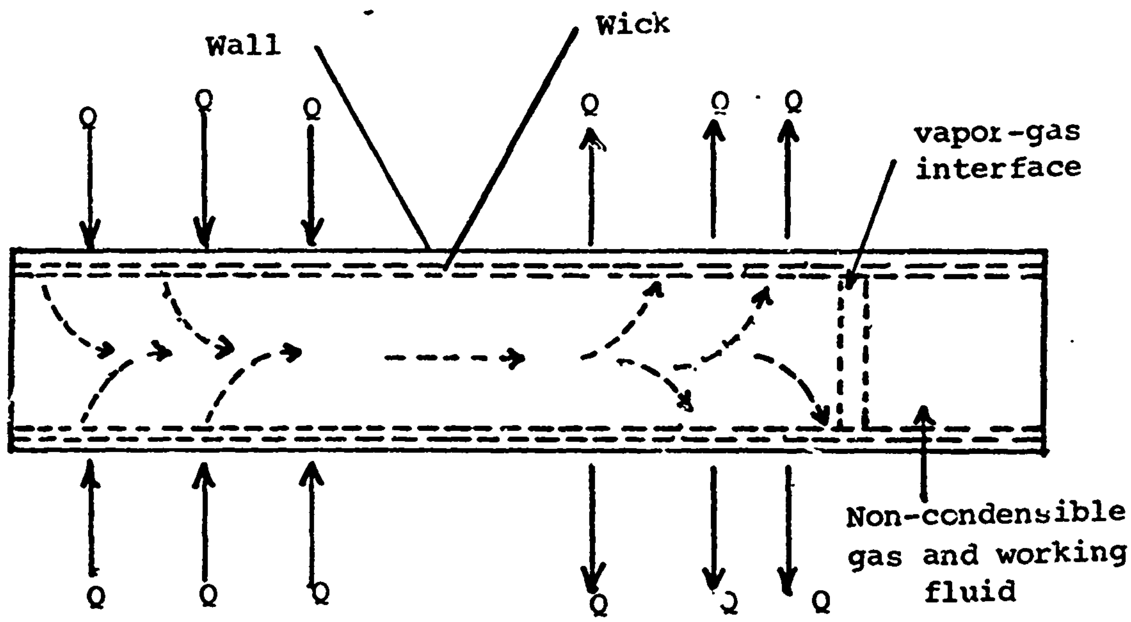
B. CONVENTIONAL HEAT PIPE THEORY

There is no physical difference between the conventional and the variable conductance heat pipe as used in this research. The only structural members required for either are the pipe itself and some wicking material along the inside surface of the pipe capable of moving the desired working fluid by capillary action. The active agent is the working fluid. While the heat pipe is operating, fluid is constantly being evaporated at the heated end, or evaporator, and condensed at the opposite or condenser end (see Fig. 1). The effectiveness of the heat pipe in transferring heat is due



Conventional Heat Pipe Operating Diagram

FIGURE 1



Variable Conductance Heat Pipe Operating Diagram

FIGURE 2

to the heat of vaporization being absorbed at the evaporator and rejected at the condenser continuously.

In terms of actual operation, the heat added at the evaporator vaporizes the working fluid in the wick in the evaporator. This vapor then travels through the center of the pipe to the condenser where it is condensed. The liquid then returns to the evaporator through the wick. These phase changes cause the heat pipe to be nearly isothermal along its length.

C. VARIABLE CONDUCTANCE HEAT PIPE THEORY

The major difference between a conventional and a gas loaded variable conductance heat pipe is that the latter has introduced into it a non-condensible gas in addition to the working fluid. The heat pipe operates as previously described, except that the non-condensible gas is convected along with the working fluid to the condenser where, since it is non-condensible, it remains to form a plug (see Fig. 2).

Eventually, the concentration of non-condensible gas at the condenser becomes very large. This causes a change in temperature towards the ambient as the condensation rate is decreased. The partial pressure of the working fluid, and therefore its concentration in a mixture at constant pressure, is directly related to its temperature. Thus, the concentration of the working fluid present in conjunction with the gas plug will decrease as the distance from the evaporator increases. This has the effect of shortening the effective condenser length of the pipe and thereby reducing the area available to transfer heat.

As the amount of heat applied to the pipe increases, the temperature in the evaporator increases, thereby causing an increase in the partial pressure of the working fluid and in the total pressure within the heat pipe. Due to the increase in the partial pressure of the working fluid, the volume occupied by the non-condensible gas decreases. This has the effect of increasing the heat transfer area and the conductance of the heat pipe. This characteristic enables a smaller operating temperature variation for a variable conductance heat pipe than for a conventional heat pipe for the same change in heat input.

Two explanations of the operating characteristics of the variable conductance heat pipe have been proferred. The first, developed by W. Bienert [Ref. 2] assumes an extremely narrow and flat vapor-gas interface perpendicular to the axis of the pipe. This theory, called the 'flat front theory', neglects axial conduction in the heat pipe walls, vapor pressure drop from the evaporator towards the condenser, and any diffusion between the working fluid and non-condensible gas.

The second, and more recent, diffuse front theory developed by B.D. Marcus and D.K. Edwards [Ref. 7] goes beyond the flat front theory in that it considers the binary diffusion which occurs between the working fluid and non-condensible gas as well as axial conduction in the heat pipe wall. Also considered are convection and radiation heat transfer from the heat pipe. The result of including these additional parameters is a far more realistic description of the gas-vapor interface distribution.

D. OBJECTIVE

The objective of this research is to investigate the operational properties and characteristics of a gas loaded variable conductance heat pipe with a length to diameter ratio of approximately thirty.

A further purpose of this research is to determine the validity of previously published theories on the shape and orientation of the vapor-gas interface when applied to a heat pipe in which there is a large difference in the molecular weights of the working fluid and the non-condensible gas. It is expected that the relative molecular weights of the non-condensible gas and the working fluid will play a significant role in the establishment and maintenance of the interface shape and orientation. Also, the orientation of the heat pipe itself will affect the front. The heat pipe will be operated in the horizontal and vertical positions at various power levels to enable as thorough an investigation as possible of the effects of these parameters.

The results to be presented are the product of the data collected from a heat pipe designed and constructed for this purpose.

II. DESIGN

A. DESIGN CONSIDERATIONS

The general design criteria were to construct a gas loaded variable conductance heat pipe with a length to diameter (aspect) ratio of approximately 30 to 1. Some previous work has been done by Humphries [Ref. 3] on a high aspect ratio (100 to 1) gas loaded variable conductance heat pipe in an effort to ascertain the one dimensional interface characteristics. As a logical sequence to this previous work, it was decided to observe the gas-vapor interface in a larger diameter pipe to determine the effects of geometry on interface shape.

To design a successful heat pipe, one must consider the interrelationship of the physical properties of the heat pipe material, the wick, and the working fluid, as well as the individual performance requirements levied on each. Specifically, the heat pipe material must possess sufficient strength to retain its structural integrity at the temperatures and pressures at which the heat pipe will be operated. The working fluid should have physical properties such as surface tension, density, and viscosity which are suitable for the desired operating conditions. And, the wick should have acceptable capillary transport properties. Furthermore, these components must not have any undesirable effects on each other.

In designing a variable conductance heat pipe, additional parameters must be considered, the most significant of which is the selection of the non-condensible gas. Also, the heat pipe wall must be thin so as to minimize axial conduction and thereby minimize the width of the vapor-gas interface.

For this experiment, a frame also had to be designed to adequately support the heat pipe and associated instrumentation while at the same time permitting it to be rotated freely from vertical to horizontal. Also, and more importantly, it must have as small an effect as possible on the operating characteristics of the heat pipe (i.e. - conduction from the heat pipe by supporting members should be reduced to the maximum extent possible; but, there should be as little interference with free convection from the heat pipe surface as possible).

B. MATERIAL SELECTION

It was decided to utilize stainless steel for the heat pipe, screen wick, spring wire, all tubing and filling valve because of its heat transfer properties, and compatibility with the other components of the system. In addition, it was desired to retain as many characteristics as possible of previous successful designs.

Methanol was chosen as the working fluid because of its compatibility with stainless steel, the fact that its surface tension was such that capillary action could move it through the wick from the condenser to the evaporator, and because its density, and viscosity were more than suitable.

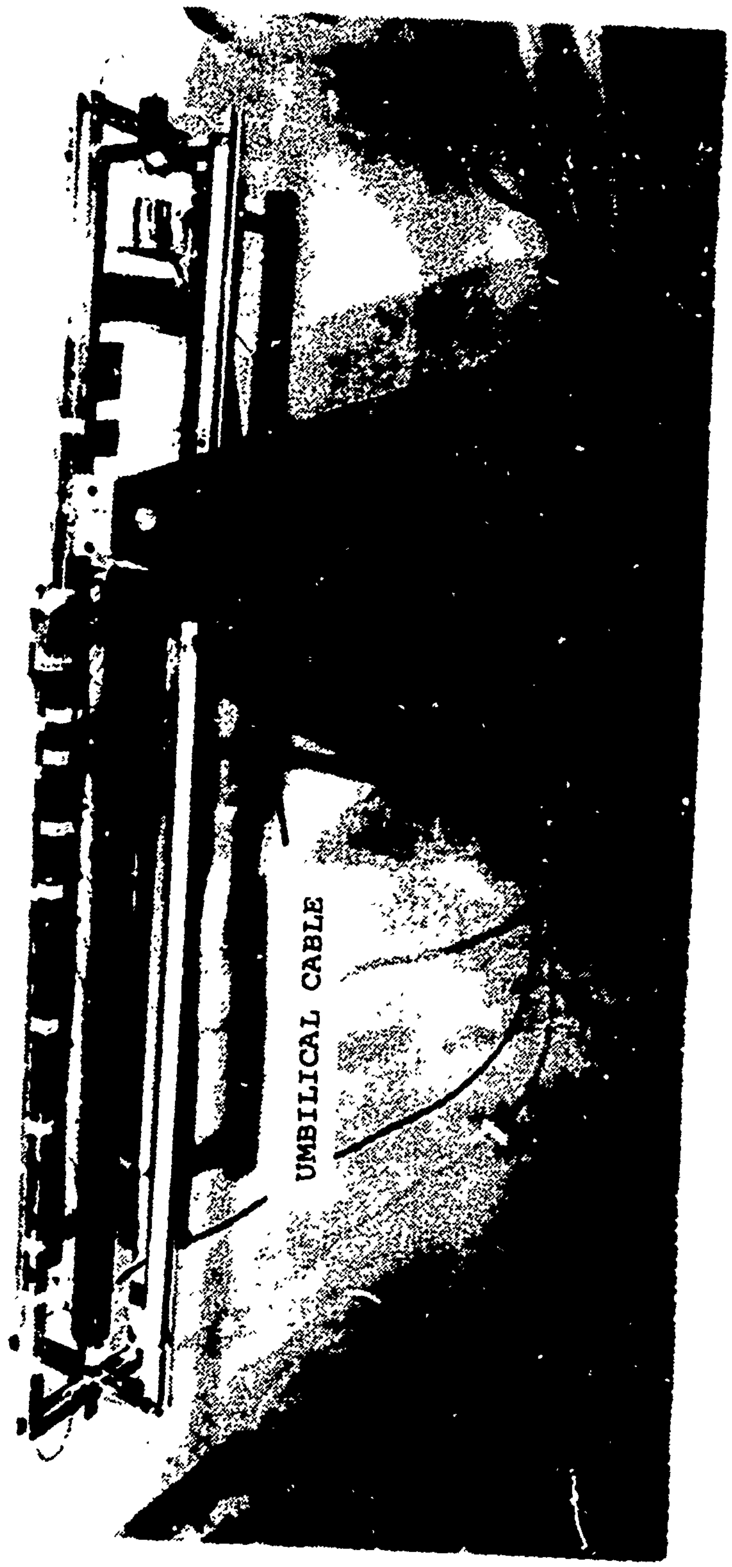
The non-condensible was selected on the basis of its compatibility with other system components and its molecular weight. Krypton, which has a molecular weight approximately 2.6 times that of methanol, was chosen.

