DEPARTMENT OF MECHANICAL AND INDUSTRIAL ENGINEERING
PHYSICAL ENVIRONMENT UNIT
ENGINEERING EXPERIMENT STATION
UNIVERSITY OF ILLINOIS AT URBANA - CHAMPAIGN
URBANA, ILLINOIS 61801

EXPERIMENTAL INVESTIGATION OF THE PERFORMANCE OF VARIOUS WICK CONFIGURATIONS IN SINGLE- AND TWO-FLUID HEAT PIPES OPERATING IN THE GRAVITATIONAL FIELD

by

D. L. HUNSBERGER

J. C. CHATO

Technical Report No. ME-TR-187

October 1970

Supported by
National Aeronautics and Space Administration
under
Grant No. NGR-14-005-103

EXPERIMENTAL INVESTIGATION OF THE PERFORMANCE OF VARIOUS WICK CONFIGURATIONS IN SINGLE- AND TWO-FLUID HEAT PIPES OPERATING IN THE GRAVITATIONAL FIELD

by

D. L. Hunsberger

J. C. Chato

Technical Report No. ME-TR-187

October 1970

Supported by

National Aeronautics and Space Administration

under

Grant No. NGR 14-005-103

ABSTRACT

In Part I, wicking material tests were made on Refrasil No. C100-28. The tests were run on 9-in. X 1-in. Refrasil strips. Displacement-time curves were extrapolated for predicting the performance of a 21-3/4 in. heat pipe.

In Part II, heat pipe tests were run with a well-defined wick length of 21-3/4 in. and a total width of 7 in. The same Refrasil was the wicking material. An open-ended dewar housed the heat pipe system which consisted of heat input, mass transfer, and heat removal sections. Two electric heaters supplied heat input, while circulating water was used for heat removal.

Both Parts I and II showed water to be a much better operating fluid than ethyl alcohol or 50 percent ethyl alcohol by weight. Ethyl alcohol appeared to be only slightly better than the 50 percent mixture. At zero degrees the maximum heat transfer capacities were 15, 4, and 2 watts, respectively, for the three fluids. The predicted wattages from Part I were generally higher due to greater ease in saturating the wicking material with fluid.

A gap effect created by sewing two layers of wicking material together greatly enhanced the heat pipe performance. At zero degrees, water transferred over 80 watts, as compared to 15 watts previously.

TABLE OF CONTENTS

LIST OF FIGURES

1. INTRODUCTION

Future space exploration will entail more and more extravehicular activity. Consequently, it becomes necessary that the astronaut's space suit provide him with a suitable living environment. One necessity is provision of a satisfactory thermal environment.

Present systems utilize liquid-cooled undergarments which need an intricate system of valves, pumps, and auxiliary equipment. Heat pipes could be used to create a system of less complexity and greater efficiency. The complexity of the system could be reduced because a heat pipe is a closed system which needs no resupply or adjustment after assembly. No pumps, compressors, or auxiliary equipment are needed during operation. Efficiency is increased because a heat pipe commonly transfers heat on the order of 100-1000 times faster than the best conducting metals.

A space suit thermoregulatory system must be capable of transferring heat from the astronaut's body to an area on the suit where the heat can be rejected either to space by radiation or to a porous plate sublimator. In performing such tasks, there are no restrictions on the orientation of a heat pipe. Wicking materials are used to provide fluid travel from the heat rejection areas to the heat input areas, even against the force of gravity.

At temperatures below 32°F, a fluid such as alcohol or an alcohol-water mixture could be used. A mixture might compensate for the low latent heat of the alcohol since the two fluids will tend to separate and occupy opposite ends of the heat pipes. Thus, the distance across which the alcohol must transfer heat would be reduced.

In space suit applications, the heat pipe configuration and heat transfer rates indicate that the ability of the wick to transfer the fluid will be more of a limiting factor than the pressure drop due to vapor flow. Consequently, this study was restricted to the performance of the wick.

During the last few years, there has been considerable activity in the study of heat pipes as indicated by the numerous technical sessions on the subject at the meetings of several professional societies, such as ASME [1]†, AIAA [2], and other combined meetings [3]. A detailed literature survey will not be presented here, however; only references which have direct bearing on this study will be given.

†Numbers in brackets refer to entries in REFERENCES.

2. HEAT PIPE OPERATION

A heat pipe consists of three distinct sections. First, a heat input section or evaporator section evaporates the operating fluid. Second, the small pressure gradient along the heat pipe forces the vapor through the mass (vapor state) transfer section to the third section, the heat rejection or condenser section. Here, the vapor condenses before returning to the heat input section via the mass (liquid state) transfer section. Capillary action or gravitational force is the means of returning the fluid.

Mass return to the heat input section by capillary action requires the use of a wicking material. Commonly used wicking materials are fine wire screens, porous solid materials, and natural or synthetic cloths.

3. GENERAL HEAT PIPE THEORY

For the purpose of this work, a relatively simple theory, such as given in [4,5], is quite adequate. This theory will only be briefly summarized here.

The temperature gradient along a wick limited heat pipe is very small due to the small pressure gradient. Consequently, conduction, radiation, and sensible convection heat transfer will all be negligible compared to heat transferred as latent heat of vaporization. Assuming all heat transferred is due to latent heat, the heat transfer rate (Q) is the product of mass flow rate ($\dot{m}$) of fluid in the wick and latent heat of vaporization (h_{fg}) of the fluid at the operating temperature of the system. The maximum heat transfer capability of the heat pipe is then $Q_{max} = \dot{m}_{max} h_{fg}$ where $\dot{m}_{max}$ may be interpreted as the flow rate occurring when the wicking material is dry at the evaporator end.

The following four basic parameters may be used to find an expression for the maximum mass flow rate. They are

(i) the capillary pumping head of the wick (ΔP_c),

(ii) the vapor pressure drop (ΔP_v),

(iii) the liquid viscous drag (ΔP_L), and

(iv) the gravity head (ΔP_g).

The acceleration effects in the wick can be ignored [5]. From a balance of pressure heads, the equation for the operation of a heat pipe becomes

$$\Delta P_c \geq \Delta P_v + \Delta P_L + \Delta P_g \tag{1}$$

Substituting appropriate terms for the pressure heads, Eq. (1) becomes

$$\frac{2\sigma}{r} \geq \frac{\mu L\dot{m}}{\rho KA} + \rho g L \cos \phi \tag{2}$$

where ΔP_v has been assumed to be negligible. $2\sigma/r$ is a maximum value for ΔP_c which occurs where the wick becomes dry. Rearrangement of Eq. (2) gives

$$\dot{m} \leq \frac{\rho KA}{\mu L}\left(\frac{2\sigma}{r} - \rho g L \cos \phi\right) \tag{3}$$

Therefore,

$$\dot{m}_{max} = \frac{\rho KA}{\mu L}\left(\frac{2\sigma}{r} - \rho g L \cos \phi\right) \tag{4}$$

and

$$Q_{max} = \frac{\rho KA}{\mu L}\left(\frac{2\sigma}{r} - \rho g L \cos \phi\right) h_{fg} \tag{5}$$

4. EXPERIMENTAL WORK, PART I

APPARATUS AND TEST PROCEDURE

The apparatus used for testing the wicking material Refrasil No. C100-28 is shown in Fig. 1. The objectives of the tests were to obtain fluid front displacement and volume transport data as functions of time.

