A FLEXIBLE HEAT PIPE

by

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PREFACE

My sincerest thanks are extended to the following individuals in appreciation of their generous assistance:

Dr. P. L. Miller, my major advisor, whose daily expressions of knowledge and advice have guided me throughout the last two years at Kansas State University.

My advisory committee, Professors P. E. McNall, R. L. Gorton, H. T. Bates and R. E. Williams, for their judgement and recommendations.

My wife, Peggy, who has given endless amounts of devotion, patience and encouragement to insure the success of this undertaking; and my daughter, Angela, for her numerous unique reflections upon the topic of this study.

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NOMENCLATURE

A_l	wick cross sectional area occupied by liquid (ft^2)
b	dimensionless constant
e	wick porosity factor
g	acceleration of gravity (ft/sec^2)
L	heat pipe length (ft)
$\dot{m}$	mass flowrate (lbm/sec)
N	number of cylindrical capillaries in wick
Δp_g	pressure loss due to difference in evaporator and condenser elevations (lbs/sq. in.)
Δp_l	pressure loss due to viscous action of flowing liquid (lbs/sq. in.)
Δp_p	capillary pumping pressure (lbs/sq. in.)
Δp_v	pressure loss due to viscous action of flowing vapor (lbs/sq. in.)
$\dot{Q}_{in}$	heat input at evaporator (BTU/hr)
$\dot{Q}_{out}$	heat rejected at condenser (BTU/hr)
R_r	Reynolds number for flow in radial direction
R_x	Reynolds number for flow in x direction based on r_v
r_c	capillary radius (ft)
r_v	radius of vapor region (ft)
r_w	radius of wick (ft)
ΔT_e	average evaporator temperature differential (^oF)
v	axial velocity (ft/sec)
v_r	radial velocity (ft/sec)

Greek

θ	wetting angle (radians)
λ	latent heat of vaporization (BTU/lb)
μ_l	viscosity of liquid (lbs/ft-hr)
μ_v	viscosity of vapor (lbs/ft-hr)
π	3.1417
ρ_l	density of liquid (lbs/ft^3)
ρ_v	density of vapor (lbs/ft^3)
σ	surface tension (lbs/ft)
ϕ	angle formed by axis of heat pipe and normal to gravitational field (radians)
ω	angle of curvature (radians)

Subscripts

c	capillary
e	evaporator
l	liquid
v	vapor

CHAPTER I

INTRODUCTION

During the recent years of accelerated technical advances, many engineers and scientists have focused their investigative efforts on the uncovering of additional knowledge concerning thermal energy and the theory of its transfer. The presence of this energy, especially in vast amounts, can be either useful, as in an electric metallurgical furnace, or undesirable, as in the core of a nuclear reactor. In some instances, the amount of thermal energy contained in a system is unimportant; however, in other situations, a thorough comprehension of the distribution of the thermal energy is quite necessary. Therefore, a tremendous number of investigations have been launched by researchers seeking to better their understanding of heat transfer. It was the results of one such investigation which provided the opportunity for the accomplishment of the experimentation described herein.

In 1963, a research group headed by G. M. Grover of the Los Alamos Scientific Laboratory developed a heat transfer device which had a capillary pump circulation system and which utilized both evaporation and condensation of a fluid inside a sealed container (1)*. This device was called a "Grover heat pipe", but it has become better known as simply a "heat pipe."

Since its discovery, the heat pipe has received wide acclaim as a reliable, lightweight appliance having a heat transfer capability "several thousand times that of the best heat conductors among metals (2)." It has been

*Numbers in parentheses designate References at the end of the thesis. (page 41) tested at temperatures which range from the cryogenic region to over 2000 C, and heat pipes have been subjected to as much as 30,000 hours of continuous operation without showing signs of failure (3,4). In addition, many geometric cross sectional shapes have been employed in heat pipe construction, and both straight and curved heat pipes have been built. However, it was brought to this writer's attention that of all the operating heat pipes, not one had the capability to accomodate changes in its configuration after it had been constructed. It is in this area, then, that the investigation outlined below has sought to provide additional information.

The objective of the program which this thesis describes was to construct and operate a flexible heat pipe, i.e., one which could be re-shaped or bent after assembly and still function properly.

CHAPTER II

GENERAL THEORY

The method of operation employed by a heat pipe is a relatively simple one and its characteristics have been the topics of many recent investigations. Detailed analyses have examined the function and configuration of the wick, heat pipe cross sectional shapes, and working fluids. The following discussion of heat pipe theory will be limited to the more important basic concepts.

Principles of Operation

The schematic of Fig. 1 illustrates, in cross section, an assembled heat pipe which is receiving heat at the evaporator, $\dot{Q}_{in}$, and discharging heat at the condenser, $\dot{Q}_{out}$. The heat input causes vaporization of the liquid with absorption of the latent heat of evaporation. The generated vapor then flows axially along the cylindrical volume with a slight decrease in the pressure. This pressure decrease causes condensation and release of the fluid's latent heat at the condenser. To complete the cycle, the condensed liquid is returned to the evaporator by capillary action through the porous wick along the inner pipe wall.

The insulated length of tubing between the evaporator and the condenser has been designated the adiabatic section. If heat is to be removed by radiation, the adiabatic section could be eliminated in order to maximize the condenser surface area without exceeding the effective length for the heat

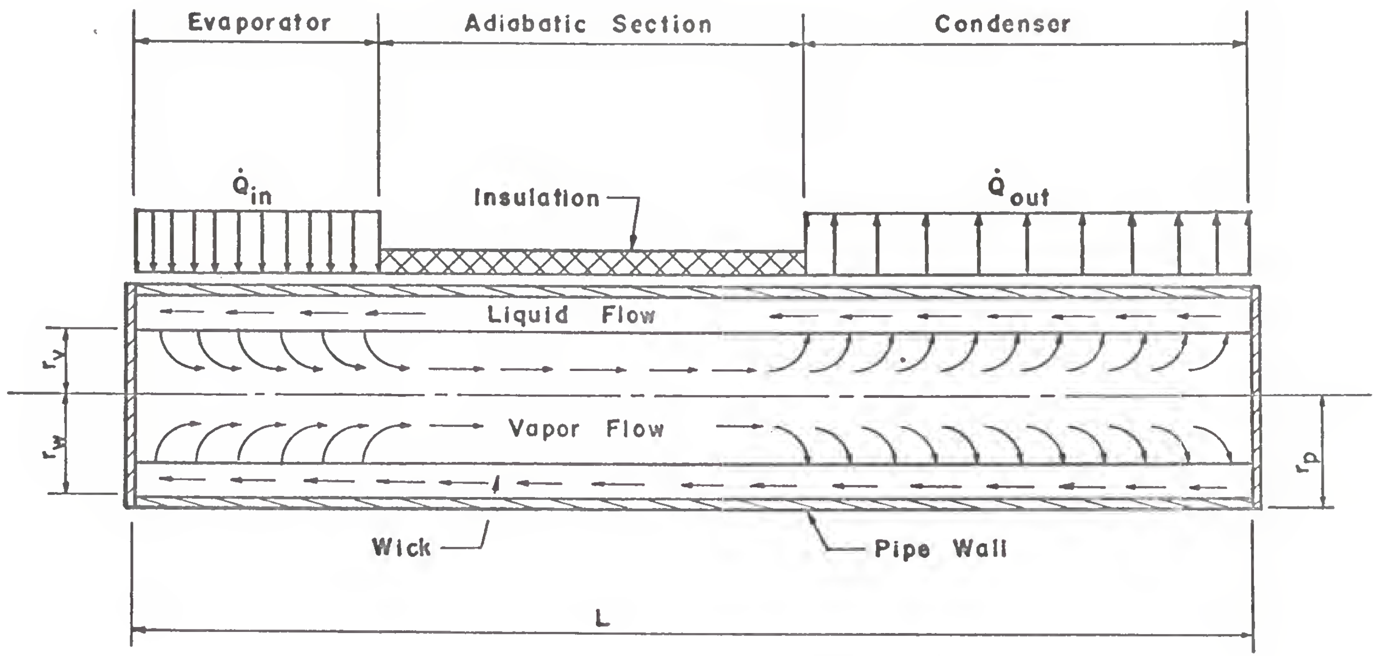

Figure I. A Schematic Diagram of the Heat Pipe.

pipe. However, if the wick will pump sufficient liquid over the total distance, inclusion of the adiabatic section in no way interferes with heat pipe performance.

Capillary Pumping and Pressure Losses

To analyze the heat pipe, a knowledge of the phenomena of capillary action and the dynamics of flow in a porous material is essential. Several recent works have been devoted to the treatment of flow through porous materials, including texts by Scheidegger (5) and Collins (6), and reports by Wageman and Guevara (7), Yuan and Finkelstein (8), and Knight and McInteer (9).