C. HEAT PIPE CONSTRUCTION

A section of 0.020 inch wall thickness, two inch outside diameter stainless steel tubing was obtained. A sixty inch length was used to achieve an aspect ration of thirty to one. The length of the evaporator section was twelve inches; that of the adiabatic section was six inches. This left forty-two inches as the condenser length. The thinness of the tube wall promised to reduce axial conduction to the minimum possible while still retaining structural rigidity.

A one hundred twenty mesh stainless steel woven cloth was used for the wick. Through utilization of appropriate design criteria in Marcus [Ref. 7], it was decided to use four wraps of this wick material. This would provide a sufficient capillary head in the wick to overcome the losses due to the pressure drop in the vapor and the liquid plus any losses due to body forces. The equation can be written as follows:

Net Capillary Head		Liquid Pressure Drop		Vapor Pressure Drop		Body Force Head
P_c	$\geq$	ΔP_L	+	ΔP_v	+	ΔP_B



Photograph of Heat Pipe - Horizontal Position

FIGURE 3

Photograph of Heat Pipe - Vertical Position

FIGURE 4

To ensure that the wick remained firmly against the inside pipe wall, a stainless steel coil spring was manufactured and then, while still mounted on its mandril, inserted inside the wire cloth inside the heat pipe and released. The mandril was then removed, leaving the wick firmly in place.

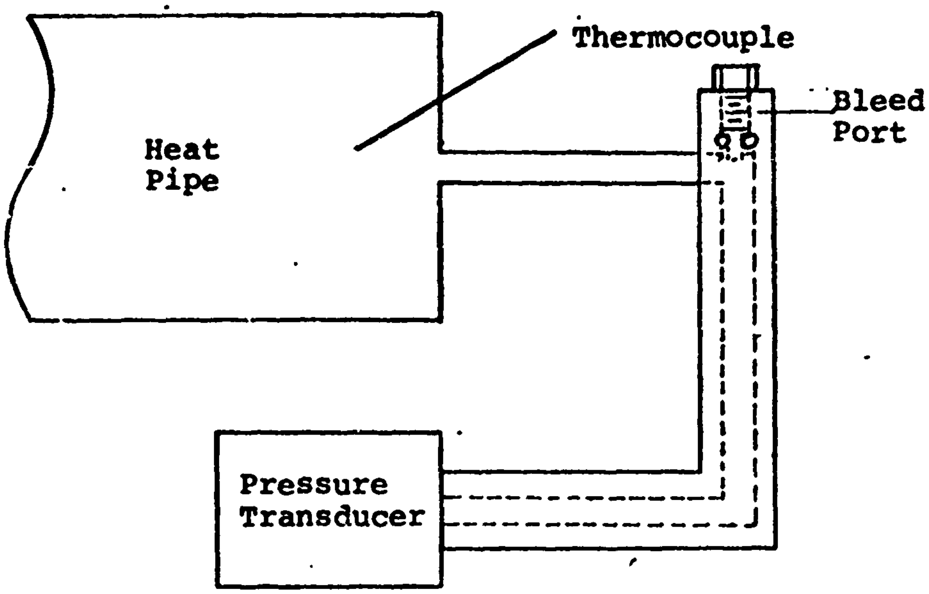
The condenser end of the heat pipe had a welded stainless steel end cap with a 0.25 inch stainless steel tube welded to the center of it to provide an inlet for a pressure transducer. The tube was formed into a U-shape to allow the pressure transducer to be mounted even with the end of the heat pipe to prevent a difference in height between the end of the pipe and transducer which would produce erroneous pressure readings. There was a small hole drilled slightly off center in the end cap to provide access for a stainless steel sheathed thermocouple. In addition, a bleeding device (see Fig. 5) was installed on the 0.25 inch tubing to enable one to bleed off any unwanted non-condensible gas from the heat pipe.

The evaporator end of the heat pipe had a removable end plate which was held in place by four bolts which attached it to a flange which was in turn welded to the heat pipe. This was sealed by means of an O-ring and groove cut into the removable section. An arrangement similar to that of the condenser end was made to enable installation of a pressure transducer and thermocouple. In addition, a stainless steel valve was welded to the 0.25 inch tubing to provide a means for introducing working fluid and non-condensible gas into the interior of the heat pipe as well as evacuating it prior to filling (see Fig. 6). Heating of the evaporator section was accomplished by passing a Direct Current through a 0.125 inch wide length of Nichrome ribbon wrapped around the heat pipe. The Nichrome ribbon was electrically insulated from the heat pipe by painting both it and the heat pipe with insulating paint prior to wrapping the ribbon around the heat pipe.

The supporting structure for the heat pipe was manufactured from one inch aluminum channel beams with the heat pipe being supported by tubes inserted into the mounting blocks for the pressure transducers. These tubes in turn were inserted into other blocks connected to the channel beams in such a way that no part of the framework was in contact with the heat pipe itself. In addition, the frame permits rotation of the heat pipe from the horizontal to the vertical.

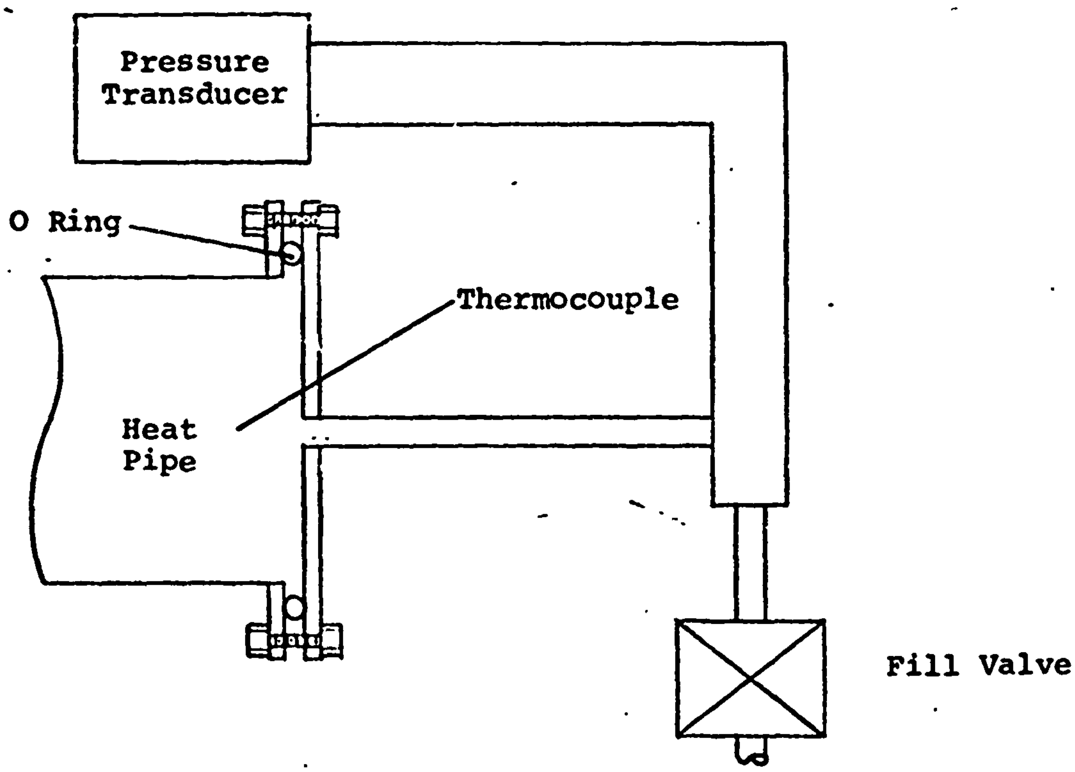
D. INSTRUMENTATION

To accurately determine the temperature profile along the heat pipe as well as the temperature at other selected points, a total of forty thermocouples were installed. The pressure was measured using two pressure transducers, one at the evaporator end and one at the condenser end of the heat pipe. Celesco Model PLC Pressure Transducers were used with the one at the evaporator having a range of zero to



Condenser End of Heat Pipe

FIGURE 5



Evaporator End of Heat Pipe

FIGURE 6

fifty pounds per square inch absolute and the one at the condenser end having a range of zero to one hundred pounds per square inch absolute. It was decided to place the transducers in this way because it was discovered in the course of calibrating them that the zero to one hundred pound per square inch transducer was sensitive to a change in temperature while the other was not. The output from the pressure transducers and thermocouples was recorded digitally on a Hewlett-Packard 2010C Data Acquisition System.

Twenty-four of the thermocouples were placed axially along the top of the heat pipe at two inch intervals beginning one inch from the end of the heated section. This placed the last external thermocouple one inch from the end of the heat pipe. The external thermocouples were attached to the heat pipe by first flattening the bead and then spot welding it to the heat pipe with a Unitek Model 1065 Welder. As previously mentioned, there was a stainless steel sheathed thermocouple placed near the centerline of the heat pipe and parallel to its axis at each end. These protruded approximately one half inch into the interior of the heat pipe itself and were used to determine the interior temperature at the evaporator and condenser. At locations eleven and twenty-nine inches from the condenser end, three additional thermocouples spaced ninety degrees apart were attached to the heat pipe in an effort to determine the temperature variation around the pipe as well as along it. One thermocouple was placed on each pressure transducer to monitor its temperature. Of the six remaining thermocouples, four were placed within and on the insulation for use in determining the percentage of heat applied to the heat pipe which was lost through the insulation. The last two were used to measure the ambient temperature near the heat pipe.

The thermocouples were made from thirty gage Teflon coated copper and constantan wires. The thermocouple end beads were formed with a Dynatech thermocouple welder. The ends of twenty of the thermocouple wires were led to the male half of a twenty-five pin AMPHENOL type connector. The female half was attached to the supporting frame for the heat pipe and connected by means of a twenty wire umbilical cable to the Hewlett-Packard 2010C Data Acquisition System. The other twenty thermocouples were similarly connected to another twenty wire umbilical cable. Each AMPHENOL type connector was covered with sponge rubber to prevent stray air currents from causing temperature gradients in the connector and thereby introducing erroneous data.

Several layers of Johns Manville Min-K insulation were wrapped around the evaporator and adiabatic sections to minimize heat loss. Two thermocouples were placed between the layers of insulation and one on its surface so that the temperature drop across a layer of insulation would be known and the heat flux could be calculated.

In order to accurately determine the amount of heat being applied to the heat pipe, a known resistance was placed in series with the Nichrome heater strip. By measuring the voltage drop across this known resistance, the current being supplied to the heater could be accurately determined. This current was used in conjunction with the voltage drop across the heater to determine the amount of Joulean heating being applied to the heat pipe. These voltages also were measured using the Hewlett-Packard 2010C Data Acquisition System.

After the thermocouples had been attached to the heat pipe and the heat pipe had been mounted in its frame, a coat of flat black paint was applied to it to form a good backdrop for viewing the liquid crystals which were applied.