Fluid was supplied to one end of the test wick by a 100 milliliter supply burette. Excess fluid dripping from the saturated end of the wick was collected in a 100 milliliter receiver burette. Therefore, the total volume transferred along the wick was the difference between readings of the supply and receiver burettes.

The test wick was supported on a 1 centimeter-square mesh screen. Fluid losses by evaporation were minimized by

(i) placing the wick and screen in a 31 mm inside diameter plexiglas tube,

(ii) using a hood to partially cover the area where the fluid was introduced, and

(iii) saturating the atmosphere around the wick by leaving some of the test fluid in the tube before each test.

A copper wire was attached to the support screen at 2-in. intervals along the wick with an additional wire placed at the first location. After removing insulation from the ends of these wires, the ends were inserted into the wick pores. All wires were connected to a 25-point thermocouple switch whose main terminals were connected to an ohmmeter. When the wick was dry, the ohmmeter indicated an infinite resistance between any wire and the additional wire at the first location. Immediately after the wick pores occupied by a wire were wetted, the resistance between that wire and the additional wire changed (over a period of about 1 second) from infinity to about 1 megohm. After one cross section was wetted, the thermocouple switch was advanced to the next wire to detect wetting there. As the wetting front reached each wire, as indicated by the first change in resistance, the total distance of wetting front down the wick and the total elapsed time of the test, measured by a stop watch, were recorded. At the 9-in. location, supply and receiver burette volumes were recorded as well. The curves in Figs. 2-6 were constructed using the displacement-time data. The results will be discussed after the description of Part II of the experimental work.

5. EXPERIMENTAL WORK, PART II

5.1 APPARATUS

Figure 7 shows the test set-up which is the same basic set-up used and described in detail previously [4,5].

Dewar: The three main elements of the heat pipe (heat input, mass transfer, and heat rejection sections) were housed in a 7.62 cm I.D., 8.90 cm O.D., 101.5 cm long double open-ended, silvered glass dewar. The dewar was made of two concentric glass cylinders with the surfaces in the annulus silvered. It provided insulation in the radial direction between the interior and the surroundings. Unsilvered window strips located axially along the top and bottom of the dewar allowed inspection of the interior after assembly. A combination of O-rings and gaskets was used to seal the dewar in the axial direction. Also, a radial tube between the inside of the dewar and the exterior was provided near the condenser end to allow charging of the wicking chamber with the desired fluid, and to provide connection of a vacuum system to the interior and passage of thermocouple wires entering the dewar.

Heat Input Section (Evaporator): Two electrical heating elements were used as the heat input source. The heater closer to the mass transfer section (main heater) was used for the total heat input, while the other (guard heater) was used to minimize axial heat flow away from the chamber. An accurate measurement of heat input to the mass transfer section could then be obtained by recording input wattage to the main heater. Electrical energy was supplied to each heater from a variable transformer. Wattmeters were used to measure the power inputs. An O-ring sealed aluminum disc wall between the main heater and the mass transfer section provided a uniform source of heat to the chamber.

Heat Transfer Section (Wicking Chamber): Wicking material Refrasil No. C100-28 was used because past experience proved it to be superior in water lift rate and horizontal transfer rate.

Four horizontal strips of wicking material were used to eliminate gravitational effects over a given cross section of the heat pipe. The two center strips were 2-1/16 in. wide, while the top and bottom strips were 1-7/16 in. wide. A wicking cage was utilized to suspend the Refrasil, as well as to prevent axial movement of the heat input and heat rejection sections. Also, circular pieces of Refrasil were attached to the aluminum discs at each end forming the heat input and heat rejection sections to evenly distribute fluid and to supply fluid to the horizontal wicking strips.

Heat Rejection Section (Condenser): An O-ring sealed aluminum disc was used in the heat rejection section, as in the heat input section, to provide uniform heat transfer. After being conducted through the aluminum disc, the heat was removed by circulating cool water through the condenser section.

Temperature Measurement: Temperatures throughout the system were measured by monitoring twenty-two copper-constantan thermocouples. One thermocouple was placed across a mica disc between the main and guard heaters. With the desired heat input set on the main heater, the guard heater was continually adjusted to keep the temperature gradient at zero. Thus, it was ensured that the desired heat input would flow into the mass transfer section. Another thermocouple was placed across the inlet and outlet cooling water lines as a check on the heat transferred along the heat pipe. The twenty remaining thermocouples were used to record temperatures at various locations on the heater and condenser aluminum discs and in the wicking chamber.

Wicking Chamber Manifold: A wicking chamber manifold was located near the condenser end to make the necessary connections to the wicking chamber. This manifold contained a large vacuum valve for evacuating the dewar through a glass access tube in the side, as well as a small needle vacuum valve to purge non-condensible gases from the chamber. Another small needle valve allowed the chamber to be charged with the working fluid. Passage of thermocouple wires into the chamber was made possible by a transfer plug. Connection of a pressure-vacuum gage to the chamber was accomplished by a union on the manifold. This union was later used in connecting a mercury manometer to the chamber. Finally, another union was made available for direct connection to the chamber itself.

Temperature Recording: A 21-point Leeds and Northrup millivolt recorder was used to record the twenty temperature measuring points. The temperature gradients between the two heaters and over the cooling water lines were monitored on a potentiometer.

5.2 TEST PROCEDURE

The wicking chamber was first evacuated to 15×10^{-2} mm of mercury. Then, for testing of single layers of Refrasil, 45 ml of working fluid was injected into the chamber. However, for testing double layers of material, difficulty was experienced in initially filling the gaps with fluid. This made it necessary to fill the entire chamber with liquid. Then, the greater part of the fluid was removed using a makeshift sump pump consisting of a gallon jug and the vacuum pump.

As heat was added through the heaters, non-condensible gases immediately began to build up in front of the condenser aluminum disc. This caused a temperature gradient to build over the length of the heat pipe. When the gradient reached 5-10°C, the needle vacuum valve was opened slightly to purge these gases from the system. At heat inputs below the burn-out wattage, the temperature gradient along the heat pipe reduced to about 1°C or a little larger for large heat inputs.

After purging several times (over a period averaging thirty minutes to an hour), the heat input was increased if burn-out had not occurred. Naturally, test runs were shorter for testing of double layers of Refrasil since the fluid flowed more rapidly. At or above the burn-out wattage, purging did not reduce the temperature of the heater aluminum disc to previous levels. Instead, the temperatures indicated by the thermocouples on the disc continued to increase.

When this happened, the heaters were shut off so the wicking material could resaturate with fluid.

Condensible gases purged from the system were condensed by a cold trap immersed in dry ice and methanol before they could reach the vacuum pump. The cold trap was changed every 2-3 hours as very little fluid accumulated there. The temperature gradient between heaters was monitored constantly during testing and the guard heater was adjusted accordingly to keep it at zero. Also, before increasing heat input, the temperature difference between the cooling water inlet and outlet lines was recorded, as was the cooling water flow rate. Unsuccessful attempts were made during purging of non-condensible gases to measure the volume and alcoholic content of fluid accumulated in the cold trap. The volume accumulated during a normal run was much too small for accurate measurement and analysis.