The capillary pumping pressure, Δp_p, is a result of the adhesive force of the liquid for the medium which contains it. This pressure differential is caused by the surface tension (intermolecular cohesive forces) of the liquid and can be expressed as

$$\Delta p_p = \frac{2\,\sigma\cos\theta}{r_c} \tag{2-1}$$

for cylindrical capillaries (10). If perfect interfacial wetting occurs in a capillary of given radius, then $\cos\theta = 1$ and the capillary pumping pressure is seen to be limited only by the surface tension of the liquid.

In applying Equation 2-1 to the heat pipe, it becomes apparent that this pumping pressure is the forcing function which causes the liquid to circulate. There are, however, several deterring forces which must be overcome before circulation will occur. These are: the pressure loss due to viscous action of the liquid flowing through the wick, Δp_l; the pressure loss due to the viscous action in the vapor region of the heat pipe, Δp_v; and the hydrostatic pressure drop in the wick, Δp_g, due to the elevation of the evaporator being greater than the elevation of the condenser. In equation form, this relationship may be expressed as

$$\Delta p_p \geq \Delta p_l + \Delta p_v + \Delta p_g. \tag{2-2}$$

Notice that the capillary pumping pressure must be equal to or greater than the sum of the pressure losses in order for the heat pipe to function. If the pressure losses become too great, circulation of the working fluid will be restricted.

It is now desirable to make a more detailed examination of each pressure loss term.

Farran and Starner (11) have expressed the pressure loss for a liquid flowing in a material having uniform circular capillaries. Darcy's Law for laminar flow was used to find

$$\Delta p_l = \frac{8 \mu_l \dot{m} L}{r_c^2 A_l} \tag{2-3}$$

where L is the length of the wick. Since the heat pipe transfers heat by the process of evaporation and condensation, the mass flowrate, $\dot{m}$, may be written as

$$\dot{m} = \frac{\dot{Q}_{in}}{\lambda} \tag{2-4}$$

where $\dot{Q}_{in}$ is the heat input at the evaporator, and λ is the latent heat of the working fluid. It should be noted also that the cross sectional area through which the liquid travels, A_l, may be expressed in terms of number of circular capillaries as

$$A_l = N (\pi r_c^2), \tag{2-5}$$

or in terms of the porosity of the wick material:

$$A_l = e\left[\pi(r_w^2 - r_v^2)\right]. \tag{2-6}$$

Next, consider the pressure drop in the vapor region of the heat pipe due to the viscous action of the vapor. For flow in the vapor region, the axial Reynolds number is expressed by $R_x = 2vr_v\rho_v/\mu_v$. The evaporation of fluid in the evaporator and condensation of the fluid at the other end of the heat pipe establish situations which are, respectively, analogous to flow with uniform injection and suction through a porous wall (12). These conditions were analyzed by Cotter (13) who determined solutions for two limiting cases in cylindrical heat pipes.

Cotter states that if the radial Reynolds number, R_r, is very low, the vapor pressure drop may be expressed as

$$\Delta p_v = \frac{4\mu_v L \dot{Q}_{in}}{\pi r_v^4 \rho_v \lambda}\ ;\ (R_r \ll 1.0). \tag{2-7}$$

This expression is Poiseuilli's equation for laminar flow in tubes (12).

In most practical cases, however, the radial velocity of the vapor in the evaporator and condenser will be large enough to cause the radial Reynolds number, $R_r = v_r r_v \rho_v/\mu_v$, to be much greater than 1. Cotter's solution for this limit was determined from the Navier-Stokes equation to be

$$\Delta p_v = \frac{\left[1 - \frac{4}{\pi^2}\right]\dot{Q}_{in}^2}{8\rho_v r_v^4 \lambda^2};\ (R_r \gg 1). \tag{2-8}$$

Rankin and Kemme (12) have noted that in the bracketed term of Equation 2-8, "the 1 represents the pressure drop in the evaporator while the $4/\pi^2$ term represents the pressure recovery in the condenser."

The final pressure drop to be examined is the hydrostatic pressure loss due to operation within a gravitational field. If the heat pipe has been oriented as shown in Fig. 2, where ϕ is the angle between the axial centerline and the normal to the gravitational field, the following expression is applicable:

$$\Delta p_g = \rho_l \, g \, L \sin \phi \tag{2-9}$$

where g is the local acceleration of gravity.

It is worthwhile to note that for a horizontal position of the heat pipe Δp_g will be zero.

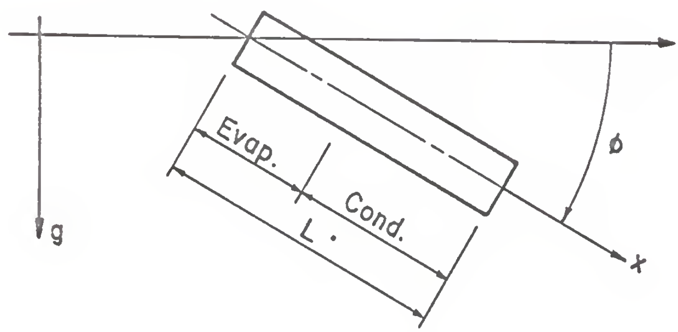

Figure 2. Heat Pipe Orientation in a Gravity Field.

Heat Transfer Characteristics

As was mentioned above, the addition of heat to the evaporator of the heat pipe vaporizes the liquid which has been supplied by the saturated wick. The energy required to cause vaporization, the latent heat, can then be convected to the condenser end of the heat pipe where it is rejected to the surroundings upon condensation of the vapor. This is by far the primary heat transfer method in the heat pipe, but there are also two other regions where heat transfer must be considered.

First, heat reaches the condenser section by conduction along the saturated wick. Evaluation of this quantity can be approximated by assuming the entire wick volume to be occupied by liquid and analyzing this as a conduction problem, using the thermal properties of the fluid. If the fluid used in the heat pipe is a liquid metal, this approximation very closely approaches the actual situation.

In the second case, the condenser may receive heat from the evaporator by conduction along the heat pipe envelope. Since the envelope is usually constructed of a material having a high thermal conductivity, this quantity becomes more significant as the thickness of the outer wall is increased. To minimize this effect, heat pipes are usually designed to have a containment vessel with as thin a wall as safety will permit.

Another heat transfer quantity which must be considered is the maximum amount of heat that can be transported by a particular heat pipe. In order to maximize this capability, the heat pipe should have maximum capillary pumping capacity and minimum pressure losses in the liquid and vapor regions. If the radii r_v and r_w have been previously specified for this heat pipe, the radius of the capillaries, r_c, becomes a significant consideration which will vary with the choice of a wick material.

In considering the optimum size of the pores in the wick material, Cotter (13) realized the necessity to strike a median point where both capillary pumping capacity and the viscous resistance of the flowing liquid are combined to provide maximum circulation. Both of these quantities are dependent upon the capillary radius but are affected oppositely by changes in r_c. To maximize the circulation rate, Cotter derived the following expression for the optimum pore radius for cases of uniform heat addition to and removal from the heat pipe:

$$r_c = \frac{b\,\mu_1\,\dot{Q}_{in}\,L}{4\pi(r_w^2 - r_v^2)\,\rho_1\,e\,\lambda\,\sigma\,\cos\theta} \tag{2-10}$$

where b is a dimensionless constant dependent upon the detailed geometry of the capillary structure. (For uniform cylindrical channels, b = 8.)

Cotter further states that this optimum value should be obtained when the pumping pressure increase, Δp_p, is equal to twice the pressure loss in the wick due to viscous action, Δp_1.

After optimizing all variables, Cotter expressed the maximum heat transfer in a heat pipe as

$$\dot{Q}_{max} = \frac{\pi\, r_w^3\,\lambda\,\sigma\,\cos\theta}{3\,L}\left[\frac{e\,\rho_v\,\rho_1}{3\,b\,\mu_v\,\mu_1}\right]^{\frac{1}{2}} \tag{2-11}$$

for values of $R_r \ll 1.0$, and

$$\dot{Q}_{max} = \frac{4\,\pi\,r_w^3}{3}\left[\frac{2\,\rho_v\,\rho_1\,e\,\sigma^2\cos^2\theta}{(\pi^2 - 4)\,b\,L\,\mu_1}\right]^{1/3} \tag{2-12}$$

for values of $R_r \gg 1.0$.

Selection of Working Fluid

In selecting a working fluid, the first prerequisite is that the fluid must be volatile over the temperature region within which it must function. Therefore, the working fluid dictates the range of operation of the device. As an example, water-filled heat pipes usually operate within the range 50-150 C, while sodium heat pipes will function between the approximate limits of 500 C to 900 C (12).