E. CALIBRATION

The thermocouples were calibrated using a Rosemount Calibration System with a platinum resistance thermometer as a standard. The accuracy of the calibration system is ±0.01 degrees Centigrade. In performing the calibration, the thermocouples were bundled together and placed as close to the platinum resistance thermometer as possible to ensure that any temperature variations in the bath had as small an effect as possible. In addition distilled water and ice made from distilled water were used in the reference junction bath. As a result of extensive calibration data being taken between seventy-five and two hundred sixty degrees

Photograph of the Entire Experimental System

FIGURE 7

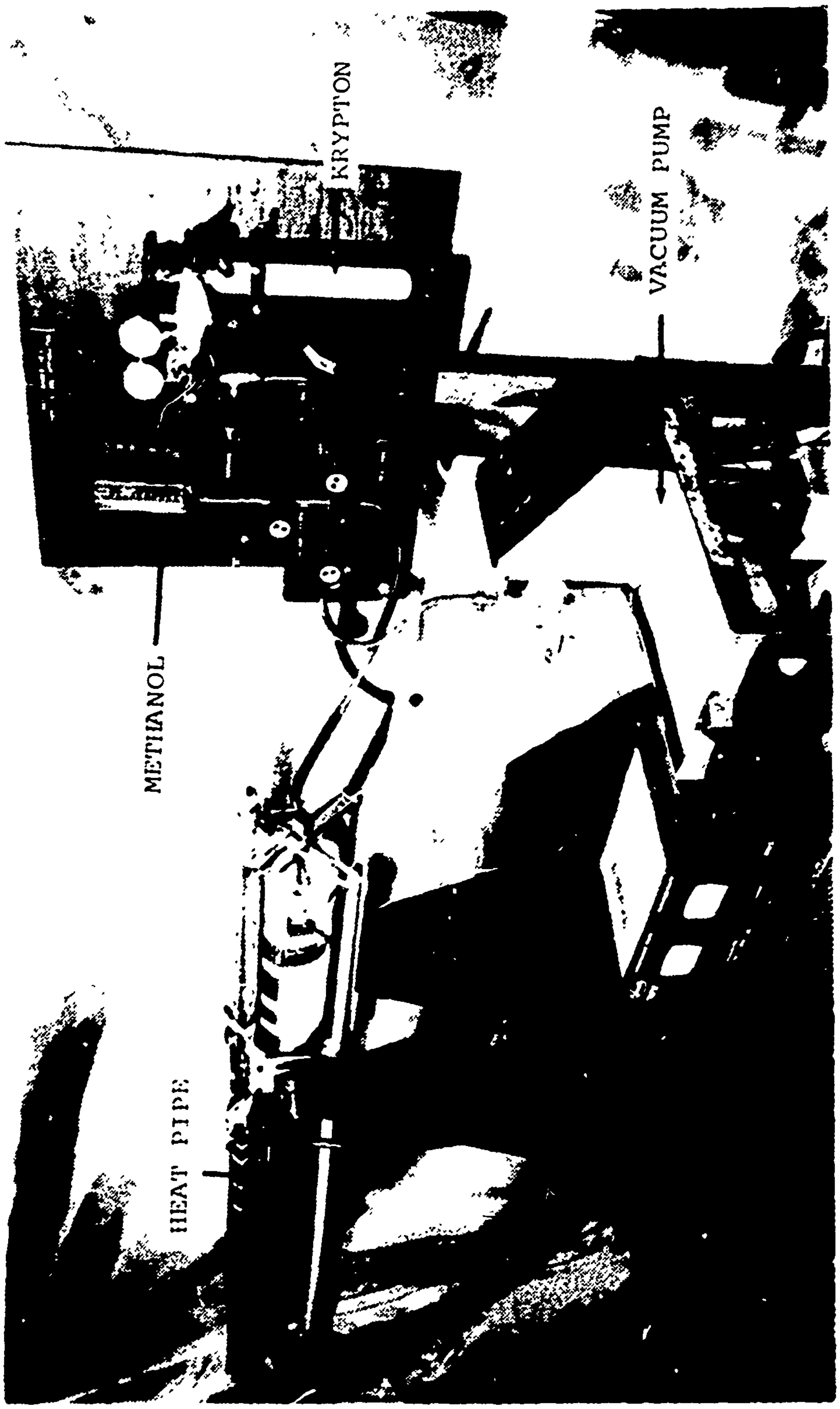
Fahrenheit, it was determined that published calibration tables for copper-constantan thermocouples could be used with an accuracy of plus or minus one half degree Fahrenheit. In addition, several thermocouples were calibrated by themselves and then tack welded to a scrap section of the material from which the heat pipe was made. It was discovered that the attachment operation introduced no discernable error or discrepancy with the earlier calibration. The data for the thermocouple calibration as well as that for the pressure transducer calibration was recorded using the Hewlett-Packard 2010C Data Acquisition System.

A U-tube mercury manometer and a ten foot mercury manometer were used as standards in calibrating the Celesco Model PLC pressure transducers.. Data was taken at pressures ranging from approximately seven to fifty pounds per square inch absolute. The transducers were also calibrated at elevated temperatures on the order of one hundred thirty and two hundred five degrees Fahrenheit. The data was then plotted and a least squares method curve fit was done on a computer. Several orders of equations were tried to obtain the best possible correlation with recorded data. A first order calibration equation for the zero to one hundred pound per square inch absolute transducer yielded an accuracy of plus or minus two tenths of a pound per square inch. A second order equation for the zero to fifty pound per square inch transducer yielded an accuracy of plus or minus one tenth of a pound per square inch.

III. EXPERIMENTAL PROCEDURE

A. EXPERIMENTAL PREPARATION

After construction of the heat pipe itself was complete, and prior to the installation of the thermocouples, the heat pipe was tested for leaks. Utilizing a vacuum pump, the pressure in the pipe was reduced to 2 X 10^{-6} microns of mercury where it was maintained for a period of twenty-four hours. The thermocouples, heater, and insulation were then mounted on the heat pipe. It was then necessary to decide at which power levels the heat pipe would be operated, and concurrently, how much methanol was required to successfully operate the heat pipe. The void volume of the wick was calculated as was the volume of the stainless steel tubing connecting the pressure transducers to get a very rough idea of the amount of methanol which would be required. After several iterations of adding methanol and operating the heat pipe, it was decided that one hundred seventy milliliters yielded a sufficiently safe operating level to prevent complete evaporation of the methanol and burnout of the heat pipe while at the same time not overloading it to the extent that a puddle of methanol would be left on the bottom of the heat pipe while it was operating. Operation with this amount of methanol at powers between zero and one hundred fifty watts yielded a maximum temperature of approximately one hundred eighty-five degrees Fahrenehit. The methanol was



Photograph of Vacuum Pump and Fill Assembly

FIGURE 8

added by simply connecting the fill valve of the heat pipe to a graduated glass bottle filled with methanol with a short length of heavy rubber tubing. The bottle was then inverted and the methanol was allowed to displace the air in the valve and tubing. The level of the methanol in the graduated glass bottle was then noted and the fill valve was opened. When the desired amount of methanol had entered the heat pipe, the valve was secured. Upon completion of the filling operation, the pressure in the heat pipe was approximately two and one half pounds per square inch absolute. Also, upon completion of the final filling operation, power was applied to the heat pipe to raise the internal pressure above atmospheric. The bleed mechanism was then opened and the non-condensible gas which had accumulated in the condenser end of the heat pipe was bled off. It is estimated that three or four milliliters of methanol escaped during this evolution.

Also, with no power being applied to the heater, and with the pipe level and at room temperature, the output of the thermocouples was monitored to ensure that no undesirable effects on the uniformity of electrical output had occurred during the attachment of the thermocouples to the heat pipe wall.

B. OPERATION OF THE HEAT PIPE IN THE CONVENTIONAL MODE

Upon completion of the bleeding operation, the heat pipe was operated in the conventional mode in both the horizontal and vertical (condenser uppermost) positions. Power was varied in increments of twenty-five watts from twenty-five watts to one hundred fifty watts and back down to twenty-five watts in both positions. The power levels mentioned are those calculated from the measurement of the voltage drop across the heater and the known resistance. The power actually absorbed by the heat pipe is this calculated power minus whatever losses occurred through the insulation (see Table I). All of the graphs presented herein show the power actually absorbed by the heat pipe and not that applied by the heater.

During this phase, the amount of time required for the heat pipe to come to steady state was monitored. After five hours, the temperature recorded by the thermocouple in the insulation nearest the heat pipe was observed to change by less than one fourth of one degree Fahrenheit per hour. The above was found when the power was being increased. When the power was being decreased, it was discovered that approximately four times as much time was required for the temperature to stabilize. Thus, data was taken at intervals of no less than six hours while the power was being increased and no less than twenty hours when the power was being decreased.

C. OPERATION OF THE HEAT PIPE IN THE VARIABLE CONDUCTANCE MODE

In the second phase of the experiment, the heat pipe was caused to act in a variable conductance mode by the introduction into the heat pipe of the non-condensible gas krypton.

TABLE I

CONDENSER AND EVAPORATOR TEMPERATURES

HORIZONTAL POSITION

Nominal Power (Watts)	Actual Power (Watts)	Evaporator Temperature (°F)	Condenser Temperature (°F)
25	23.9	98.3	97.2
50	47.9	115.7	114.6
75	71.9	134	133
100	96	147.4	146.6
125	119.9	167.2	166.5
150	144.2	183.4	182.8

VERTICAL POSITION

Nominal Power (Watts)	Actual Power (Watts)	Evaporator Temperature (°F)	Condenser Temperature (°F)
25	23.4	95.5	94.5
50	47	120.7	119.7
75	70.8	138.8	138.3
100	94.9	148.1	147.7
125	118.7	160.2	159.7
150	142.6	176.7	176.4

The gas was added while the heat pipe was operating in the conventional mode at a medium power level. It was thought that if the gas was added in this manner, then the interface between the working fluid and non-condensible gas could be accurately positioned at the desired location along the heat pipe. As a precautionary measure, the approximate quantity of gas needed to half fill the condenser portion of the heat pipe was calculated prior to the actual introduction of the gas. During the actual loading of the heat pipe with non-condensible gas, it was discovered that the interface did not develop as quickly as had been anticipated. This necessitated judging the amount of gas to be added solely on the basis of the change in pressure in the heat pipe.

The procedure used to load the non-condensible gas was as follows. The fill rig was first evacuated to a pressure of 2×10^{-6} microns of mercury, then purged with a small amount of krypton, and then re-evacuated to the above pressure. The system was then pressurized and the fill valve on the evaporator end of the heat pipe opened slowly. When the pressure in the heat pipe had increased a predetermined amount, the valve was secured and the procedure was complete.

The exact amount of gas added was determined by securing the power to the heat pipe and allowing it to come to steady state at ambient temperature. The pressure was then measured and recorded. The internal temperature was used to enter the methanol saturated vapor tables [Ref. 8]. This was the partial presure of methanol in the heat pipe at that particular temperature. The partial pressure of methanol was then subtracted from the total pressure to obtain the partial pressure of krypton in the heat pipe. Using this pressure, the volume of the heat pipe, and its internal temperature, the amount of non-condensible gas present could be calculated.

After the charging with 5.65×10^{-3} LBM of krypton had been accomplished, the heat pipe was operated in both the horizontal and vertical positions at the same power levels at which it had been operated in the conventional mode. The criteria used to determine whether or not steady state had been reached were the same as those used in the conventional mode of operation.

D. LIQUID CRYSTAL EXPERIMENTATION

The third phase of the experiment was a qualitative study of the vapor-gas interface which utilized temperature sensitive liquid crystals to show the orientation of the interface relative to the heat pipe axis. The operational characteristic of the variable conductance heat pipe which makes such observation possible is the sharp change in temperature across the vapor gas interface. By choosing a liquid crystal which would activate at some temperature between that of the condensing methanol and that of the body of non-condensible gas, the location and orientation of the interface could be determined.

The liquid crystals used were manufactured by the National Cash Register Company and belong to the general class of encapsulated cholesteric liquid crystals (see Ref. 4).

One side of the heat pipe was coated with R-37 crystals. The 'R' indicates that the change in optical properties occurs over a change in temperature arbitrarily deemed to be regular in size about 2.2 °C, while the 37 indicates the temperature in °C at which the color change begins. Specifically, the temperature color changes take place at: red, 37.2 °C, green, 37.6 °C, and blue 39.1 °C.