6. DISCUSSION AND RESULTS

In Part I, the fluid transport rate along the wick was found using the equation,

$$\dot{v} = A \frac{dx}{dt} \qquad (6)$$

where $\dot{v}$ is the volumetric flow rate, A is the cross-sectional area of flow and dx/dt is the velocity of the wetting front.

The velocity of the wetting front was found by differentiating the displacement-time equations plotted in Figs. 2-6. The equations of the curves were found using first-order, least squares approxi- to the log-log plots.

Since the cross-sectional area of flow was not known, it was necessary to make an approximation using volumetric flow data. The total volume input over the test length divided by the total elapsed time gave the volumetric flow rate for a wick length about one-half that of the test length (see development in sample calculations). Since tests with water at zero degrees inclination produced the most consistent volumetric flow data, those data were used in finding the cross-sectional area of flow. This area was used as an approximate area for other tests. All testing in Part I was done on 9-in. X 1-in. test strips.

The results (see Tables I and II and Figs. 8-14) indicate water has the highest flow rate, while pure ethyl alcohol and 50 percent ethyl

TABLE I

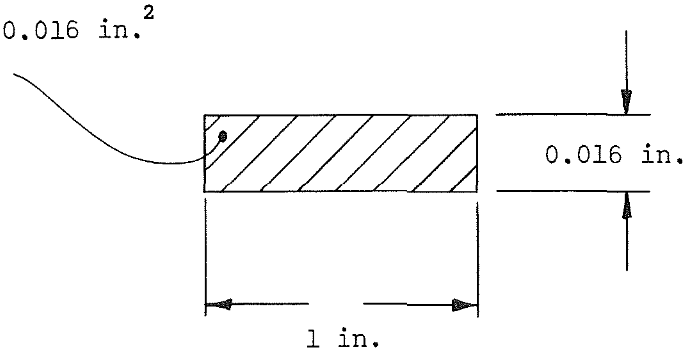
SINGLE LAYER OF REFRASIL NO. C100-28

VOLUMETRIC FLOW RATE AT 21-3/4 IN. (ML/MIN)
PER INCH WIDTH

	H_2O	E. Alc.	50% E. Alc.	
0°	0.073	0.017	0.006	Wick #2 Figs. 2 and 8
0°	0.030†	0.011		Wick #8 Figs. 5, 11, and 12
5°	0.032	0.004		
10°	0.027	0.001†		
15°	0.023	0.000††		
90°	0.009	0.002††		

Refrasil

$A_{solid} = 0.016 \text{ in.}^2$



Area of flow = 0.011 in.^2

= 68 percent of A_{solid}

---

†Figure 5 indicates a bad point; therefore, 0.074 ml/min may be more accurate.

††These numbers are only approximate due to the extremely small flow rates and relatively high evaporation rates of the fluid.

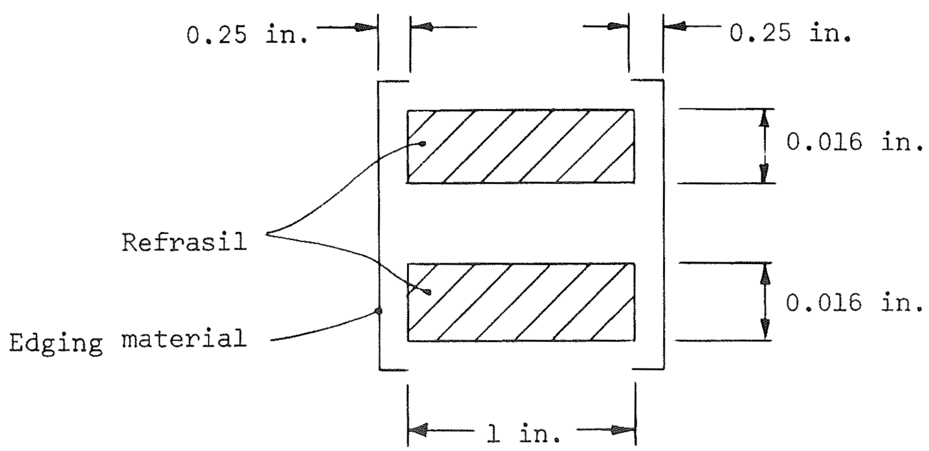
TABLE II

DOUBLE LAYER OF REFRASIL NO. C100-28

VOLUMETRIC FLOW RATE AT 21-3/4 IN. (ML/MIN)

PER INCH WIDTH

	H_2O	E. Alc.	50% E. Alc.	
0°	0.865	0.346	0.617†	Wick #5 Figs. 3 and 9
0°	0.809	0.213	0.191	Wick #6 Figs. 4 and 10
0°	0.735	0.400		Wick #9 Figs. 6, 13, and 14
5°	0.471	0.182		
10°	0.248	0.691		
15°	0.122	0.034		
90°	0.016	0.006		



Area of flow = 0.040 in.^{2}

= 125 percent of $2A_{solid}$

†Appears to be inaccurate, comparing Figs. 9 and 10.

alcohol by weight were much lower. The pure alcohol appeared to be only slightly better than the 50 percent mixture. These results are in good qualitative agreement with the analytical studies of [6]. Comparison of Figs. 9 and 10 indicates the 50 percent alcohol run at zero degrees in Fig. 9 is inaccurate. This run is also shown in Fig. 3.

With a single layer of wicking material, the calculated water flow rate for a 21-3/4-in. X 1-in. wick was from 0.073 ml/min at zero degrees to 0.009 ml/min at ninety degrees. The flow rate for pure alcohol was from 0.011 ml/min at zero degrees to 0.002 ml/min at ninety degrees.† The 50 percent mixture flowed at 0.006 ml/min at zero degrees.

Two layers of wicking material sewn together increased the flow rate of water by a factor of about ten at zero degrees. This factor diminished to about two at ninety degrees, indicating that water did not flow in the vertically oriented gap. The improvement factor for alcohol was extremely erratic, although much larger than for water. The inconsistency suggests either inaccuracies in the extrapolation of distance-time curves or inconsistent filling of the gap. At zero degrees, the factor was about 25 and, at ninety degrees, the factor was 3. The 50 percent mixture increased by a factor of about 30 at zero degrees. The increase in flow rates due to a gap next to the wick has been suggested by previous workers [7,8].

---

†See second footnote in Table I.

Results of Part I were used to predict the heat transfer capacity of a 21-3/4-in. heat pipe operating with similar wicking, fluid, and inclination. Test results for a single wick gave predicted wattages of 21.0 at zero degrees and 2.6 at ninety degrees for water and 0.90 at zero degrees and 0.17 at ninety degrees for pure ethyl alcohol. A predicted range of 0.5-1.7 watts for the 50 percent mixture at zero degrees was found using latent heats of alcohol and water, respectively.

Results tabulated in Table III show that the burn-out wattages obtained with water in Part II were lower than those predicted from tests outside the heat pipe. Apparently, there is difficulty in saturating the wicking material in the heat pipe and in keeping the gaps between the wick completely filled. A bottleneck in the mass return process occurs at the condenser end where the fluid must travel vertically in the circular Refrasil patches. The differences are especially large for double layers of wicking material since rapidly moving fluid must be supplied to the gaps. The situation was alleviated somewhat by placing several circular Refrasil patches on the condenser aluminum plate. Ends of the horizontal strips were directed vertically downward between these patches to create a better gap effect. Also, thin spacers were placed between the aluminum disc and the wicking cage. Finally, the wicking cavity was temporarily filled with fluid prior to testing. After removing most of the fluid, the gaps could then resupply themselves by capillary action.