Vacuum-Fill System

In preparing the heat pipe for operation, it was necessary to evacuate the containment vessel and dry wick for several hours to insure removal of entrapped air and other gases. This outgassing procedure was accomplished by connecting the heat pipe to an NRC Model HK6-1500 high vacuum pump with $\frac{1}{4}$" copper tubing. A Pirani gage was used to leak test the apparatus, but no changes in the vapor pressure were noted when the joints and welds were painted with acetone. Diaphram type vacuum valves were included to close the vacuum line and also the fill line after the heat pipe had been charged with 200 ml of distilled, deionized water. This volume of water was calculated to be the amount required to saturate the wick and provide a 20% excess. After adding the fluid and disconnecting the heat pipe from the vacuum pump, preliminary testing was initiated.

Heat Input Apparatus

To supply power to and accurately control the resistance heater, a variable auto-transformer was connected to the 115 volt a.c. line current. The auto-transformer output was supplied to the heater through a 3 KVA step-down transformer which reduced the voltage by a 4:1 ratio. (See Fig. 3)

Heat Removal System

As indicated by Fig. 3, tap water was circulated through the concentric tube condenser in order to remove heat from the heat pipe. Included in the system to insure more accurate and reproducible results were a water softener and a fine particle filter. A variable area flowmeter was also used to provide a visual indication of fluctuations in the coolant flowrate. Finally, the flowrate was controlled by means of a metering needle valve.

Instrumentation

Electrical power input to the heating coil was determined by the product of the voltage and current readings which were obtained from alternating current meters as shown in Fig. 3. As the evaporator section received heat, its temperature was measured with six copper-constantan thermocouples which were embedded between the coils of heating ribbon at the locations shown in Fig. 6. Figure 6 also indicates the positions of 10 other Cu-Cn thermocouples which determined the temperatures along and near the heat pipe. At the condenser, a thermocouple was placed in both the inlet and outlet tube flow streams to record the temperature differential of the coolant. In addition, the ambient air temperature and the barometric pressure were also recorded.

Experimental Procedure

After sufficient pre-test experimentation had been accomplished, a test routine was established whereby comparable data were obtained for four representative angles of curvature along the length of the flexible heat pipe. The angles of uniform curvature chosen to compare the bending effects with respect to the straight line configuration were 45°, 90°, 135°, and 180°. Figure 7 contains a schematic of the configurations tested.

Startup

Before switching on the power to the resistance heater, a check of all thermocouples was made to insure that the emf output of each was the same for the room temperature heat pipe. The startup routine was then initiated.

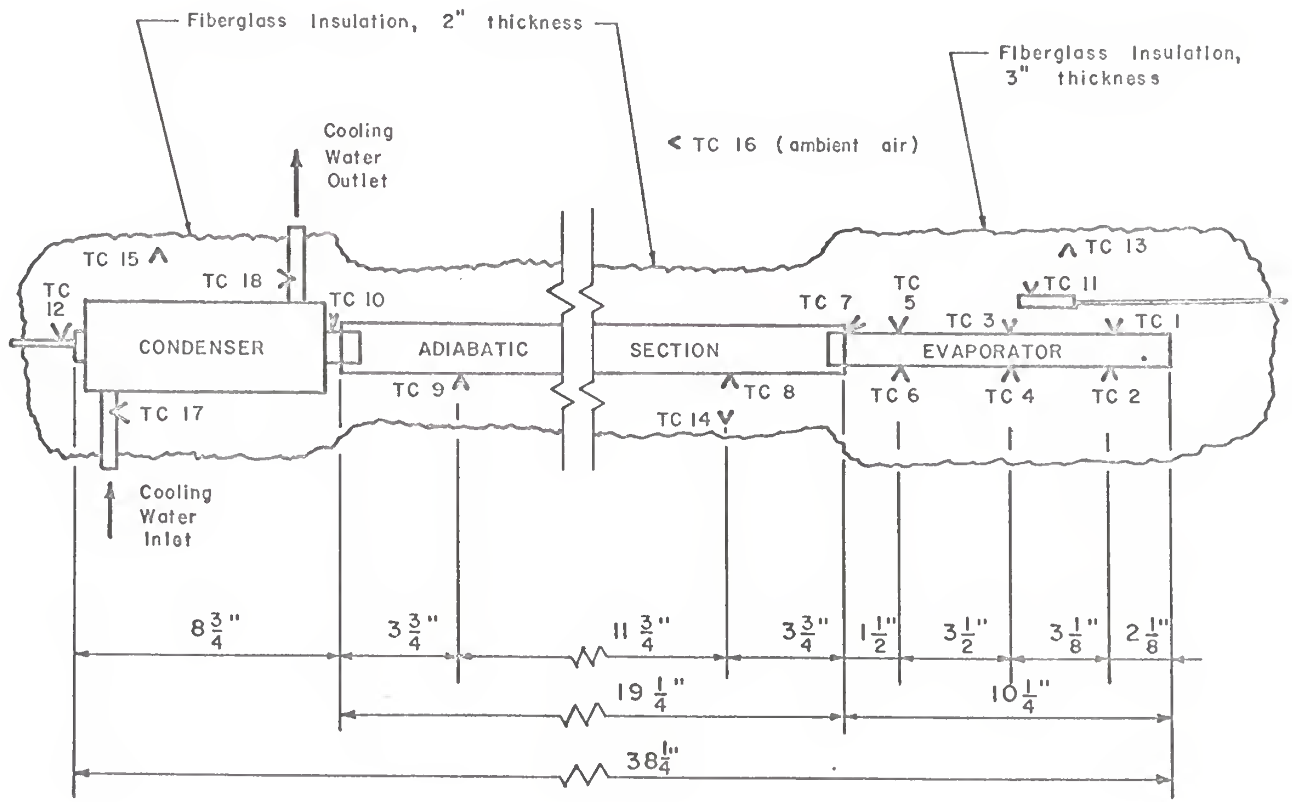

Figure 6. Thermocouple Locations.

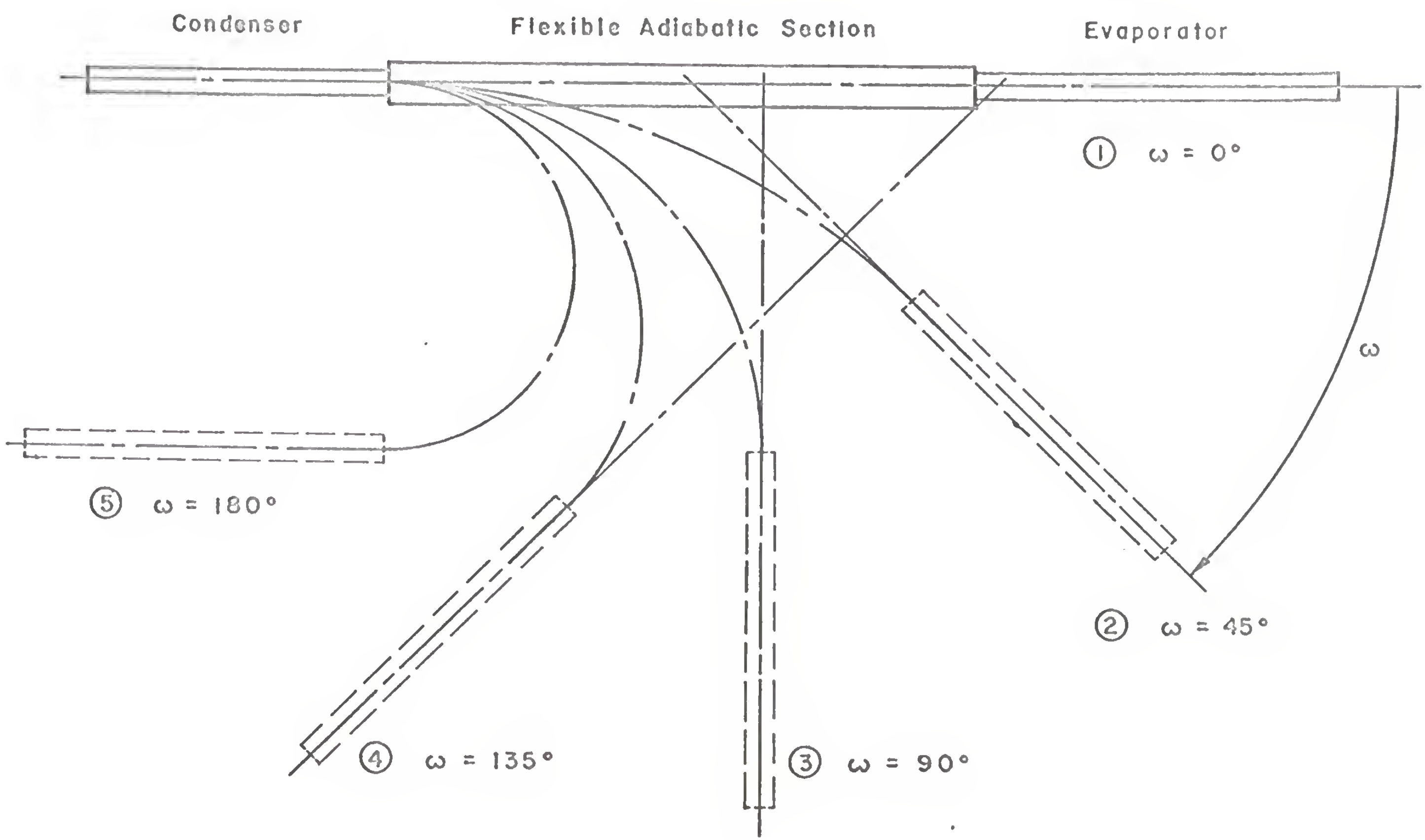

Figure 7. A Schematic Diagram of Test Configurations.