The following application procedure was followed. First, the entire heat pipe was thoroughly cleaned. Then, after attaching the thermocouple beads to the heat pipe, it was sprayed with a flat black paint because the crystals are transparent and require a dark, non-reflective background to enhance the visibility of their color changes. The liquid crystals were then diluted with distilled water to facilitate their application and several thin coats were applied to the heat pipe. Some difficulty was encountered in completely dissolving the crystals and in wetting the heat pipe surface with the result that the coating on the heat pipe appears somewhat blotchy in some areas. This is in part due to the fact that the crystals had inadvertently been frozen prior to their use. Though this did not change their light reflective properties, it did affect their physical consistency to the extent that they were difficult to apply.

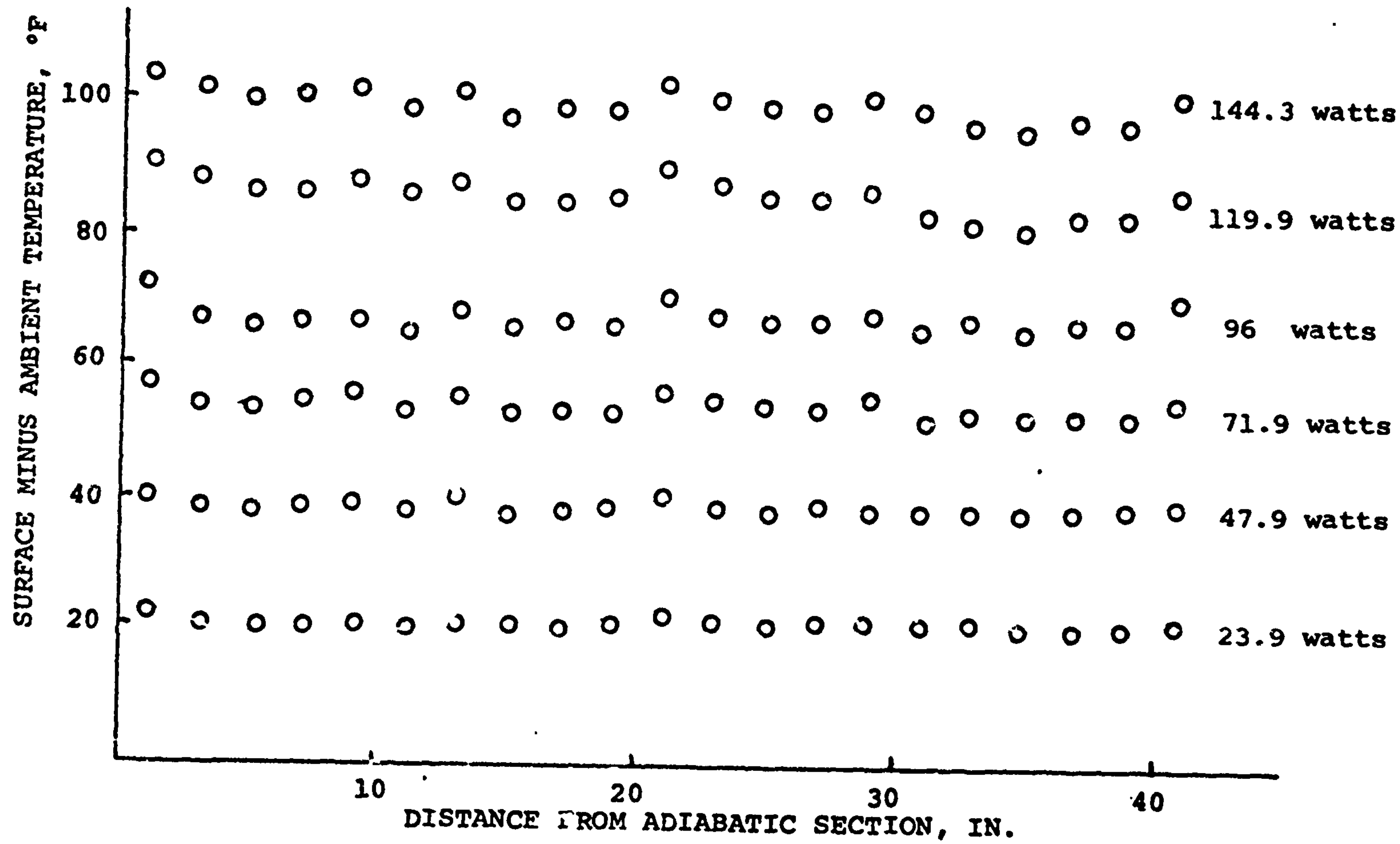
To increase the temperature range over which the orientation of the vapor-gas interface could be observed, two other liquid crystals with higher activation temperatures were mixed and applied to the other side of the heat pipe. These crystals were designated S-43 and S-62, the 'S' denoting that the change in optical properties occurs over a smaller temperature range (about 1.1 °C) than the 'R' type crystal. The mixing of the two crystals does not affect their activation temperatures, but it does decrease the brightness of the color shift and therefore one's ability to photograph it.

The experimental procedure was essentially the same as in the second phase of the experiment except that color pictures of the vapor-gas interface were taken.

IV. EXPERIMENTAL RESULTS

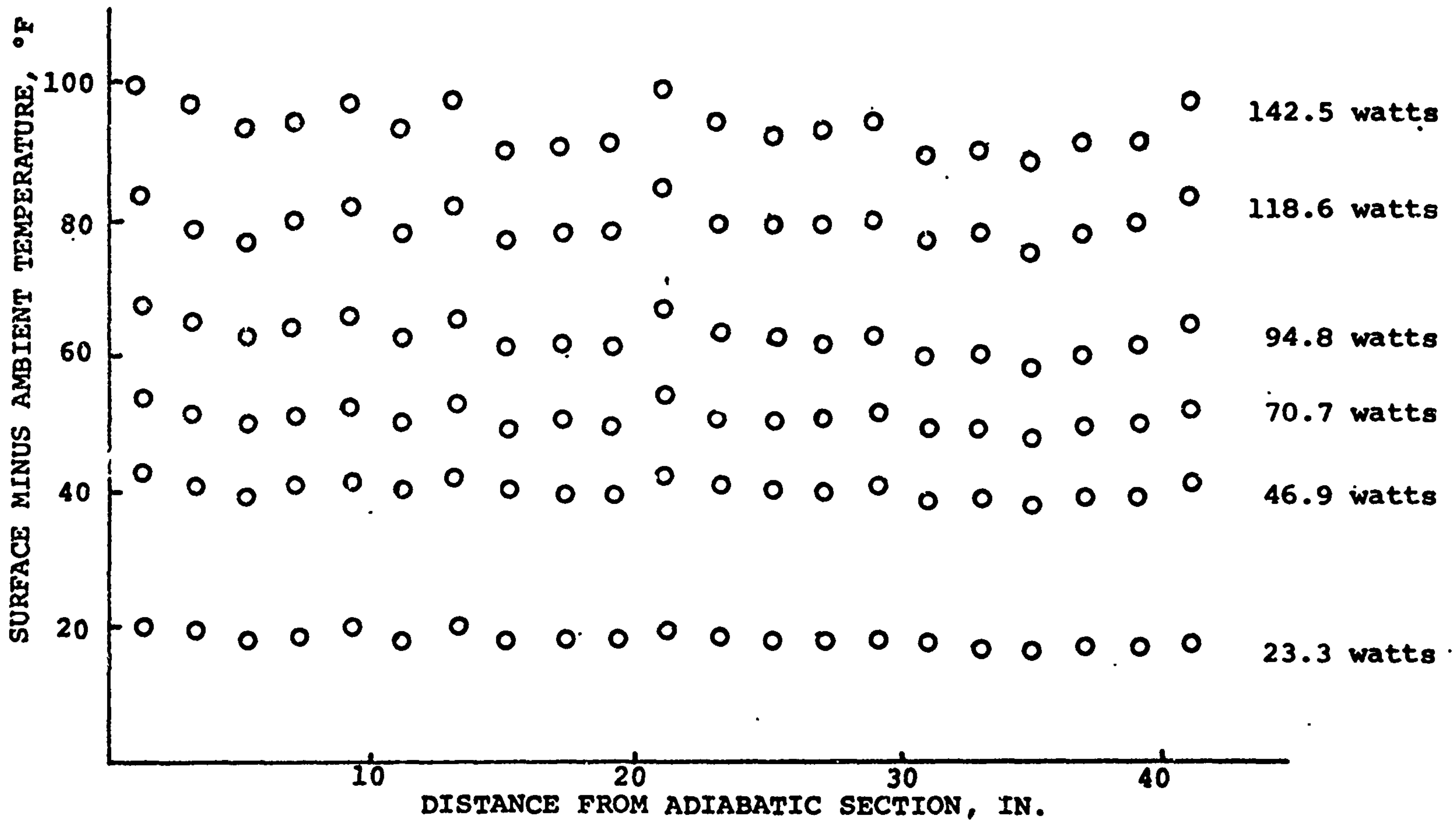
A. CONVENTIONAL HEAT PIPE RESULTS

The first phase of the experiment was to operate the heat pipe in the conventional mode. As previously stated, the heat pipe was operated in the horizontal and vertical positions at input powers of 25, 50, 75, 100, 125, and 150 watts (see Figs. 9 and 10 for the data obtained). From the data, it can be seen that the device is operating as a heat pipe (i.e. in an isothermal or nearly isothermal manner). The maximum internal change in temperature between the evaporator and condenser occurred at twenty-five watts input power with the heat pipe in the horizontal position and was approximately 1.4 °F. The smallest drop in temperature occurred at one hundred fifty watts input power with the heat pipe in the vertical position and was approximately 0.4 °F (see Fig. 10). However, the external thermocouples indicate that the change in temperature along the surface of the heat pipe is slightly greater than the change in the interior of the heat pipe. Also, the variations in temperature along the surface of the heat pipe indicated by the thermocouples on the surface is somewhat greater than that indicated by the internal thermocouples. Data obtained from these thermocouples will be used to indicate general trends vice specific variations in temperature.



Surface Temperature Minus Ambient Temperature vs. Distance from the Adiabatic Section; Horizontal Position, Conventional Mode