The equation for maximum heat transport in a heat pipe was given by Eq. (5).

TABLE III

PREDICTED (PART I) AND ACTUAL (PART II) BURN-OUT WATTAGES FOR A WICK LENGTH OF 21-3/4 IN. AND FOR A TOTAL WIDTH OF 7 INCHES

		Predicted Wattage (Part I)		Actual Wattage (Part II)	
Fluid	Inclination to Horizontal†	Single Wick	Double Wick	Single Wick	Double Wick
Water	0°	21.0	211, 232, 248	15^{+1}_{-2}	≥ 80††
Water	1°			12^{+1}_{-2}	
Water	2°			11 ±1	
Water	3°			9 ±1	
Water	4°			8 ±1	
Water	5°	9.1	135	7 ±1	45 ±5
Water	8°			5 ±1	
Water	10°	7.7	71	5 ±1	45 ±5
Water	15°	6.5	35	5 ±1	25 ±5
Water	90°	2.6	4.5		
Ethyl Alc.	0°	0.90, 1.4	17.9, 28.8, 33.5	4 ±1	
Ethyl Alc.	5°	0.33	15.2		
Ethyl Alc.	10°	0.08	5.8		
Ethyl Alc.	15°	0.00†††	2.8		
Ethyl Alc.	90°	0.17	0.5		
50% E. Alc.	0°	0.5-1.7	16-54.7	2 ±1	
25% E. Alc.	0°			1 ±1	
75% E. Alc.	0°			2 ±1	

†Flow against gravity if not 0°.

††This was the maximum power input of the heater.

†††See second footnote in Table I.

For the horizontal position, the maximum heat transfer capacity Q_{max} becomes inversely proportional to the wick length L. Tests by the author and by others [4,5] support this relationship with burn-out wattages of 15 (the author) and 10 for wick lengths of 21-3/4 in. and 32-1/4 in., respectively, for the same total width of 7 in.

Tests with water at various angles of inclination produced burn-outs from 15 watts at zero degrees to 5 watts at eight, ten, and fifteen degrees. The reduction in capacity change rate with an increasing operating angle agrees with the behavior of the cosine function [see Eq. (5)].

At zero degrees inclination, the burn-out wattage for pure ethyl alcohol was 4 watts. This represents a decrease by a factor of 3.75 from water. Equation (5), however, predicts a factor of about 13.5. However, the burn-out wattage of 4 is higher than predicted by tests outside the heat pipe. This is probably due to inaccuracy of heat input measurement for such low inputs. Second, the burn-out wattage of 15 for water is low because the wicking material in the heat pipe could not be thoroughly saturated with fluid. Part I tests indicated factors of 15.0-23.1 for a single wick and 6.30-13.9 for a double wick. Both ranges have the right order of magnitude when compared to the predicted value of 13.5.†

---

†It is notable that wick performance at times varied considerably for different test sections of Refrasil (see Tables I and II). This is probably due to variation in pore size.

Fifty percent alcohol had a burn-out wattage of only 2 watts at zero degrees. This was expected. A burn-out of 1 watt with 25 percent alcohol agrees with Woo [6], but disagrees with Feldman and Whitlow [9]. However, in a wick limited heat pipe, the effect of vapor flow is negligible, whereas in Ref. [9], the improvement may have been due to compression of the vapor. Seventy-five percent alcohol resulted in 2 watts burn-out.

Figure 15 shows temperature variation along the heat pipe for 25, 50, and 75 percent alcohol by weight mixtures. Due to the very low wattages, the 25 and 50 percent curves do not represent steady state conditions, so these curves may not be very useful for comparison of temperature differences. The curves, however, do seem to indicate correctly the location of a transition from alcohol at the condenser end to water at the heater end in agreement with Feldman and Whitlow [9].

Usable data for cooling water temperature change in Part II tests were not obtained. Such data could have been used with the cooling water flow rate as a check on heat input measurement. Heat conduction through the water piping and other heat losses prevented accurate thermocouple recordings. Also, purging of non-condensible gases from the system caused a refrigeration effect at the condenser end due to the small amounts of working fluid extracted. The magnitude of this effect was estimated to be about 1.5 watts from the material collected in the cold trap. Cooling water was actually cooled at times, instead of heated. Occasionally, formation of ice near the manifold in the vacuum hose as well as cooling of the manifold were noted.

7. CONCLUSIONS AND RECOMMENDATIONS

The simple experimental set-up shown in Fig. 1 can be used to obtain quantitative design data on wick materials with relatively volatile fluids, too. Appropriate precautions, however, must be made to minimize evaporative losses.

A liquid-filled gap adjacent to and running parallel to a wick increases the liquid transfer rate severalfold. Consequently, it can be recommended that a wick should always be designed with an adjacent gap provided by either another layer of wicking or a suitable wall. The gap should be as narrow as possible since refilling a wide gap inside a sealed heat pipe could be practically impossible. For the same reason, during the initial filling of the heat pipe, steps should be taken to ensure complete filling of all such gaps with the working fluid. The results of [7] indicate that gaps of any appreciable size, i.e., if the wick is not touching the wall or the next layer forming the gap, tend to lose their liquid fill, particularly in an adverse gravity gradient.

In a wick limited heat pipe, the addition of water to alcohol does not improve the performance, but tends to affect it adversely. A temperature difference, however, can be established across the heat pipe by the use of such mixtures. The property parameter suggested by [6], $\rho h_{fg}\sigma/\mu$, seems to give good qualitative predictions for the performance of mixtures in a wick limited heat pipe. If the heat pipe is vapor-limited, however, the effect of higher vapor densities occuring with the mixtures could improve the heat transfer rates as suggested by [9].

The following recommendations can be made for future work on the heat pipes :

(1) Circular Refrasil patches should be placed between the aluminum disc and the wicking cage at each end of the heat pipe. Longer wicking strips should then be employed so that the ends can be run vertically between the patches. This will increase fluid flow at the ends of the wicks, particularly at the condenser end.

(2) Larger heater wires will allow finding the maximum heat transfer capability of double wick layers.

(3) Tests should be run with the condenser end of the heat pipe kept at a temperature below 32°F. To do so, it would be necessary to use a fluid, such as Freon, circulating in a closed system through the cooling chamber at the condenser end. A refrigeration unit would be needed to cool the circulating Freon.

(4) Investigate chemical methods of determining alcoholic content of purged gases. This will aid in determining how the two fluids in a mixture separate in the heat pipe.

REFERENCES

1. ASME, Aviation and Space Conference, Los Angeles, California, June 1970.

2. AIAA, Thermophysics Conference, Berkeley, California, June 1969.

3. Fourth Intersociety Energy Conv. Eng. Conf., Washington, D.C., September 1969.

4. Streckert, J. H., and Chato, J. C., "Development of a Versatile System for Detailed Studies on the Performance of Heat Pipes," Technical Report No. ME-TR-64, Department of Mechanical and Industrial Engineering, University of Illinois, Urbana, Illinois, December 1968.

5. Chato, J. C., and Streckert, J. H., "Performance of a Wick-Limited Heat Pipe," ASME Paper No. 69-HT-20, 1969.

6. Woo, W., "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits," TRW System Group, Ames Research Center, Report No. 06462-6007-RO-00, November 1968, pp. 6-17.