The power supply was adjusted to 9.0 amps, resulting in a voltage drop across the heating coil of 17.3 volts. Coolant flowing through the condenser was adjusted to about 4 lbs. of water per minute, and the evaporator temperature rise was observed on a millivolt-type recorder. During all tests, the startup procedure was performed with the heat pipe in the straight line configuration, i.e., position #1 in Fig. 7.

Initial Condition

Preliminary testing of the heat pipe revealed that a gradual temperature rise was experienced in the evaporator for any given $\dot{Q}_{in}$. This condition was attributed to a gradually increasing vapor pressure caused by the outgassing of the heat pipe material. (Application of a sealing compound and additional testing with acetone produced no evidence of a leak in the containment vessel.) It was then decided that the objective could best be accomplished by providing an initial condition of operation which could be repeated prior to the start of each test. The condition chosen for initial reference was the level of minimum vapor pressure, and lowest evaporator temperature, inside the heat pipe with a given load. This was accomplished by opening the valve on the heat pipe fill line momentarily to a vacuum system to remove the non-condensable gases. During and after this removal of gases, the decreasing evaporator temperature was observed for 3-4 minutes. This gas removal technique was repeated continuously until no further reduction in the evaporator temperature was noted.

Test Routine

Each test was officially started 2.67 hrs. after the time the lowest evaporator temperature had been reached. During the course of each test, all thermocouple emf's were measured with a Leeds and Northrup Model 8686 potentiometer. Current and voltage readings were recorded, and coolant flowrate was measured by using the stopwatch and balance technique. The room temperature, as well as all data mentioned above, were recorded every 30 minutes throughout each test.

At the start of each test, the heat pipe was positioned in its straight line configuration. After 3/4 hrs. had elapsed, the heat pipe curvature was changed to one of the positions shown in Fig. 7. After 1.5 hrs. in the curved posture, the heat pipe was returned to its original shape for approximately 1.0 hrs. at which time the test was terminated.

CHAPTER V

EXPERIMENTAL RESULTS

Operation of the flexible heat pipe emphasized the existence of two characteristics, one typical and one which was not entirely anticipated. First, a sizable temperature gradient along the outer wall of the heat pipe was observed, as expected. Figure 8 gives a typical example of the surface temperature versus length curve. The second characteristic, which was not expected, was the transient temperature exhibited by the heat pipe when operated at a constant heat input. This transient condition was probably caused by the outgassing of the Tygon[R] tubing and the fiberglass wick which increased the internal pressure of the system. A minute leak in the container is another possibility; however, previously mentioned leak detection methods failed to show a leak, and application of a sealing compound to the joints and welds failed to check the gradual temperature rise.

Because of the temperature rise with time, it was necessary to establish a reference for the operating heat pipe at the 0° curvature position with a constant load condition. With the heater at 155.7 watts, the average evaporator temperature rise was found to be about 1.9 F/hr. Figure 9 indicates the actual variance of evaporator temperatures recorded for one of the $\omega = 0^{\circ}$ tests, where ω represents the angle of curvature. In Fig. 10, the average evaporator temperature change has been plotted versus time for both of the $\omega = 0^{\circ}$ tests.

Upon establishment of a 0° reference, the heat pipe operating character-

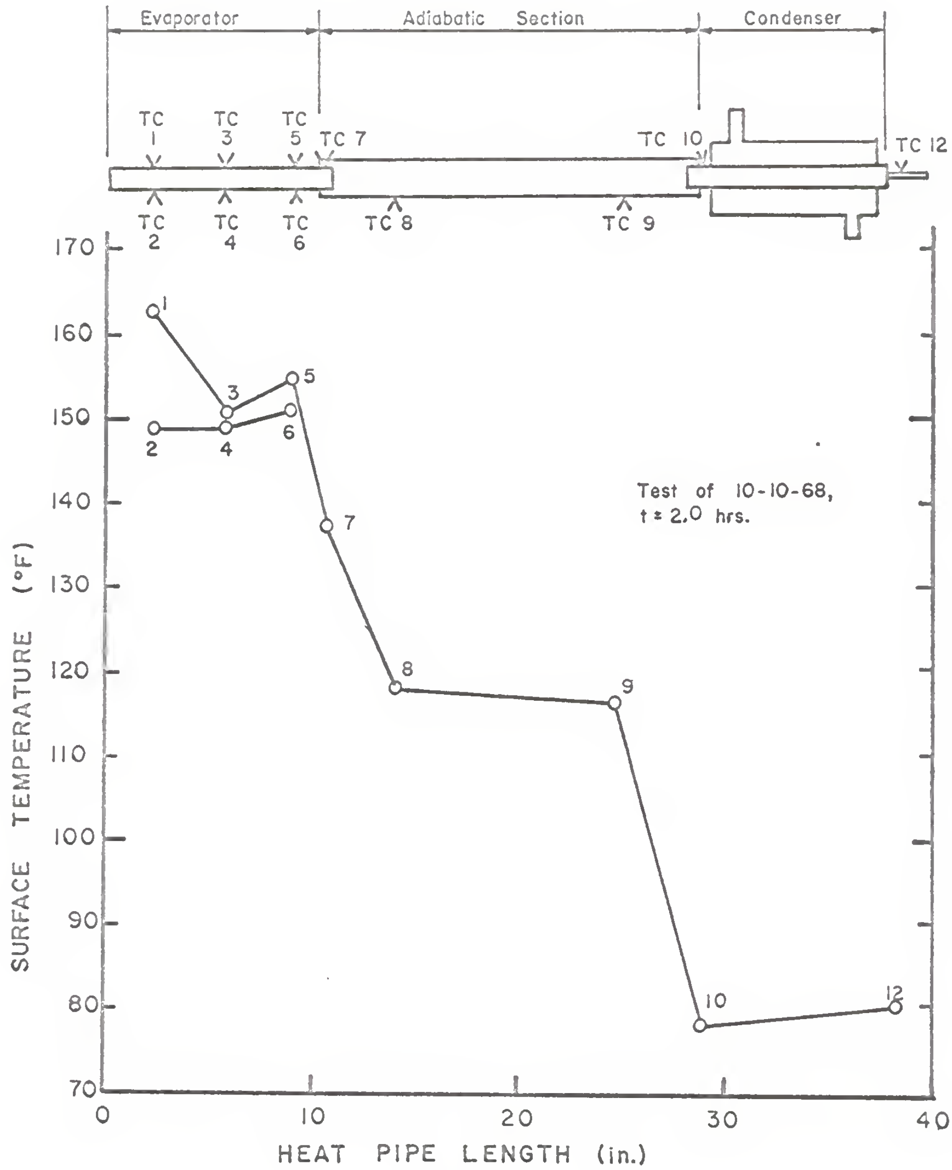

Figure 8. Surface Temperature versus Heat Pipe Length.