FIGURE 9



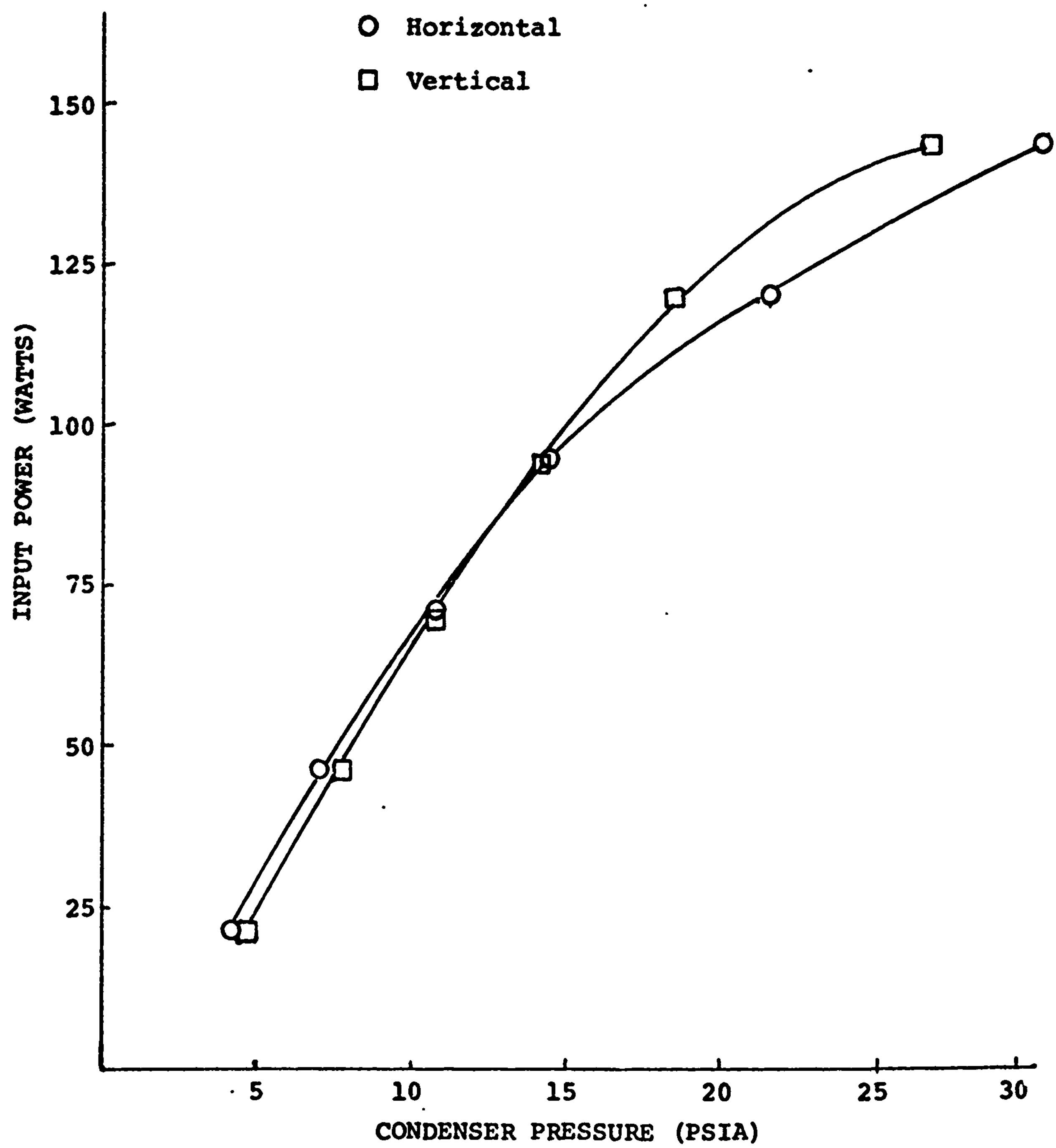
Surface Temperature Minus Ambient Temperature vs. Distance from the Adiabatic Section; Vertical Position, Conventional Mode

FIGURE 10

The data expressed in Figures 11 and 12 support the assumption that the methanol in the heat pipe is operating as a saturated vapor. The observed condenser pressure was compared to the pressure obtained from the saturation table for methanol [Ref. 8] which was entered with the temperature measured inside the condenser. In no case did the observed and tabulated pressures differ by more than four per cent with the values obtained for the vertical position being closer to the tabulated value than those obtained for the horizontal position.

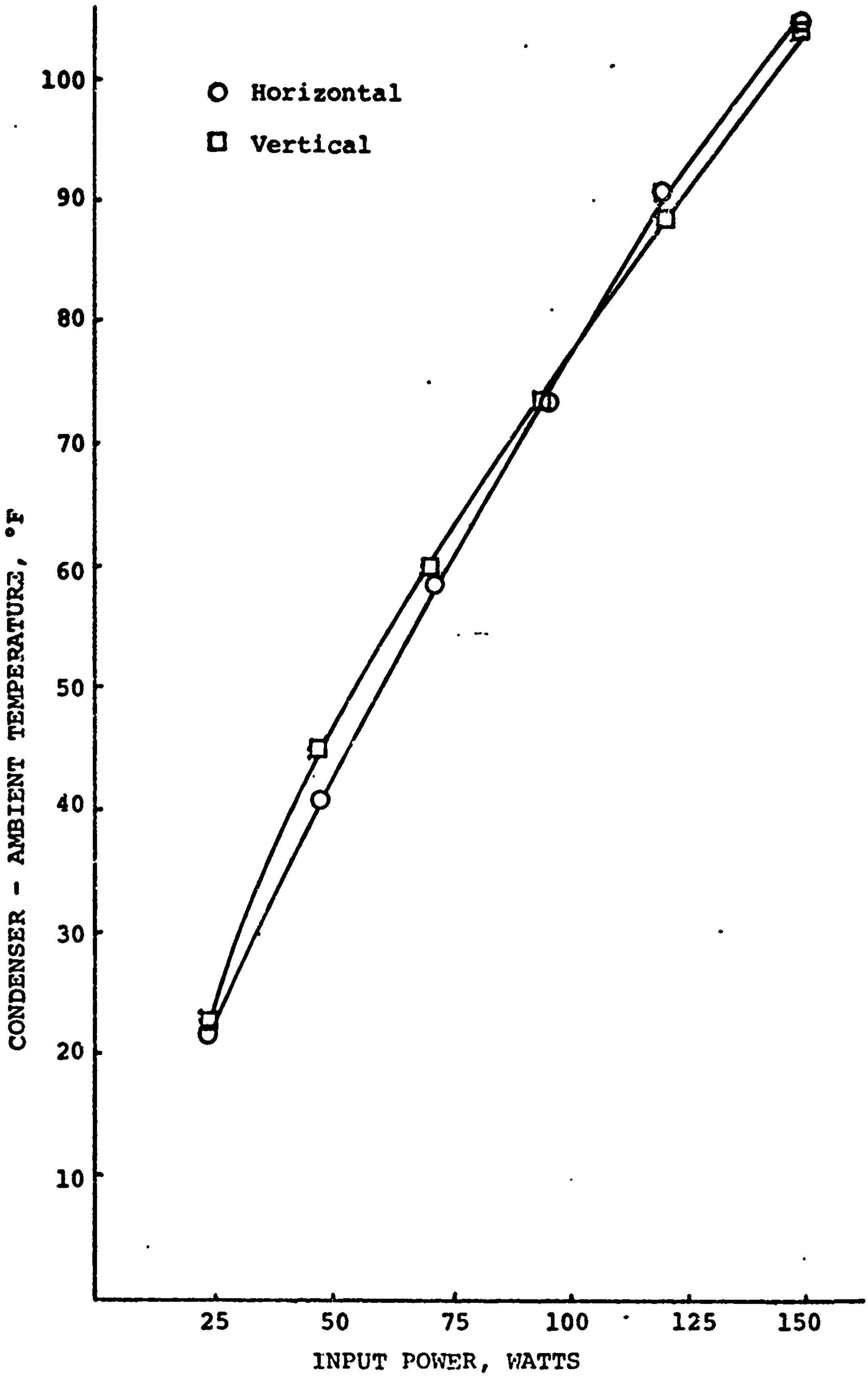
To eliminate the effects of a change in ambient temperature on the performance parameters, the ambient temperature was subtracted from the observed temperature. In this way, a valid comparison could be made between various data runs even though the ambient temperature may have changed by as much as eight or ten degrees Fahrenheit.

It was noted that the evaporator temperature differences varied insignificantly between the horizontal and vertical positions. Thus, it seems that gravity has a very small effect on the heat pipe when it is operated in the conventional mode. It is apparent that the higher heat transfer coefficient for the pipe in the horizontal position is compensated for by the increased flow rate with the pipe in the vertical position. This is true assuming that the pressure difference caused by a five foot column of methanol vapor is negligible.



Input Power vs. Condenser Pressure

FIGURE 11



Input Power vs. Condenser Temperature

FIGURE 12

It was also noted that the temperature on the surface of the heat pipe at a position near the end of the condenser section was higher than the temperature several inches back towards the evaporator. This increase varied, in the horizontal position, from approximately one half of a degree Fahrenheit at twenty-five watts to some four degrees at one hundred fifty watts. In the vertical position, it was approximately one degree at twenty-five watts and six degrees at one hundred fifty watts. Though it is thought that this phenomenon has some relationship to the presence of the end plate, it is not known how the end plate causes it to occur.

The drop in pressure from the evaporator to the condenser was much greater in the vertical than the horizontal position as could be expected. It was noted, however, that the change in pressure at 25 watts in the vertical position measured when the heat pipe was first placed in the vertical position was slightly greater than the value recorded at the same power but after the power had first been increased in steps to one hundred fifty watts and decreased in steps back to twenty-five watts. This higher than expected pressure drop was probably caused by a pool of methanol in the evaporator end of the heat pipe. That this was the case is borne out by the fact that there is almost a ten degree Fahrenheit change in temperature between the evaporator and condenser with the evaporator being lower. If there were a pool of methanol at least one half inch deep in the evaporator, then the internal thermocouple there would be covered with the cooler liquid methanol. Apparently, twenty-five watts is not enough power in this position to completely evaporate all of the liquid methanol. But, after it has been evaporated at a higher power, the twenty-five watts is enough to maintain it in the wick and keep it from puddling in the bottom.

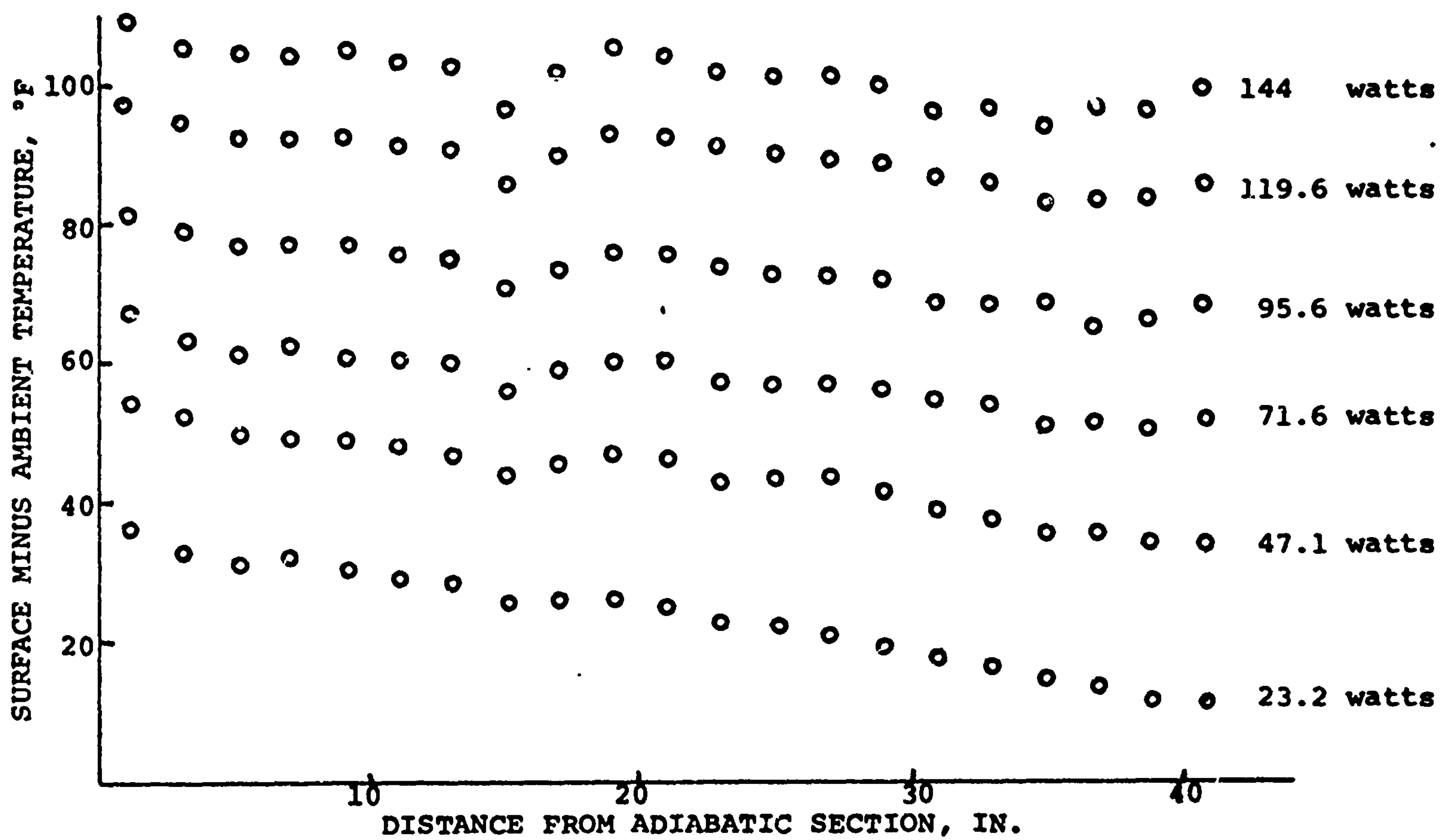
B. VARIABLE CONDUCTANCE MODE RESULTS

The original one hundred seventy milliliters of methanol were left in the heat pipe and a charge of 5.65×10^{-3} LBM of krypton was added. The heat pipe was then operated in the same two positions and at the same power levels as in the conventional mode.