7. Shaffer, J., "The Effects of the Proximity of a Wall on the Performance of a Wick," unpublished report, Department of Mechanical and Industrial Engineering, University of Illinois at Urbana-Champaign, 1969.

8. Kemme, J. E., Verbal Report, ASME, Aviation and Space Conference, Beverly Hills, California, June 1968.

9. Feldman, K. T., Jr., and Whitlow, G. L., "Experiments with a Two-Fluid Heat Pipe," Proc. 4th Intersociety Energy Conv. Eng. Conf., Washington, D.C., 1969, pp. 1025-1032.

LIST OF SYMBOLS

A	cross-sectional area of wick (ℓ^2)†
g	acceleration of gravity (ℓ/t^2)
h_{fg}	latent heat of vaporization of liquid (q/M)
K	wick permeability (ℓ^2)
L	actual length of wicking material (ℓ)
$\dot{m}$	mass flow rate (M/t)
ΔP_c	capillary pumping head (F/ℓ^2)
ΔP_g	gravitational head (F/ℓ^2)
ΔP_L	liquid viscous drag (F/ℓ^2)
ΔP_v	vapor pressure drop (F/ℓ^2)
Q	heat transfer rate (q/t)
r	mean pore radius of wicking material (ℓ)
V	velocity (ℓ/t)
$\dot{v}$	volume flow rate (ℓ^3/t)
μ	absolute viscosity of liquid (F-t/ℓ^2) or (M/t-ℓ)
ρ	liquid density (M/ℓ^3)
σ	liquid surface tension (F/ℓ)
ϕ	angle between heat pipe axis and gravitational field

†Dimensions in parentheses are: F--force, M--mass, ℓ--length, q--heat (F-ℓ), t--time.

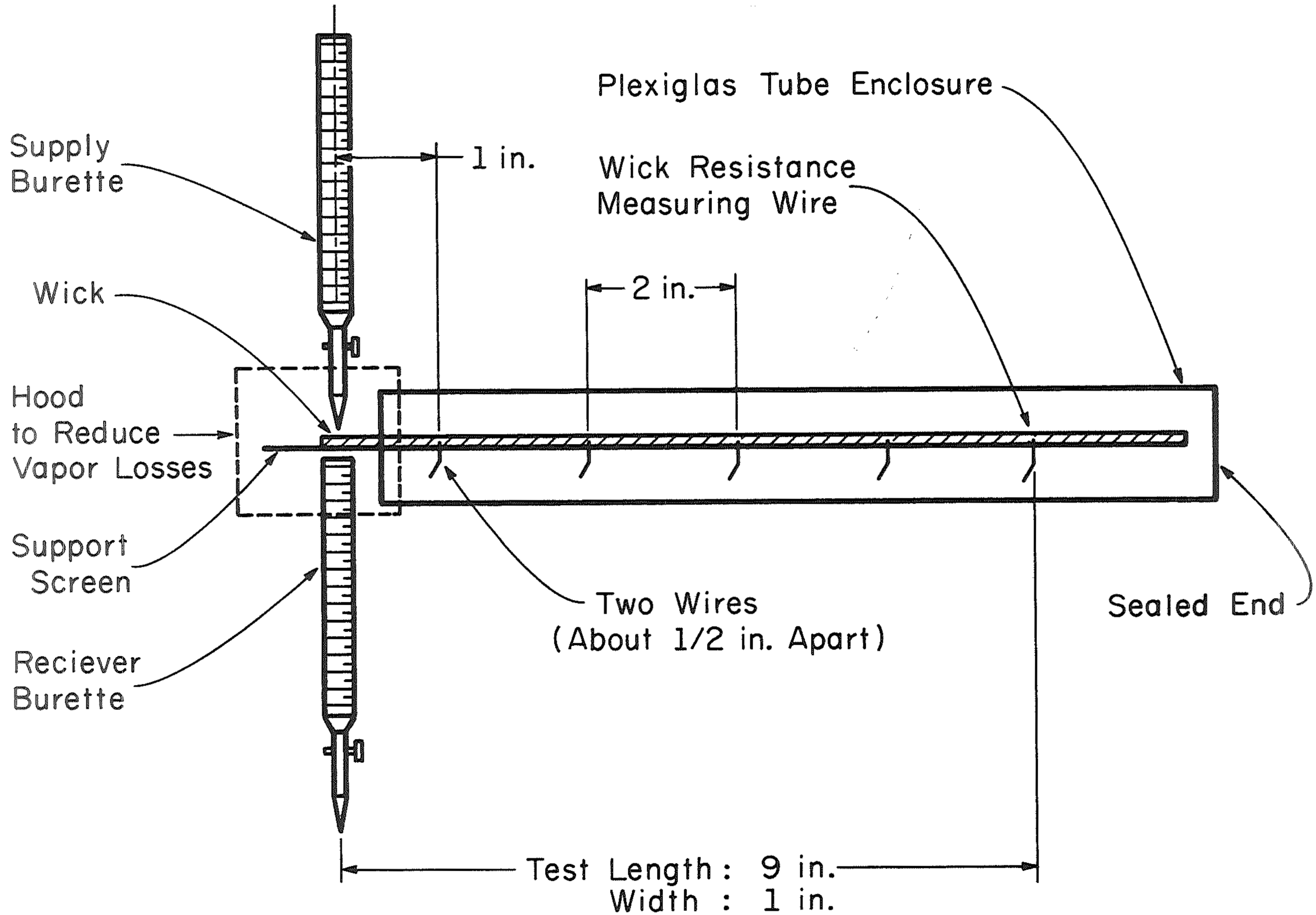


Figure 1. Schematic of test arrangement for measurement of capillary flow in wicks in the gravitational field