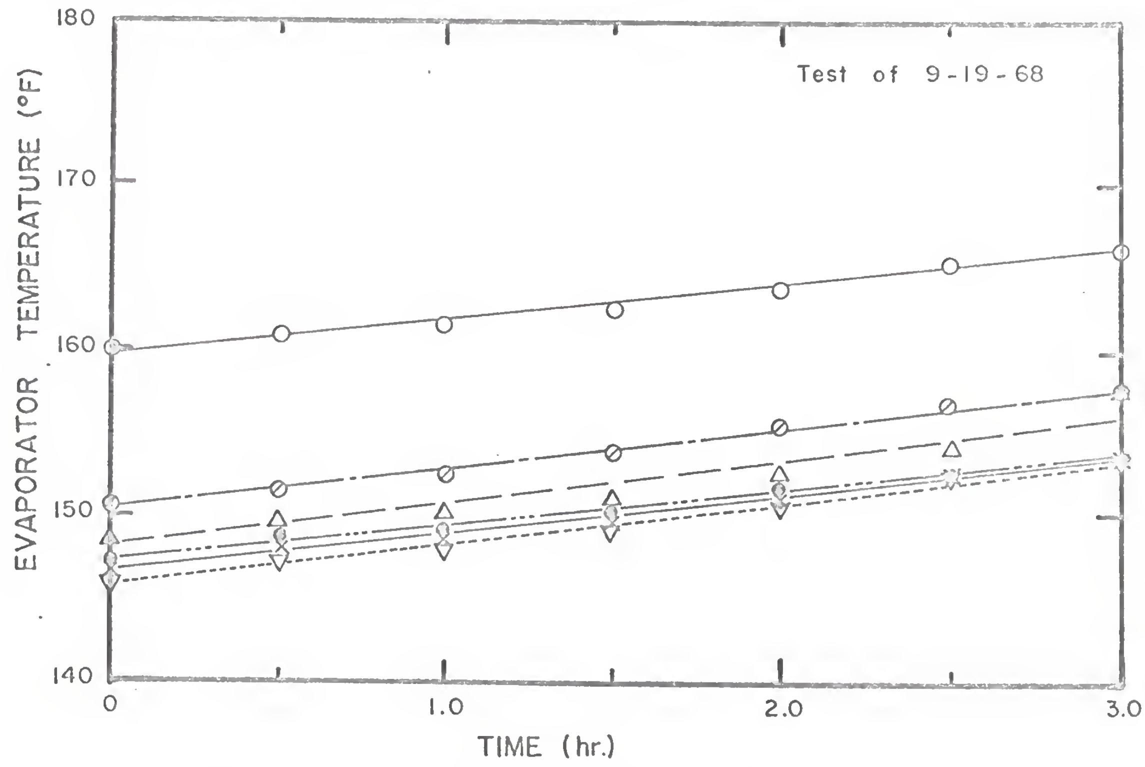

Figure 9. Evaporator Temperature versus Time for 0° Curvature.

Legend: TC 1 ○———○ TC 2 ×———× TC 3 △— — —△
TC 4 ●-----——● TC 5 ⊘—-—⊘ TC 6 ▽-------▽

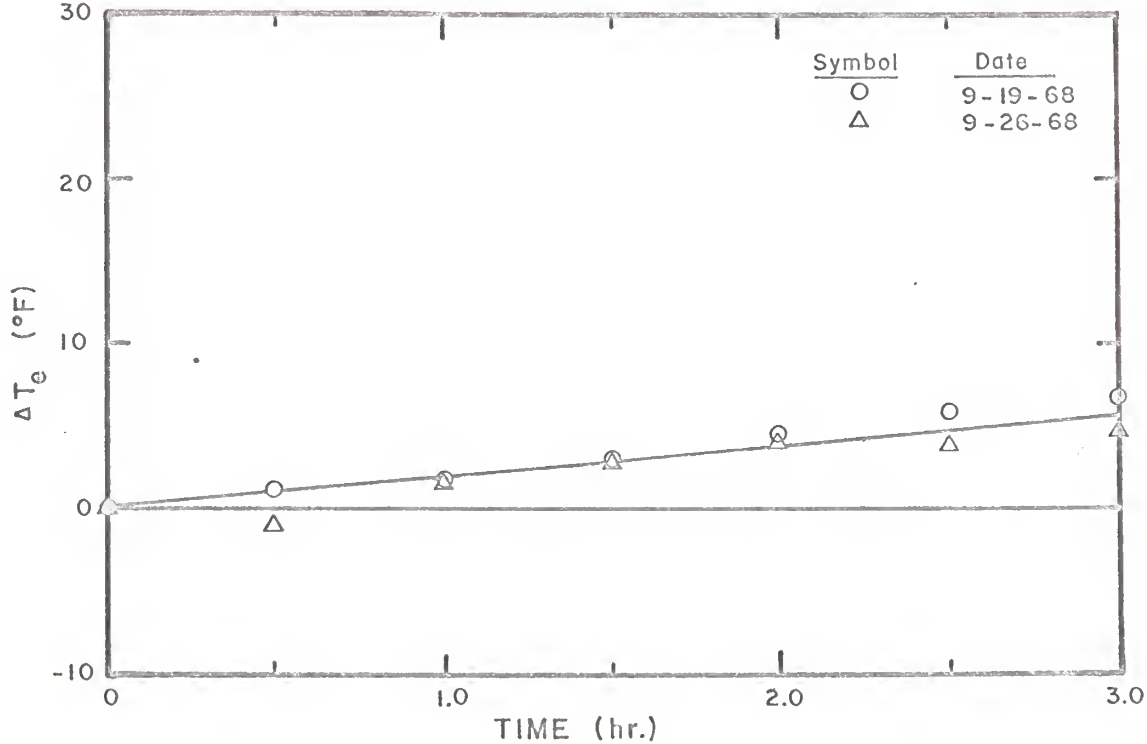

Figure 10. Average Evaporator Temperature Differential versus Time for 0° Curvature.

istics were obtained for values of ω equal to 45°, 90°, 135°, and 180°.

In Fig. 11, the values of average evaporator temperature rise which were obtained from the $\omega = 45^{\circ}$ tests have been plotted. For comparison, the 0° reference curve has been reproduced with a dashed line. Notice the agreement obtained when the heat pipe was changed to the curved position. Only after the configuration had been returned to $\omega = 0^{\circ}$ did a slight divergence appear.

Figures 12, 13, and 14 indicate the results from tests of the 90°, 135° and 180° configurations respectively, which also compare very favorably with the $\omega = 0^{\circ}$ tests. In several instances, there was a tendency for $\Delta \dot{T}_e$ to remain almost constant after the heat pipe was returned to the straight line configuration. This has been interpreted as an indication that the capillaries in the curved wick were slightly restricted, but the restriction was subsequently removed at $\omega = 0^{\circ}$. However, this restriction was not observed during every test as shown by Fig. 13.

It is noteworthy that, although slight differences in the average rate of evaporator temperature increase were obtained by these tests, the maximum deviation from the 0° reference curve was less than 2 F. This can be compared with the maximum difference obtained during the $\omega = 0^{\circ}$ tests which was also about 2 F. It is then reasonable to conclude that bending or re-shaping the flexible heat pipe does not have a significant effect on its ability to function as an efficient heat transfer device.

It is worthwhile, however, to compare the actual evaporator temperature obtained for each test configuration, and this has been shown in Fig. 15. The average evaporator temperatures at t = 2.0 hrs. were extracted from the test data and the arithmetic mean of these values has been indicated by a horizontal line.

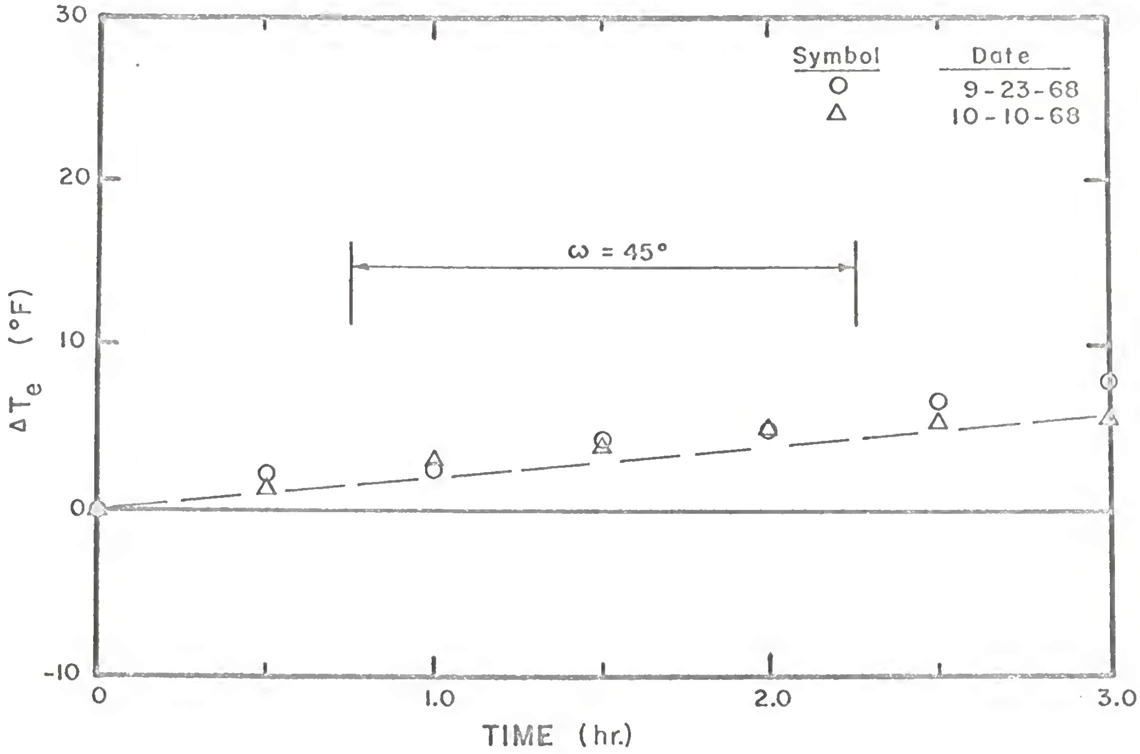

Figure 11. Average Evaporator Temperature Differential versus Time for 45° Curvature.