Figures 13 and 14 show the temperature profiles for the horizontal and vertical heat pipe positions. The data plotted was obtained from the thermocouples on the surface of the heat pipe and the comments made earlier concerning the local temperature variations apply equally here.

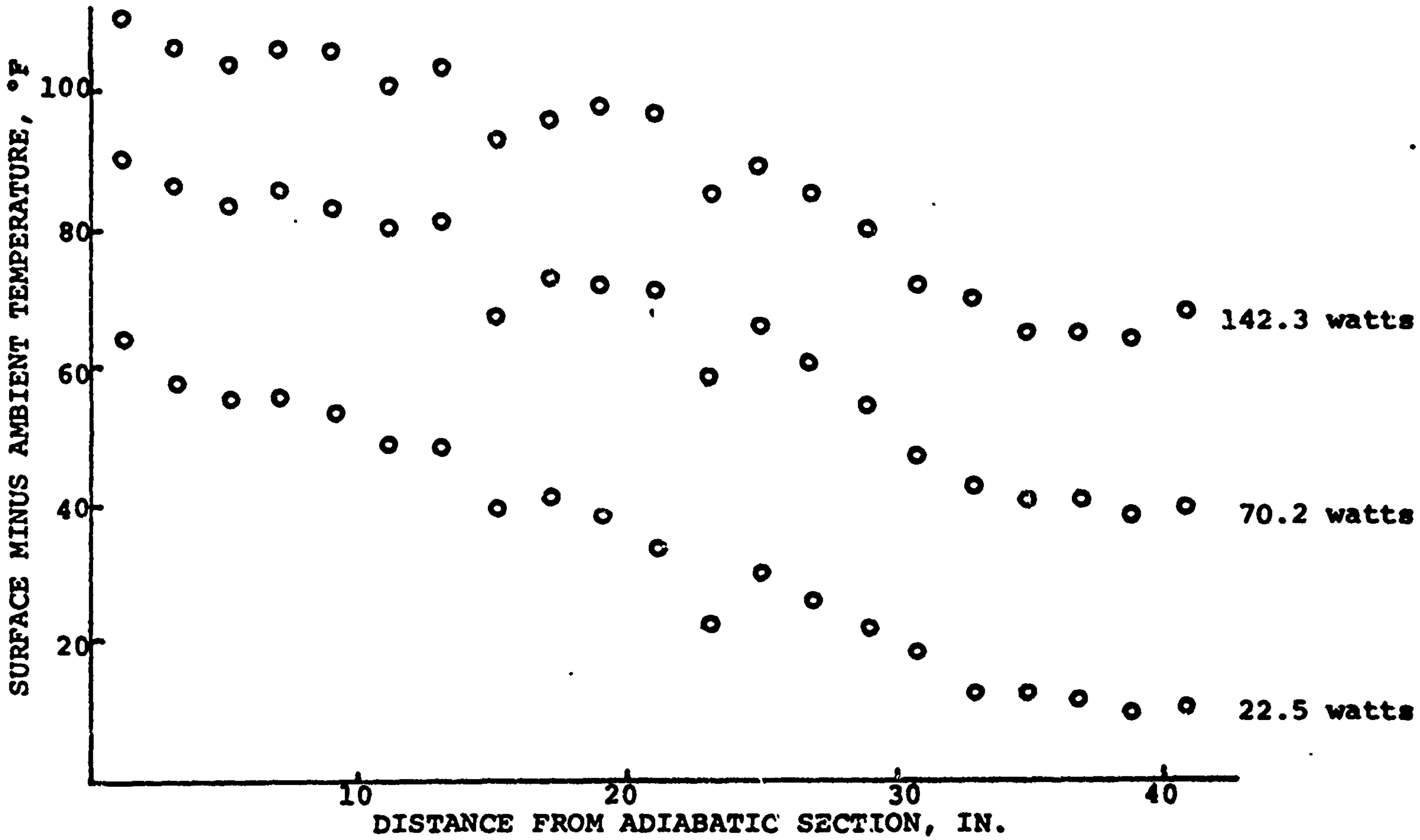
As noted previously, the primary objective of this research was to observe the nature of the vapor-gas interface in a variable conductance heat pipe when there is a large difference in the molecular weights of the working fluid and the non-condensible gas.

With the heat pipe in the horizontal position, the vapor-gas interface was nearly horizontal at twenty-five watts



Surface Temperature Minus Ambient Temperature vs Distance from Adiabatic Section - Horizontal - Gas Loaded

FIGURE 13



Surface Temperature Minus Ambient Temperature vs Distance from Adiabatic Section - Vertical - Gas Loaded

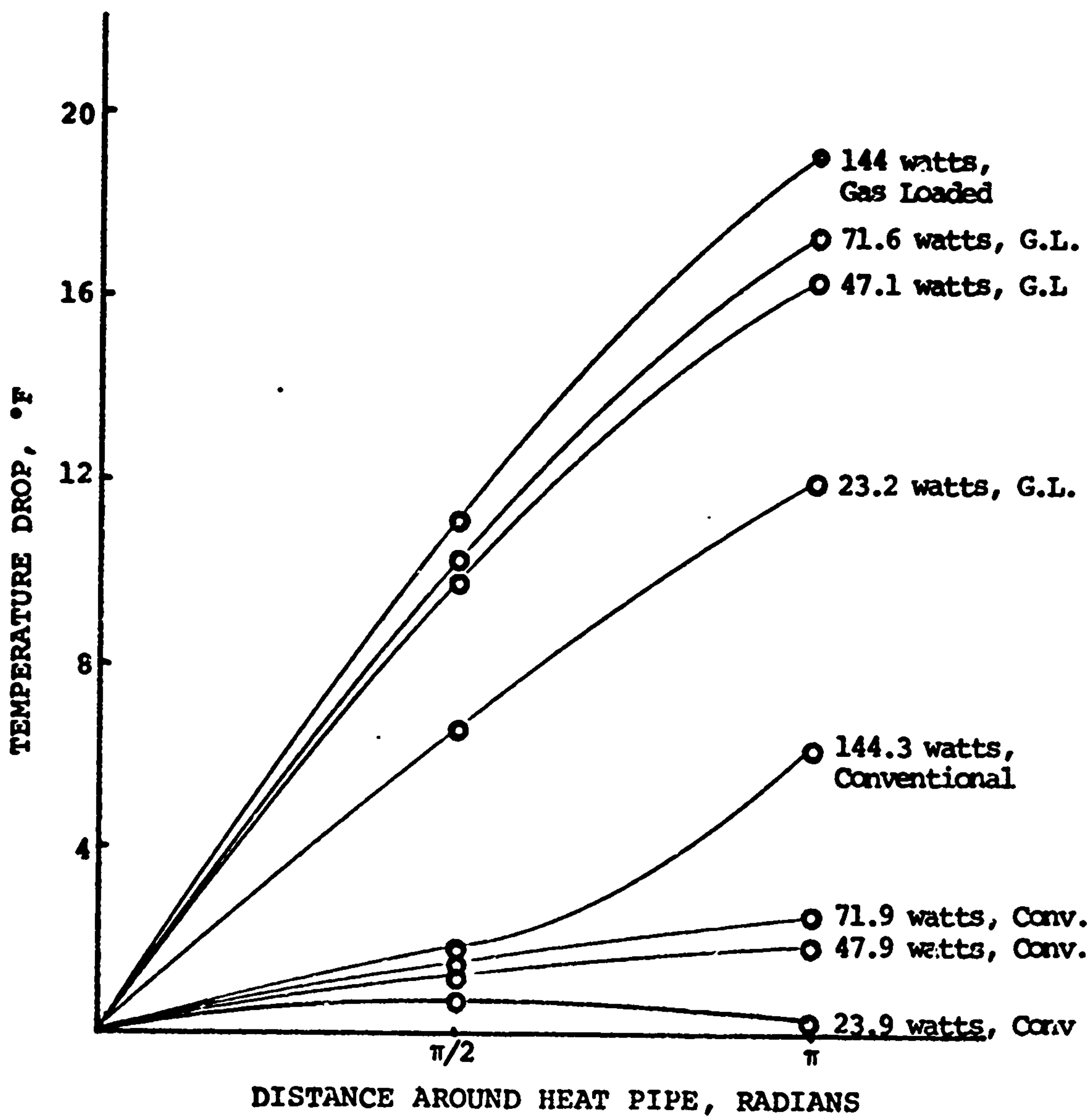
FIGURE 14

and then became effectively horizontal at the higher power levels. Utilization of the data collected makes possible reasonable assumptions concerning the position and orientation of the gas-vapor interface.

As shown in Fig. 13, there is really no well-defined front which can be shown with the thermocouples located along the top of the heat pipe. However, in Fig. 15 the change in temperature from the top to the bottom of the heat pipe in the variable conductance mode can be seen to be much greater than the difference in the conventional mode. This implies the existence of a vapor-gas interface between the top and bottom of the heat pipe. The fact that this large temperature difference exists near the beginning of the condenser and near the end of it implies that the interface is horizontal or at least nearly so.