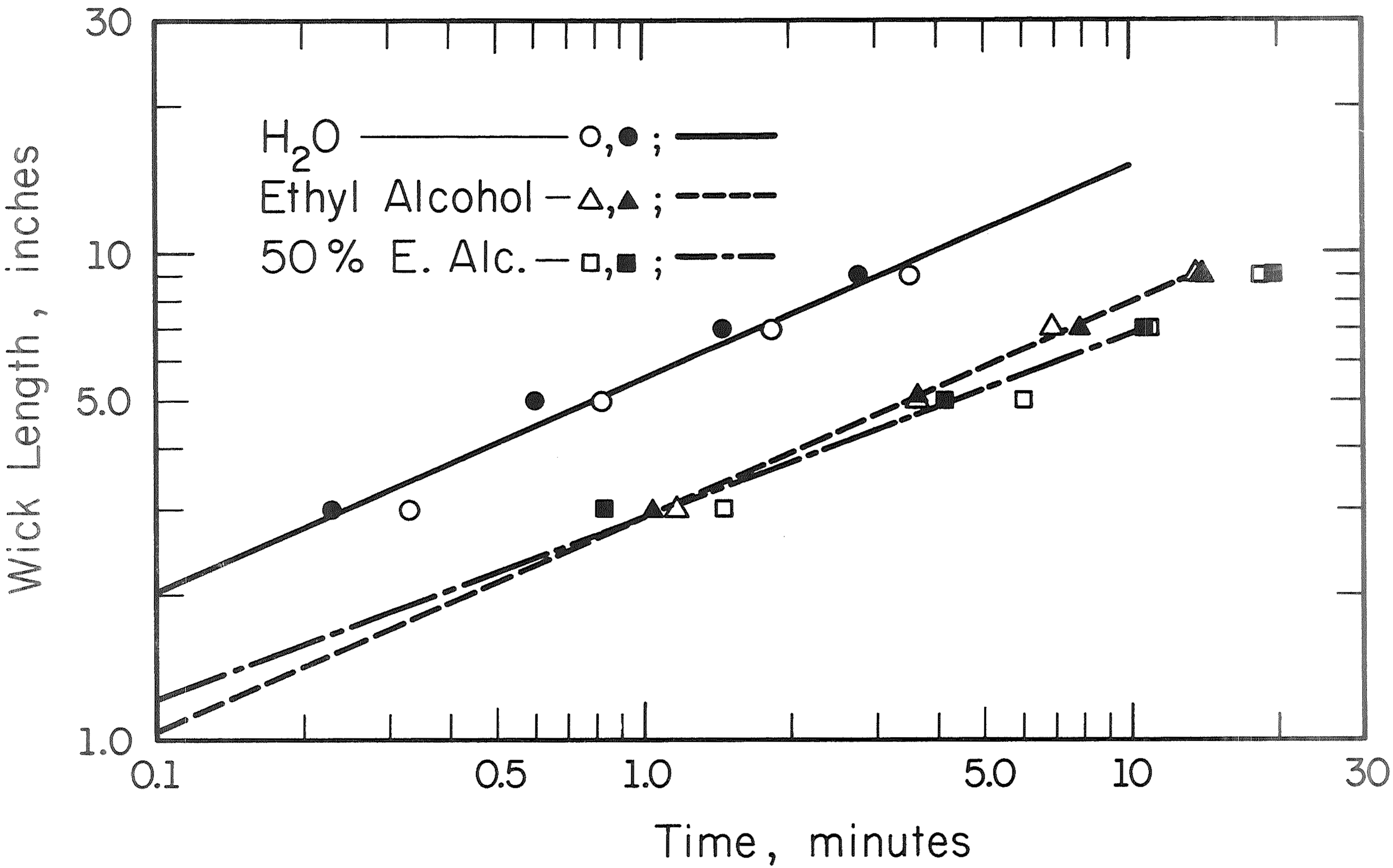


Figure 2. Wick distance vs time graph
Wick No. 2. Single layer at 0°

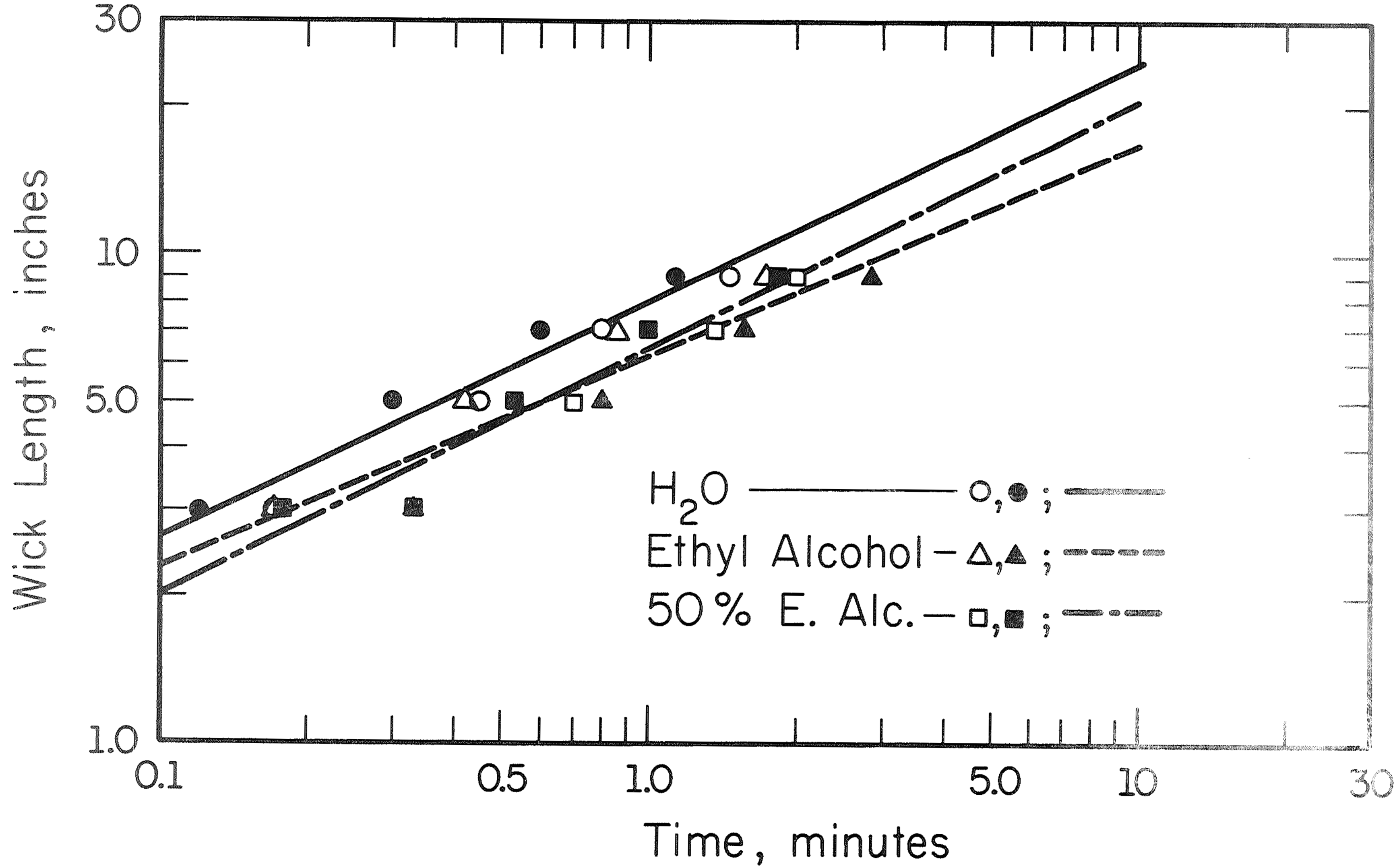


Figure 3. Wick distance vs time graph