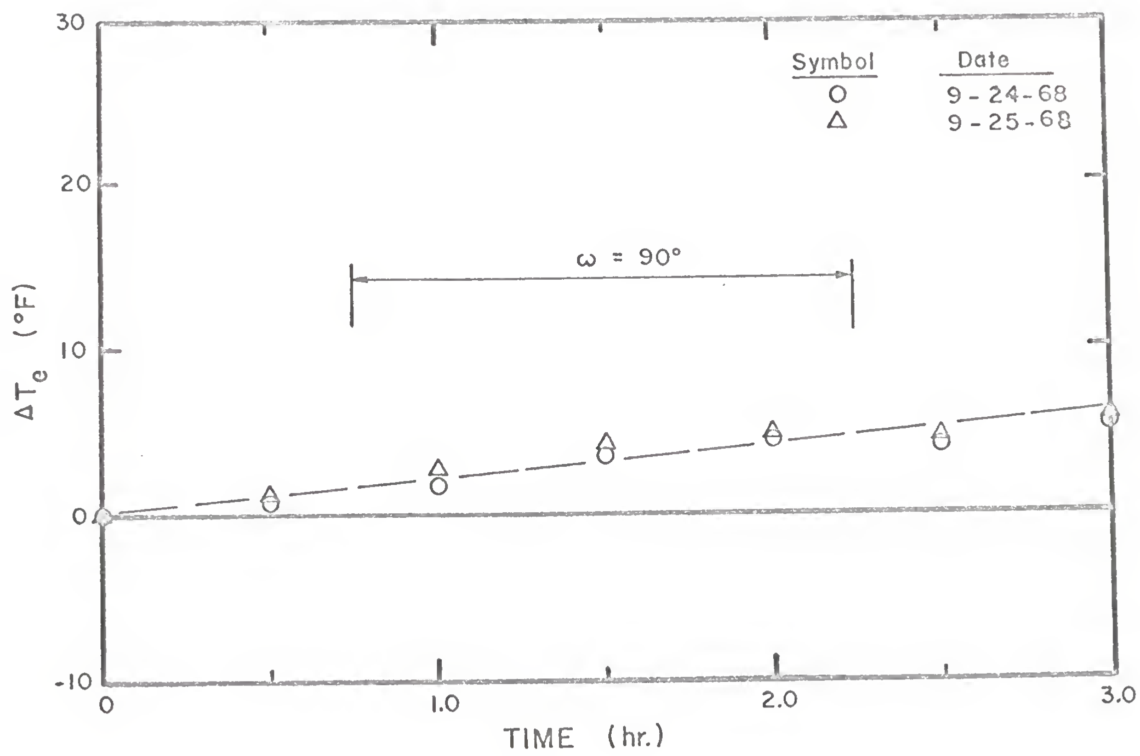

Figure 12. Average Evaporator Temperature Differential versus Time for 90° Curvature.

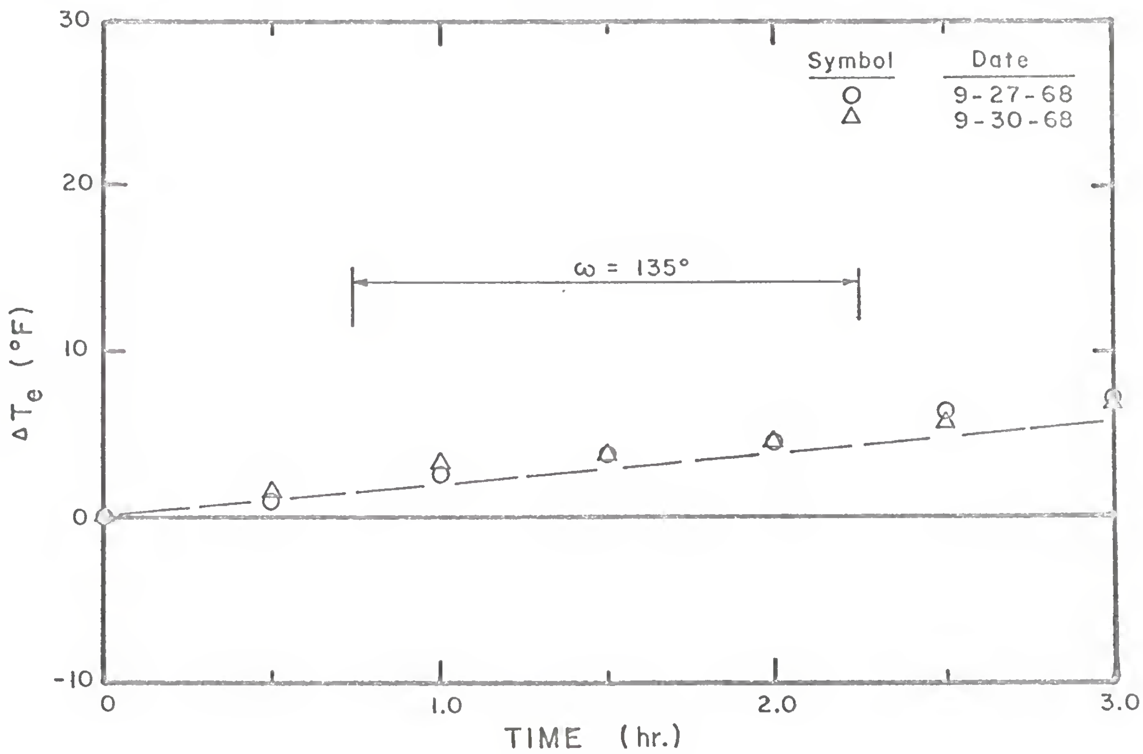

Figure 13. Average Evaporator Temperature Differential versus Time for 135° Curvature.

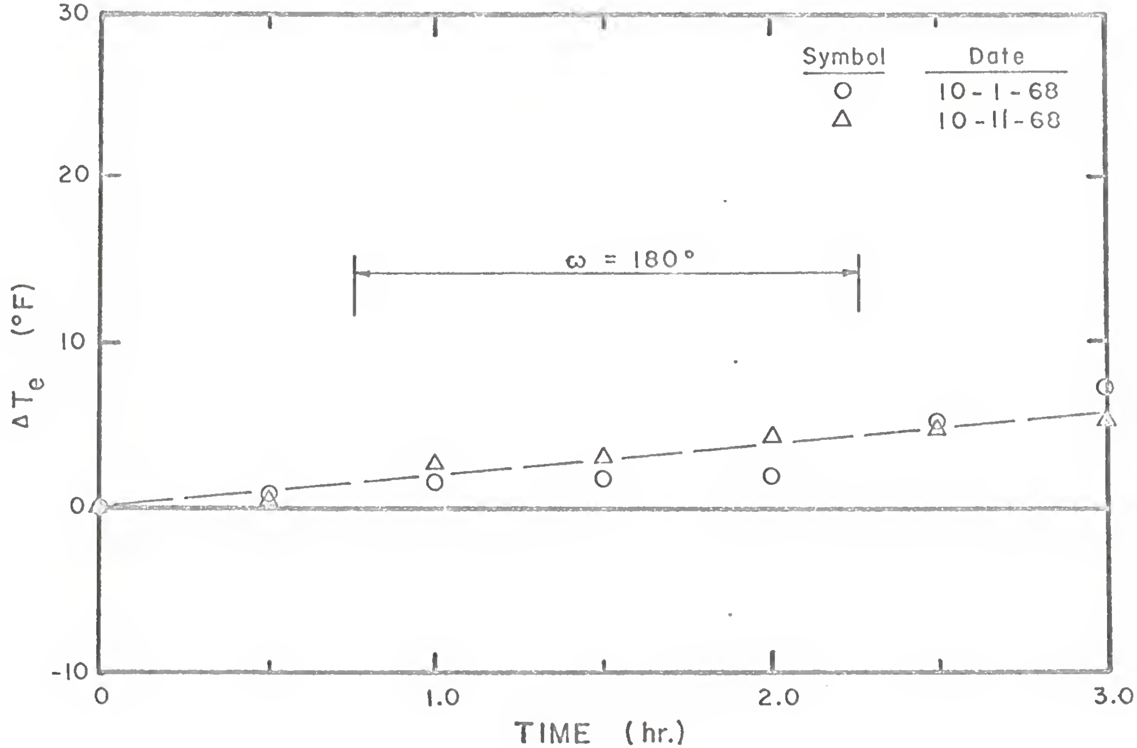

Figure 14. Average Evaporator Temperature Differential versus Time for 180° Curvature.