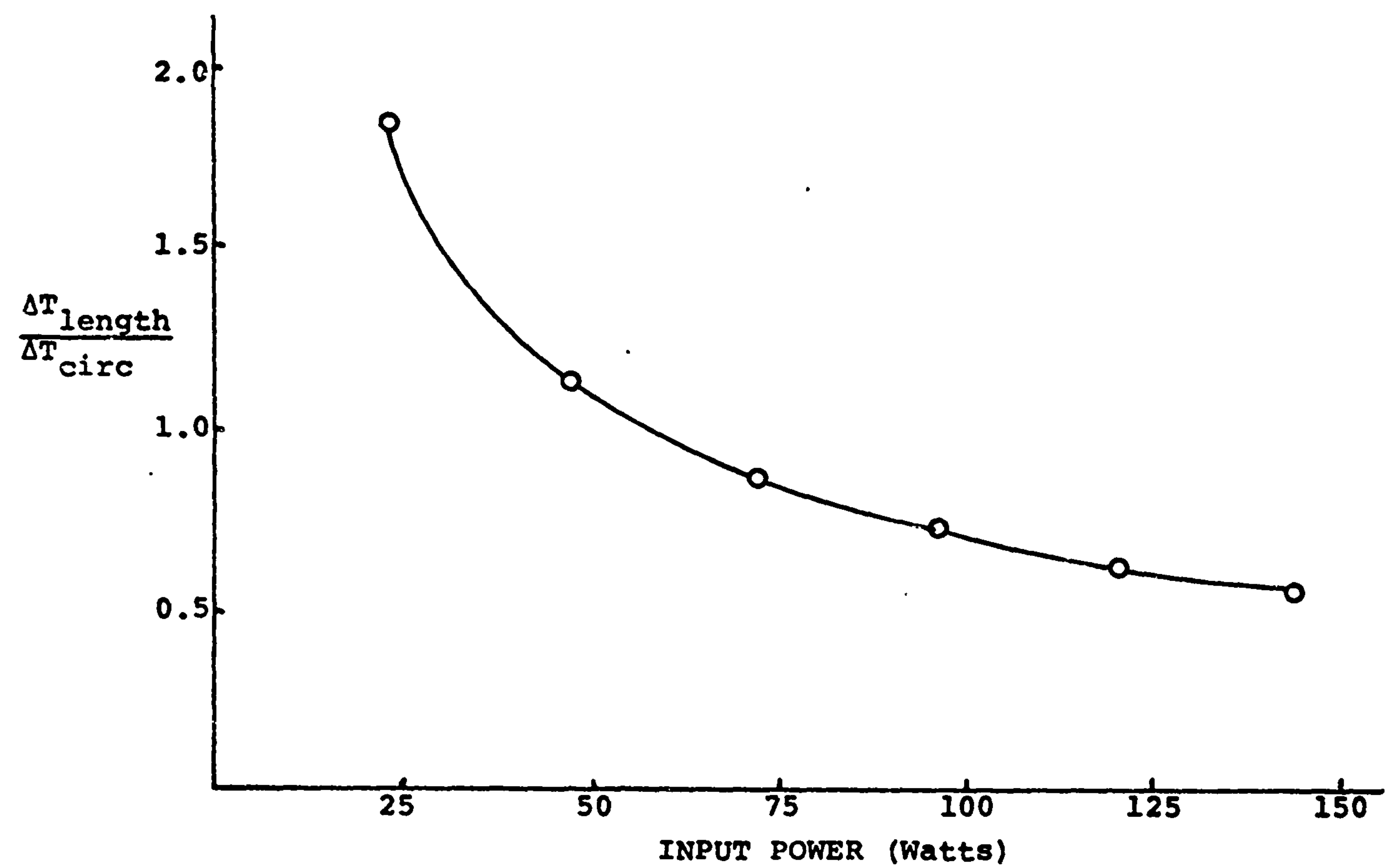
Fig. 16 shows the relation between the temperature drop along the length of the heat pipe divided by the temperature drop from the top to the bottom of the heat pipe versus the input power. At the lowest power, the ordinate has its maximum value. As the power is increased, the value of the ordinate decreases as a result not only of a lessening in the temperature drop along the heat pipe, but also because of an increase in the magnitude of the change in temperature from the top to the bottom.

Thus, it is apparent that at least in the horizontal case, the heavier non-condensible gas forms a pool along



Temperature Drop vs. Distance Around Pipe

FIGURE 15



Change in Temperature Along Length Divided by Change in Temperature from the Top to the Bottom of the Heat Pipe vs. Input Power

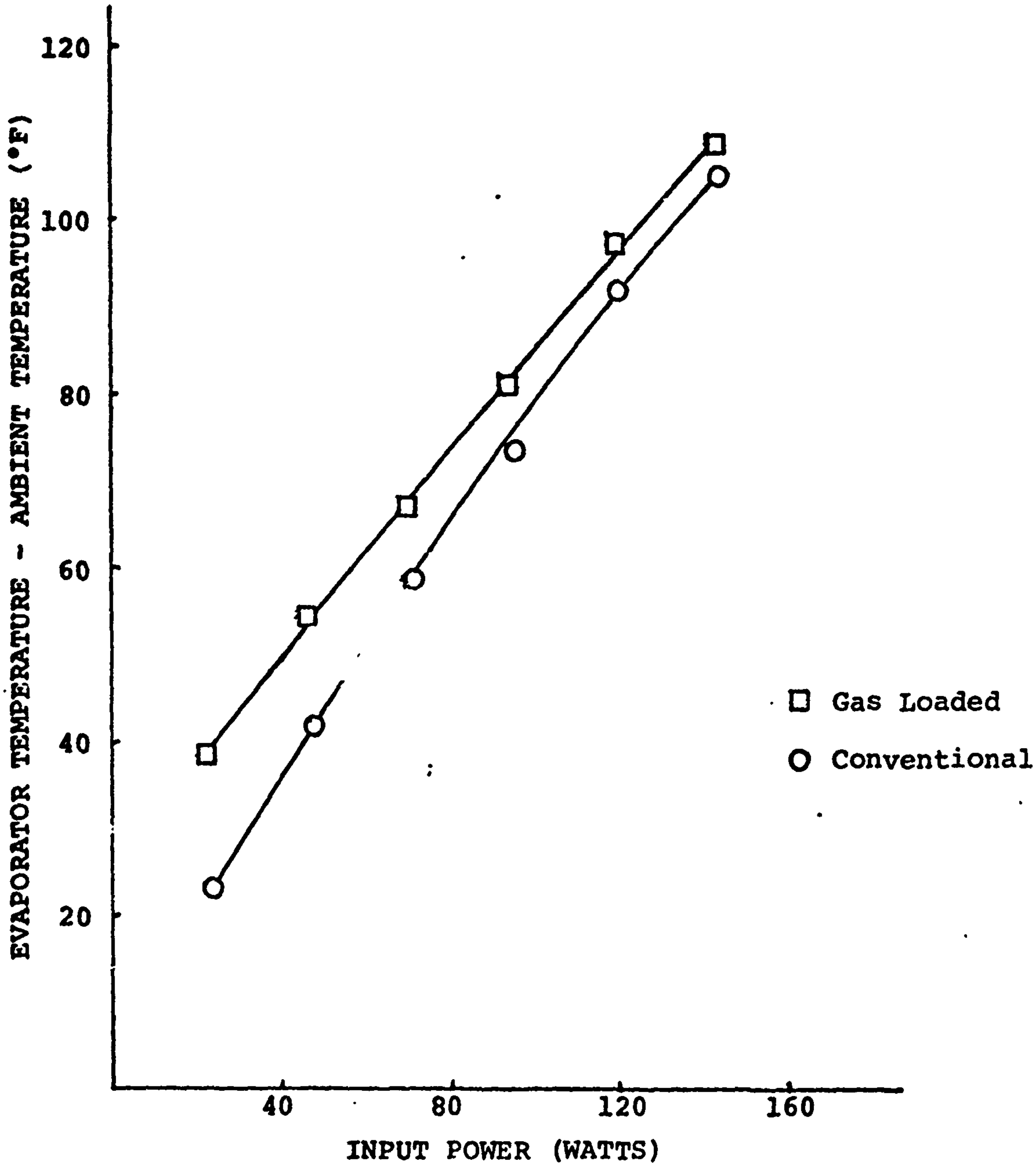
FIGURE 16

the lower half of the heat pipe. This of course is not the behavior predicted by either the flat front theory which stipulates a vertical interface or the diffuse front theory.

When the heat pipe is operated in the vertical position however, an interface does form which is more sharply inclined relative to the axis of the heat pipe than that of the horizontal position (see Fig. 14). Furthermore, the interface itself appears to become more clearly defined with increasing heat input. This can be explained by the fact that the force which maintains the non-condensible gas in the condenser is due largely to the momentum of the vapor particles as they move to the condenser. Gravity tends to oppose this as the non-condensible is heavier than the vapor. At the lowest power, the front is actually quite diffuse. But, as the power is increased, the momentum of the vapor particles is also increased and there is less of a tendency of the non-condensible to move towards the bottom of the heat pipe.

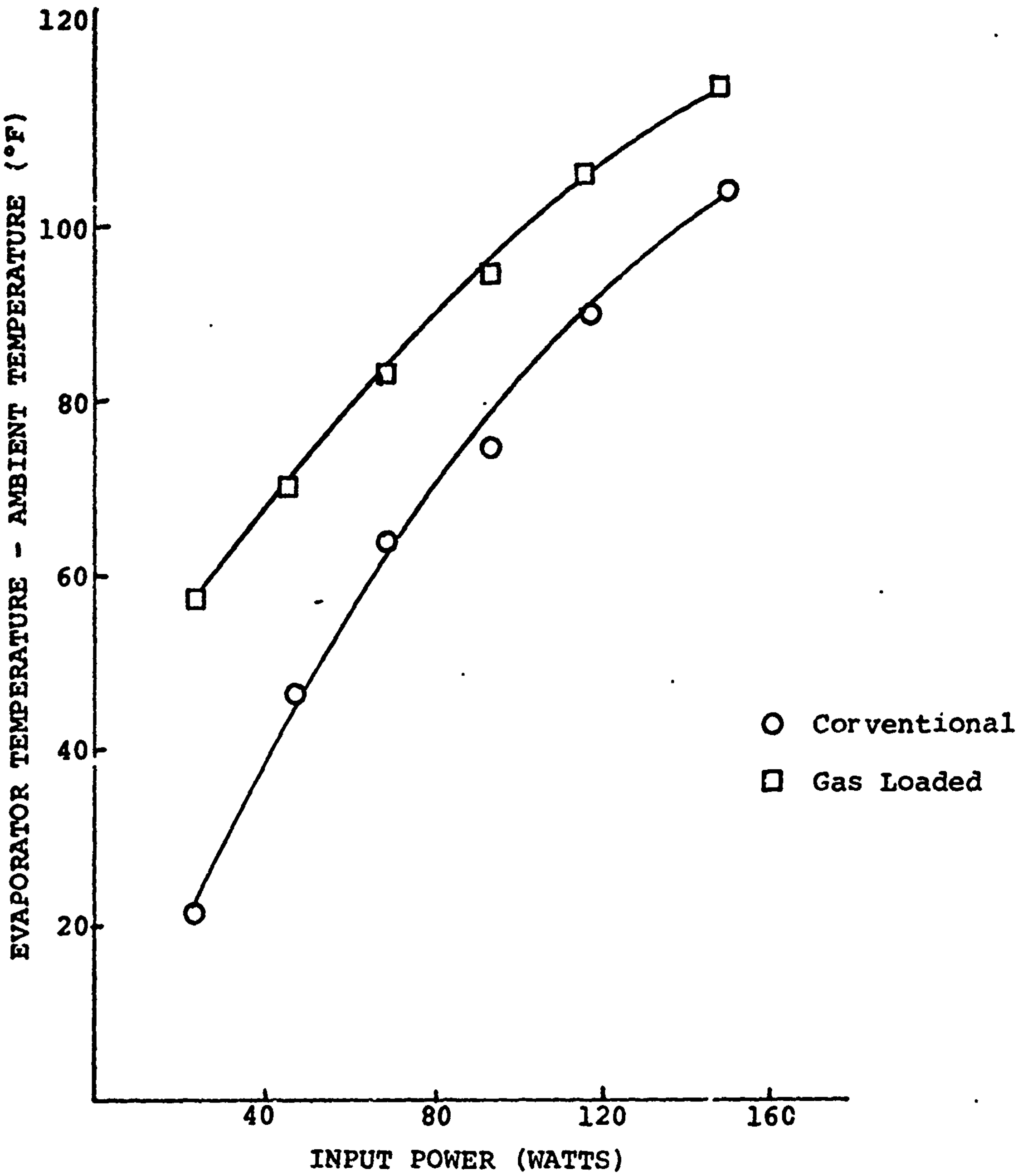
C. LIQUID CRYSTAL RESULTS

The third phase of the experiment was to operate the heat pipe in the variable conductance mode, using temperature sensitive liquid crystals to obtain a qualitative representation of the vapor-gas interface. As noted, one side of the condenser had been coated with two coats of R-37 while



Surface Temperature Minus Ambient Temperature vs. Input Power for Conventional and Gas Loaded Modes - Horizontal Position