Wick No. 5. Double layer at 0°

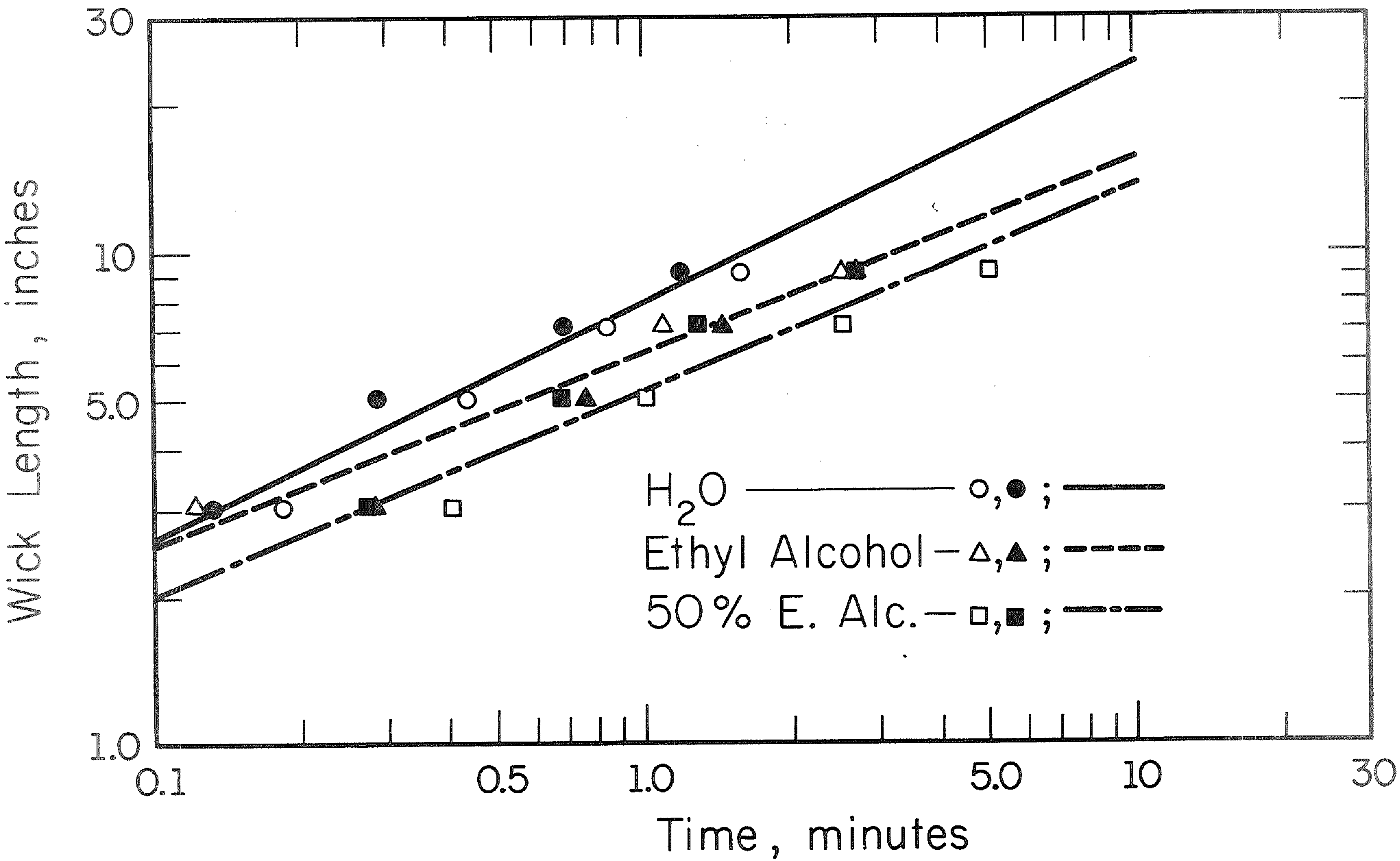


Figure 4. Wick distance vs time graph
Wick No. 6. Double layer at 0°

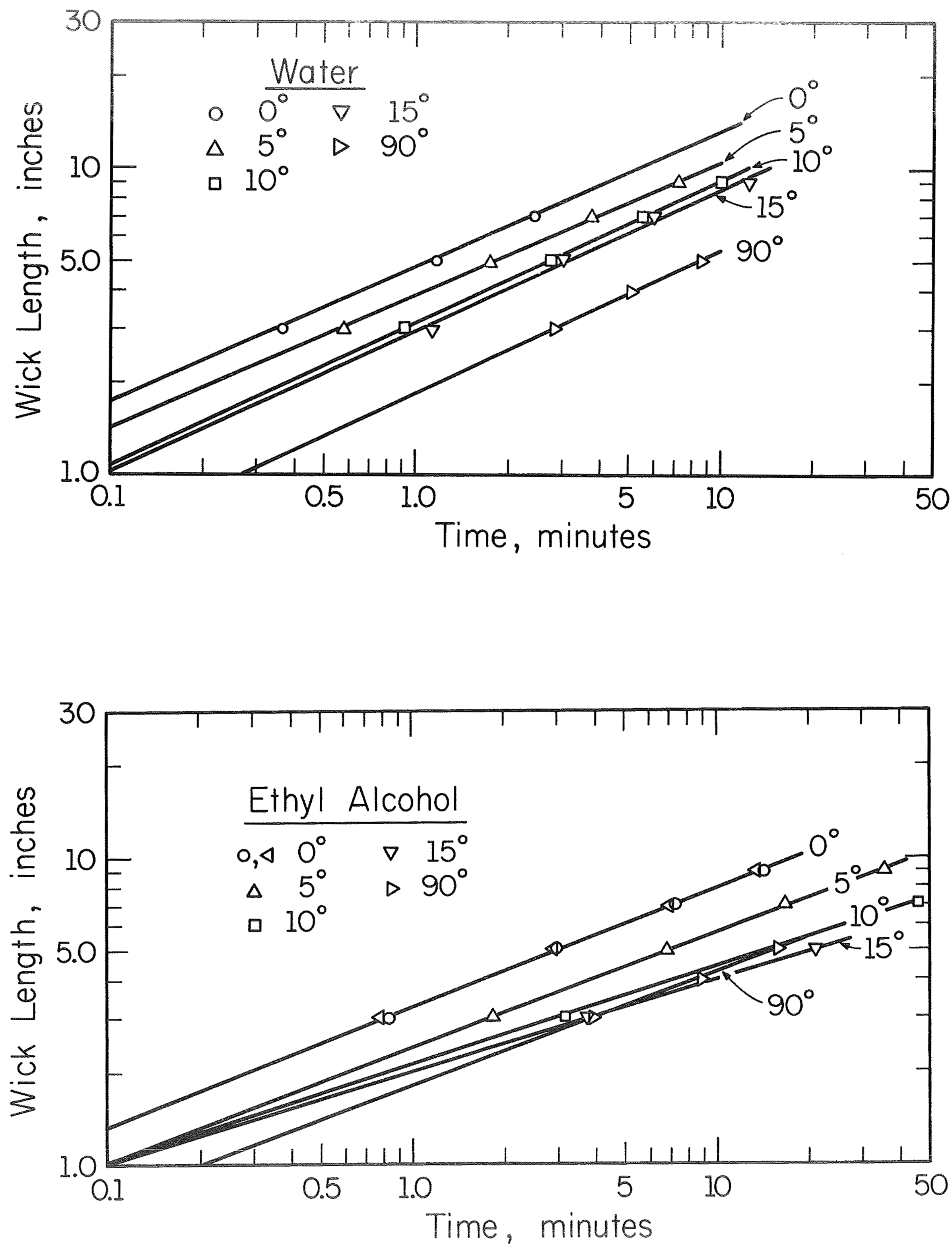


Figure 5. Wick distance vs time graph