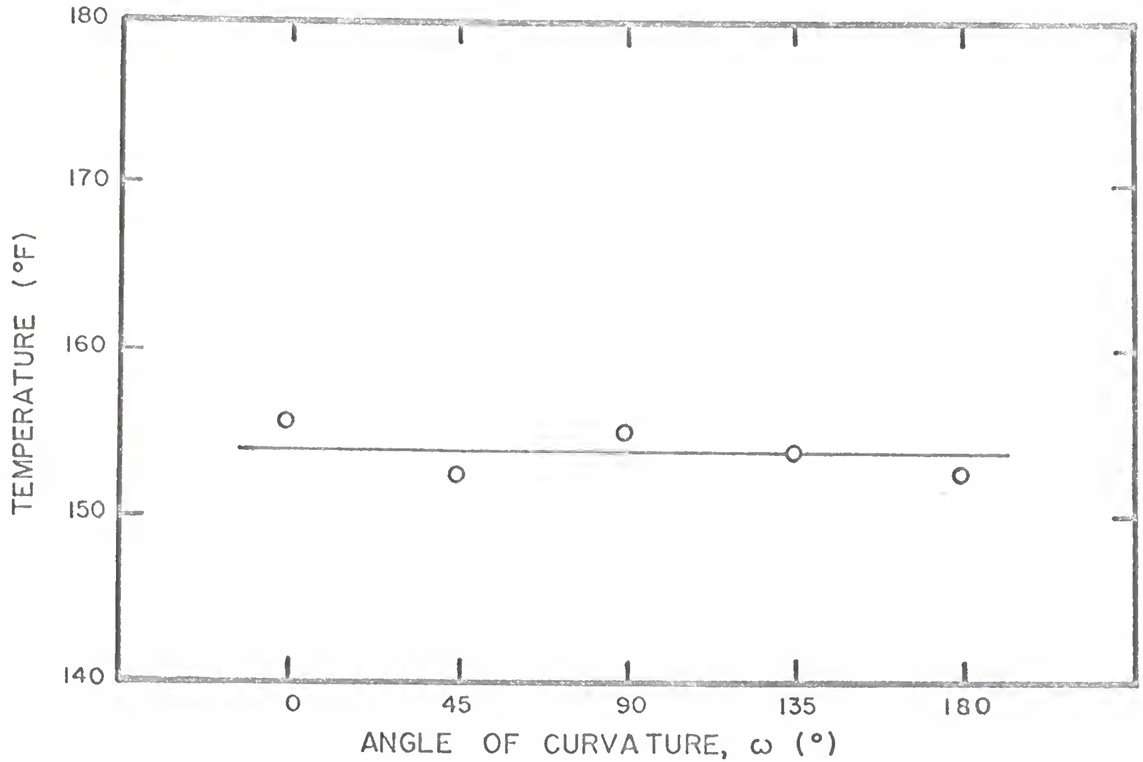

Figure 15. Average Evaporator Temperature versus Angle of Curvature for $t = 2.0$ hrs.

Several other characteristics of the flexible heat pipe also deserve mention. These traits include the evaporator temperature changes which were experienced due to changes in the input power, the maximum amount of heat which can be transported by the device, and the conditions which caused thermal runaway.

Two tests were made to establish the evaporator temperatures for various levels of heat input. Preliminary preparations for each test were the same as previously outlined, and each test was conducted over the same time frame.

For the first test, the power to the resistance heater was reduced from 155.7 watts to 70.2 watts approximately 20 minutes prior to t = 0 hr. At t = 0 hr., data were recorded and the power input was increased to 94.5 watts. Data were taken subsequently at half hour intervals, and upon completion of the readings for each interval, the a.c. current to the heater was incremented by 1.0 amps to a maximum value of 12.0 amps. Results are shown by the solid line in Fig. 16.

The second trial was conducted in the same manner as the first except that the power input to the heater was varied in decreasing increments from the maximum value to the minimum. As the dashed line in Fig. 16 indicates, the results obtained were slightly different from the first trial. This can be accounted by the fact that the outgassing rates of the heat pipe materials varied with the temperature and pressure inside the heat pipe. It is estimated that an average value of the two curves in Fig. 16 would be the most accurate indication of the heat pipe's actual performance characteristic.

The maximum heat transfer capability of the test apparatus was not tested, as such, but it was evident that sustained operation at power levels above 280 watts could be detrimental to the device. Evaporator temperatures in excess of 190 F were obtained, and it was felt that further temperature

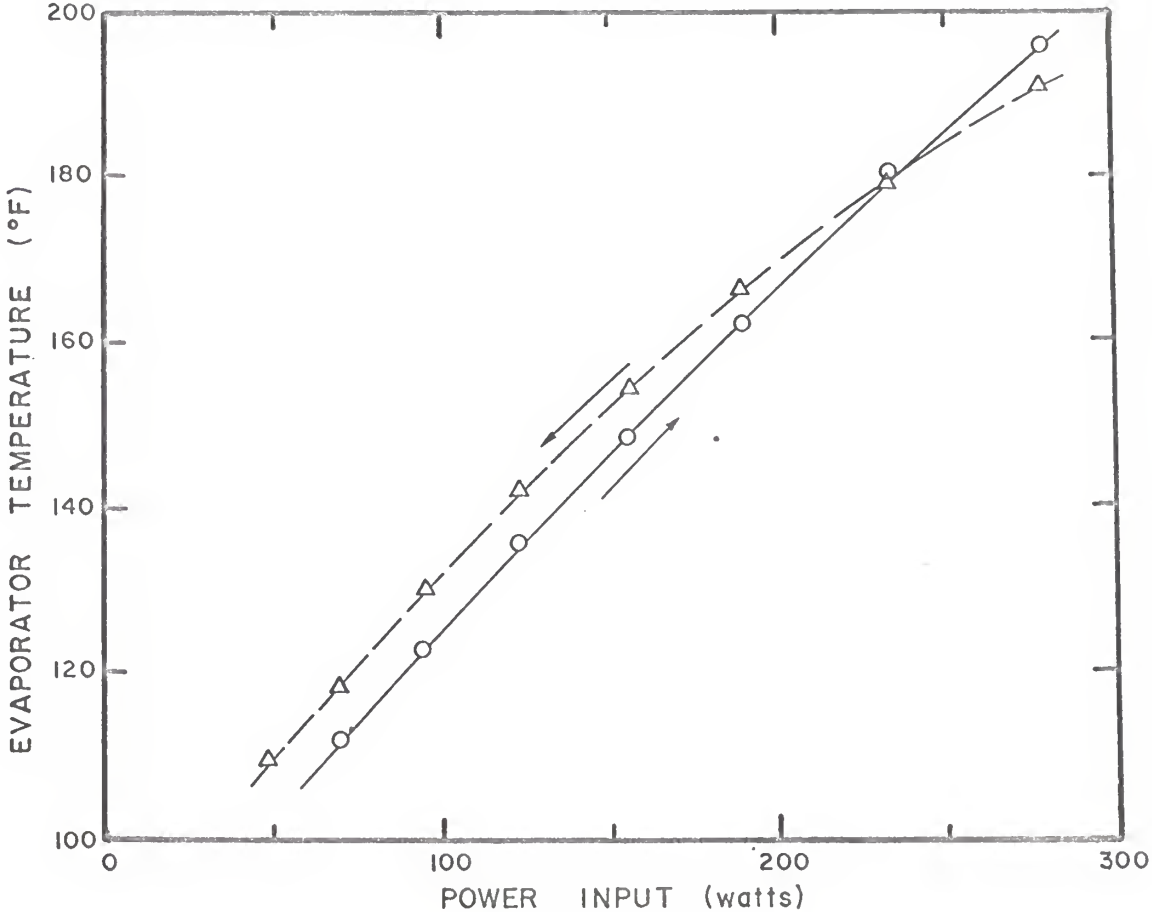

Figure 16. Evaporator Temperature versus Power Input.

increases would only serve to advance the risk of rupture in the flexible tubing.

Thermal runaway in the heat pipe did not occur during the comparative tests which were described above. During the preliminary tests, however, it was noted that changes in curvature when operating the heat pipe at temperatures in the 170-190 F range would sometimes cause an excessive temperature increase at specific points, usually TC 1 and TC 5. In most instances, the wick was able to recover and supply sufficient working fluid to the evaporator before the thermostatic shutdown device was activated. Several times, the amount of fluid pumped into the evaporator was so limited that almost all of the input thermal energy had to be absorbed by the wick and the copper tubing. This necessarily caused a very rapid increase in the evaporator temperature, a condition which is called thermal runaway.