FIGURE 17



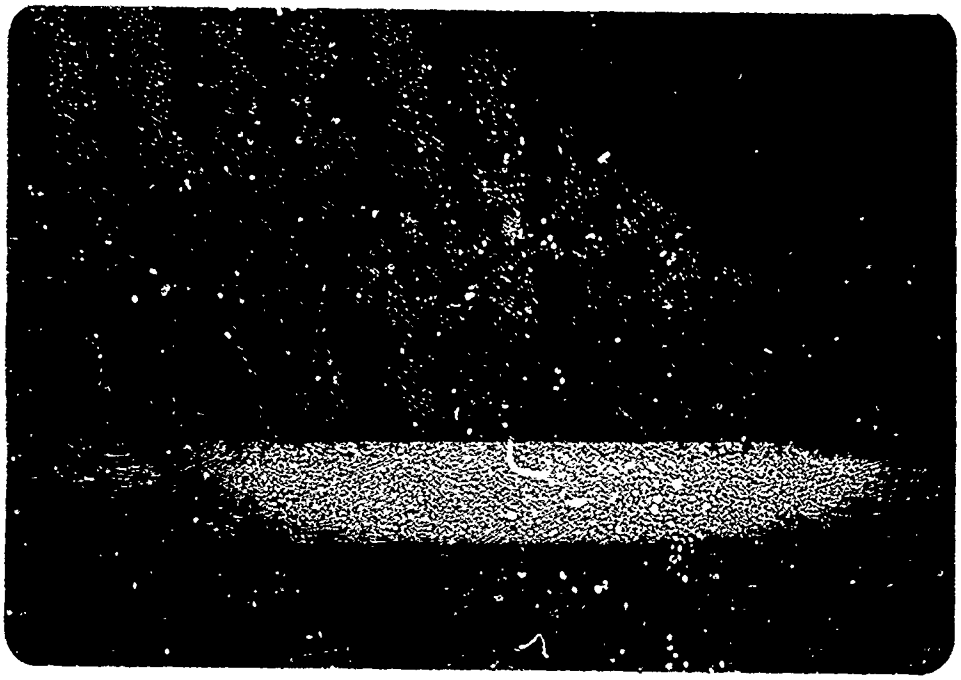
Surface Temperature Minus Ambient Temperature vs. Input Power for Conventional and Gas Loaded Modes - Vertical Position

FIGURE 18

the other had been coated with two coats of a mixture of S-43 and S-62. The heat pipe was operated in the horizontal and vertical positions at the same power input levels as were previously used.

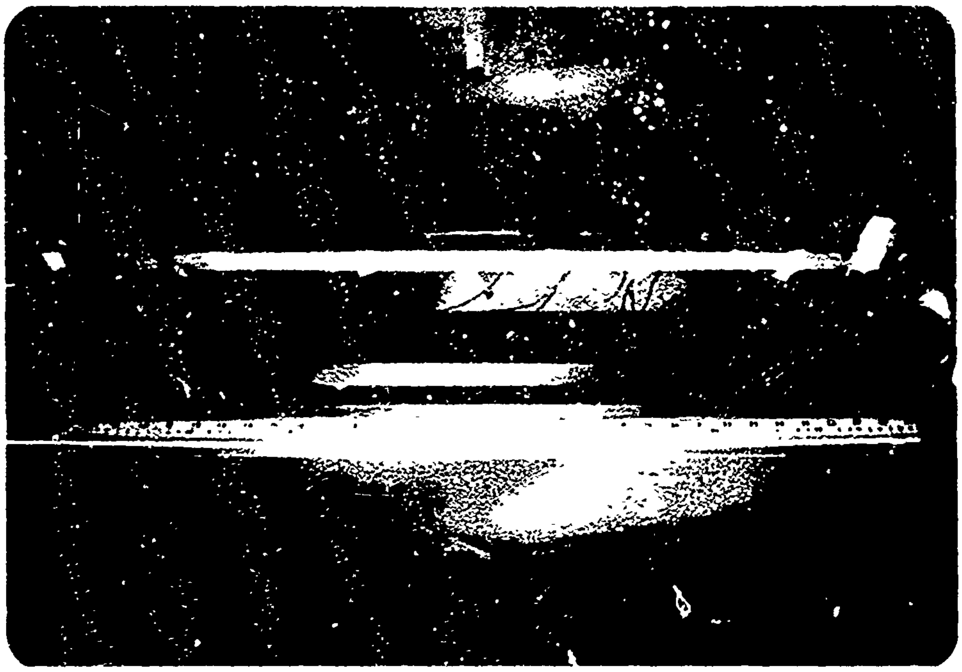
The liquid crystals made vividly clear the orientation of the vapor-gas interface in both the horizontal and vertical positions. With the heat pipe in the horizontal position, it could be seen that the bottom of the interface was closer to the evaporator than the top of the interface by about half of the condenser length at the lowest power setting. As the power was increased, the front rotated toward the horizontal and appeared to extend into the evaporator. Unfortunately, this could not be verified due to the layer of insulation covering the evaporator section. The temperature of the gas rose with that of the working fluid as the power was increased, but there was always an interface between the gas and vapor shown by the liquid crystals.

In the vertical position, the gas-vapor interface maintained a nearly constant angle of inclination regardless of the power being applied. A change in power did cause the interface to change its axial location however. It was also noted that the front appeared sharper and more clearly defined at the higher power levels.



Photograph of Liquid Crystals on Condenser - Horizontal

FIGURE 19



Photograph of Liquid Crystals on Condenser - Vertical

FIGURE 20

V. SUMMARY

A. SUMMARY OF RESULTS

In the first phase of the experiment, it was shown that this heat pipe operated in a nearly isothermal manner in the conventional mode in both the horizontal and vertical positions. The second and third phases of the experiment demonstrate the effects of the addition of a non-condensible gas which has a significantly larger molecular weight than the working fluid.

It was shown that in the horizontal position, the heavier non-condensible gas tended to collect along the bottom of the heat pipe while the working fluid filled the space above it forming a horizontal or nearly horizontal interface between the two substances.

In the vertical position, an interface also formed, but was much closer to being perpendicular to the axis of the heat pipe than the interface was with the heat pipe in the horizontal position. The front changed position on the heat pipe in accordance with the power being applied; the more power, the further it was from the evaporator. But, it did not change its angular orientation with the heat pipe axis. Also, the front was more diffuse at the lower power levels.

This experiment has clearly shown that neither the flat front theory nor the diffuse front theory adequately explain the nature of the vapor-gas interface region when the non-condensible gas used has a significantly higher molecular weight than the working fluid. It is apparent that the effects of gravity must be considered in the analysis of this particular situation.

B. RECOMMENDATIONS

It is recommended that further experimentation be conducted with this heat pipe. The heat pipe should be modified in several respects. First of all, more thermocouples should be placed along one side and the bottom of the heat pipe to enable a better measurement of the temperature variation in directions other than axial. This could and perhaps should be accomplished by removing some of the thermocouples along the top of the heat pipe. One side of the heat pipe should be left with no thermcouples attached to permit a clearer observation of the gas-vapor interface with a new coat of liquid crystals. In addition to the liquid crystals, the possibility of using infra-red photography to determine the location of the gas vapor interface and its orientation should be investigated.

Upon completion of the above work with the present charge of krypton, the heat pipe should be evacuated and recharged with the same amount of methanol and helium. The experimental procedure used in this experiment should then be duplicated with the lighter gas.

Returning to the problem of thermocouples, a considerable amount of effort was expended in this experiment to ensure the error associated with the thermocouples was minimized. However, due to some unknown cause or causes, some significant temperature variations along the heat pipe were indicated by the thermocouples installed on its skin. All of the wiring associated with the thermocouples was inspected closely to ensure no loose junctions or possibilities of temperature anomalies across junctions existed. Therefore, the error is thought to be caused either by the attachment procedure or by the wick not being held firmly enough against the pipe wall. The attachment procedure caused no error when tried on other thermocouples which were calibrated both prior and subsequent to attachment to a section of pipe. There was no way to check to see how well the wick was being held against the pipe wall. At any rate, a further attempt should be made to discover and eliminate the source of these temperature variations.

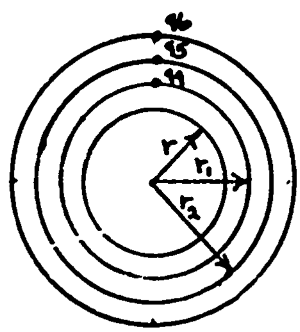
Finally, an analytical study should be made to determine ths nature and exact location and orientation of the gas-vapor interface since neither the flat front nor diffuse front theories adequately explain the observed phenomena.

APPENDIX A

CALCULATION OF POWER LOSS ERROR ANALYSIS

Power loss between the power supply and the heat pipe is considered negligible when compared to that lost through the insulation on the heat pipe.

A sample calculation of the power lost through the insulation follows:



Insulation Heat Loss Diagram

FIG. 21

$\Delta T = T_{94} - T_{95}$

K = thermal conductivity of insulation $\left(.02 \dfrac{BTU}{hr.\ ft\ °F}\right)$

L = length of insulation (1.1 ft)

r = radius of heat pipe (2 inches)

r_1 = radius to thermocouple 94 (25/16 inches)

r_2 = radius to thermocouple 95 (35/16 inches)

$$q = \frac{T}{\frac{\ln r_2/r_1}{2\pi KL}}$$

Sample calculation for nominal power set at 75 watts, horizontal position, conventional mode:

$$\Delta T = 25.5\ °F$$

$$q_{LOST} = \frac{(25.5\ °F)(.02\ \frac{BTU}{hr\ ft\ °F})(1.1\ ft)}{\ln(35/25)\ 3.412\ \frac{BTU}{watt\ hr}}$$

$$q_{LOST} = 3.1 \text{ watts}$$

Table 1 gives the results for other situations.

ERROR ANALYSIS

The greatest error in computing the heat loss occurs in the case where the heat pipe is being operated in the variable conductance mode in the vertical position at the lowest power input level.

The error in the heat loss is give by (see Ref. 5):

$$w_q = [(w_t\frac{dq}{dt})^2 + (w_k \frac{dq}{dk})^2 + (w_L \frac{dq}{dL})^2 + (w_r \frac{dq}{dr})^2]^{\frac{1}{2}}$$

where w_i is the uncertainty in the ith element,

$$\frac{dq}{dt} = \frac{(2\pi)(K)(L)}{\ln r_2/r_1} = \frac{(2\pi)(.02)(1.1)}{\ln (1.4)} = 0.41$$

$$\frac{dq}{dK} = \frac{(2\pi)(\ T)(L)}{\ln r_2/r_1} = \frac{(2\pi)(20.8)(1.1)}{\ln (1.4)} = 427.3$$

$$\frac{dq}{dL} = \frac{(2\pi)(\Delta T)(K)}{\ln r_2/r_1} = \frac{(2\pi)(20.8)(.02)}{\ln (1.4)} = 7.77$$

$$\frac{dq}{dr} = \frac{(2\pi)(\Delta T)(K)(L)}{r_1/r_2} [\frac{1}{r_2} - \frac{r_1}{r_2^{\,2}}] = (2\ )(20.8)(.02)(1.1)(1.4)$$

$$[.343 - .245]$$

$$= 0.39$$

$$w_q = [(0.41(1))^2 + (427.3(0.002))^2 + (7.77(0.06))^2 + (0.39(0.002))^2]^{\frac{1}{2}}$$

$$= \pm 1.06 \ w$$

$$\text{percentage error} = \frac{w_q}{q} = \frac{1.06}{22.5} = \pm 4.7\%$$

This error is indicative of the uncertainty in the heat loss computation. Therefore, the uncertainty in the heat actually being applied is considerably smaller, on a percentage basis, then that shown above.

	QUANTITY	ERROR BOUND
T	TEMPERATURE	± 1 °F
P	PRESSURE	± 0.2 psia
V	VOLTAGE	± 0.05 volts
R	RESISTANCE	± 0.001 ohms
Q	HEAT LOSS THROUGH THE INSULATION	± 4.7%
K	THERMAL CONDUCTIVITY OF THE INSULATION	± 0.002 $\frac{\text{BTU}}{\text{hr ft °F}}$
L	LENGTH OF INSULATION	± 0.06 ft
r	RADIUS OF INSULATION	± 0.02 ft

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