It is also noted that, during the preliminary experimentation, thermal runaway was more likely to occur if the heat input to the device was increased sharply. For example, changing the power input to the heater from 156 watts to 278 watts would probably cause a runaway condition; increasing the power by smaller increments at regular intervals, as described previously, did not cause the heat pipe temperature control to be lost. Therefore, it can be stated that sudden loading of the heat pipe was sometimes detrimental.

Error Discussion

The purpose of this section is to discuss the possible errors which may have affected the results described above. These errors include fluctuations in the input power to the resistance heater, heat losses through the insulation, thermocouple wires, and heater wires, and the effects of bending upon the wick structure.

Although the amount of power input was closely attended, fluctuations in the line voltage supplied to the experimental area did occur. When these fluctuations were noted, the auto-transformer was adjusted accordingly. The variance in the current supplied to the resistance heater was estimated to be ± 0.2 amps, thus making the maximum difference in power input approximately $\pm 2\%$. It should be pointed out however, that these fluctuations, when they did occur, were always of short duration (less than 10 minutes), and the deviation from the total power input for each trial was probably much less than the $\pm 2\%$ instantaneous difference.

Heat losses from the assembly were estimated for conduction of heat through the insulating material, along the thermocouple wires and through the heater power cables. Losses through the insulation around the evaporator and the flexible section were estimated to be 6 BTU/hr. and 8 BTU/hr. respectively. At the condenser, the gain of heat from the ambient air was estimated to be approximately 2 BTU/hr. Thus, the net loss of heat through the insulation was about 12 BTU/hr., or 2% of the heat input. Conduction of heat along the thermocouple wires was considered negligible; however, it was estimated that 2 BTU/hr. was conducted away by the heater cables.

Inside the heat pipe, a situation probably developed in the wick to cause a slight difference between the recorded temperatures and the actual temperature of the circulating fluid. Repeated bending and unbending of the heat pipe probably caused the wick to pull away from the wall in the evaporator and condenser sections. This condition would increase the resistance to the transfer of heat in both sections and would be characterized in the evaporator by areas of slightly elevated temperature. In the typical data sheet of Appendix B, readings obtained for TC 1 and TC 5 indicate the existence of such a condition during the 45° curvature test.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

It is concluded from these experimental results that a flexible heat pipe can be constructed so that changes in its linear configuration do not affect its ability to function as an efficient heat transfer device. This experiment tested only one specific example, i.e., a water-filled horizontal heat pipe in a standard gravity environment; however, the results of this investigation are equally applicable to heat pipes which utilize other working fluids and which must operate under different environmental conditions. Spacecraft have become a particularly attractive field for application of these flexible principles since the reduced gravitation field would permit flexibility outside the horizontal plane without adversely affecting the pumping capability of the wick.

For future investigations, it would be desirable to test various other flexible materials, such as bellows tubing, so that the temperature range of operation would not be as limited as it presently is by the physical properties of the containment vessel. Further treatment of flexible wicking materials would also be most helpful.

In addition, a recommendation is made to investigate the size limits of a flexible heat pipe. Miniature versions of the test device could have a tremendous potential for application in an area such as electronic component packaging where size is an important consideration.

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APPENDIX

APPENDIX A

SUPPORT EQUIPMENT LIST

The following is a list of the supporting equipment and instruments which were required for this investigation.

1. Vacuum Pump, National Research Company, model no. HK6-1500.
2. Vacuum Pump, Central Scientific Company, catalog no. 9051, ME 1179.
3. Mercury Manometer, King Engineering Corporation, model no. BUS-36, serial no. 8322.
4. AC Ammeter, Hallmark Standards, model SPFB, serial no. H1070.
5. AC Voltmeter, Ballantine Laboratories, Inc., model 300G, serial no. 924.
6. Flowmeter, Fischer and Porter Company, serial no. V3-1518/46.
7. Transformer, 3 KVA, serial no. 6904702.
8. Thermostat, 60-250 F, Edwin L. Wiegand Company, catalog no. AR-2514 T.
9. Soft Water Bottle.
10. Water Filter, Pall Corporation, model no. 30D, ME 3883.
11. Variable Auto-Transformer, Standard Electrical Products Company, scale 0-140 volts.
12. Micro-Regulating Stem Valve, Whitey Research Tool Company, type 10RS8.
13. Stopwatch, 30 minute, Galco, ME 123.
14. Scale, Fairbanks Scales.
15. Weigh Barrel (5 gal.).
16. Millivolt Potentiometer, Leeds and Northrop Company, model 8686, serial no. 1629282.

17. Millivolt Recorder, Leeds and Northrop Company, model R-504-L-857, ME 1319.

APPENDIX B

TABLE I

Typical Data - Test of 10-10-68

Current: 9.0 amps | Voltage: 17.3 volts | Power Input: 155.7 watts

Item	Time (hrs.) 0	0.5	1.0	1.5	2.0	2.5	3.0
Curvature (°)	0	0	45	45	45	0	0
TC 1 (°F)	157.3	158.5	159.9	160.6	161.7	162.7	163.1
TC 2 (°F)	144.1	145.5	147.1	147.8	148.8	149.6	150.1
TC 3 (°F)	145.3	146.7	148.1	148.6	149.8	151.0	151.4
TC 4 (°F)	144.7	145.9	147.6	148.1	149.3	150.1	150.4
TC 5 (°F)	148.3	149.6	151.7	153.6	154.4	154.0	154.2
TC 6 (°F)	146.2	147.5	149.8	150.2	151.3	150.9	150.8
TC 7 (°F)	131.8	133.3	135.1	135.8	136.9	137.5	137.9
TC 8 (°F)	113.0	114.5	115.5	116.5	117.4	118.3	119.0
TC 9 (°F)	111.0	112.4	114.6	115.4	116.4	115.8	116.5
TC 10 (°F)	77.4	77.7	78.0	78.2	78.0	78.9	78.8
TC 11 (°F)	142.0	143.8	144.8	145.5	146.3	146.6	147.1

TABLE I -- Continued

Item	Time (hrs.) 0	0.5	1.0	1.5	2.0	2.5	3.0
TC 12 (°F)	84.8	83.9	82.0	80.8	80.3	81.4	80.4
TC 13 (°F)	85.0	85.8	86.9	87.1	87.3	86.8	87.1
TC 14 (°F)	79.7	80.1	79.9	79.9	80.4	80.7	81.2
TC 15 (°F)	71.9	72.3	73.0	73.1	73.4	73.5	73.7
TC 16 (°F)	73.1	73.7	73.1	73.3	74.0	73.9	74.2
Cooling Water Inlet (°F)	61.4	61.6	61.2	61.1	61.1	61.3	61.4
Cooling Water Outlet (°F)	63.6	63.5	63.5	63.0	62.9	63.4	63.3
Room Temperature (°F)	73.5	73.7	73.5	73.7	74.0	74.2	74.5
Flowrate (lbs./min.)	4.27	4.40	4.42	4.34	4.62	4.25	4.37

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A FLEXIBLE HEAT PIPE

by

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B. S., University of Missouri at Rolla, 1964

AN ABSTRACT OF A THESIS

submitted in partial fulfillment of the

requirements for the degree

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Scope and Method of Study: The heat pipe is a highly efficient device which transfers heat by means of a closed two phase system and utilizes a capillary pump liquid return process. Heat pipes have generally been constructed to have a fixed geometrical shape; however, no experimental evidence prior to this thesis has been presented concerning the operation of a flexible heat pipe which could have its shape modified to fit several applications.

The characteristics of a flexible heat pipe were investigated by constructing and operating a cylindrical heat pipe made of copper and Tygon^R (plastic) tubing. A supple wick of woven fiberglass was formed by rolling the matted material into a hollow cylinder. Distilled water was selected to be the working fluid. The apparatus was tested in a standard gravity environment, and bending of the heat pipe was restricted to the horizontal plane. The following representative angles of curvature were examined: 0°, 45°, 90°, 135°, and 180°.

Findings and Conclusions: Experimental data were obtained by monitoring the temperature of the heat pipe evaporator at six positions along its length. Changes in the observed temperatures for the curved configurations, as compared with the straight heat pipe, were so small that they could not be attributed to the bending of the heat pipe.

Evaluation of the recorded data has shown that bending does not have a significant effect on the ability of the flexible heat pipe to

efficiently transfer thermal energy. It was noted, however, that temporary disturbances did occur when the fiberglass wick's capillary structure was distorted by bending; but only under extreme circumstances did these disturbances cause thermal runaway.

MAJOR PROFESSOR'S APPROVAL _P. L. Miller_