Wick No. 8. Single layer

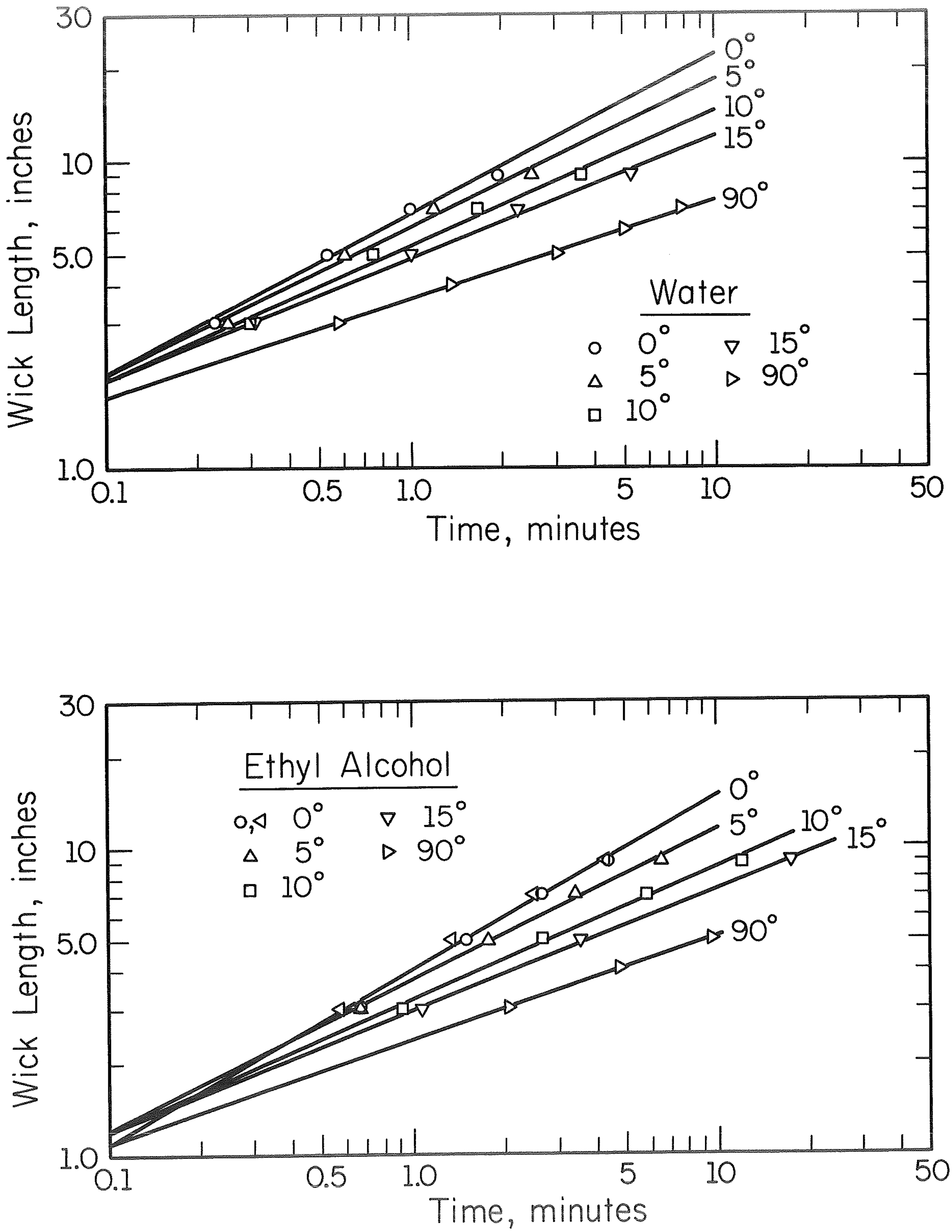


Figure 6. Wick distance vs time graph
Wick No. 9. Double layer

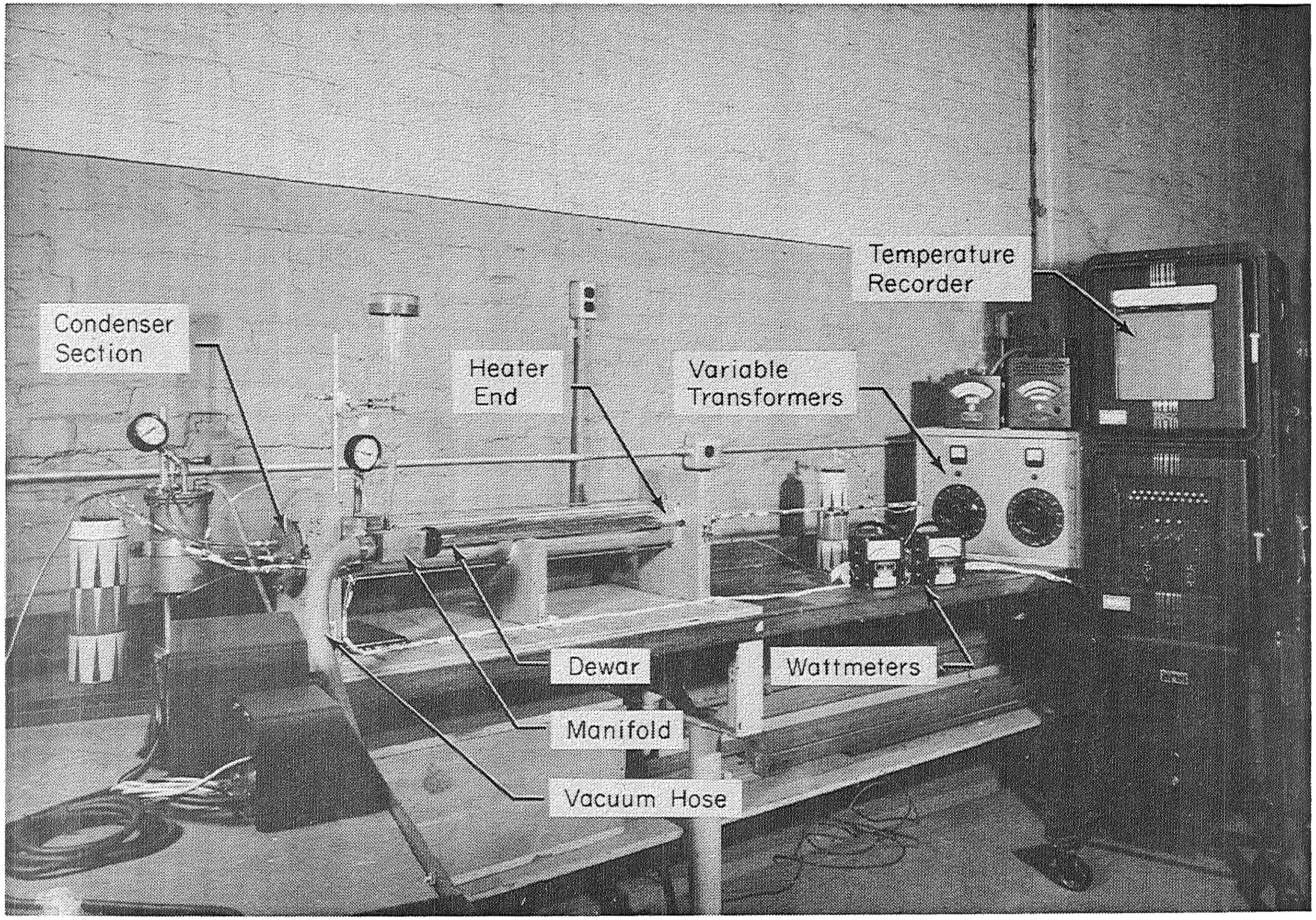


Figure 7. Heat pipe system

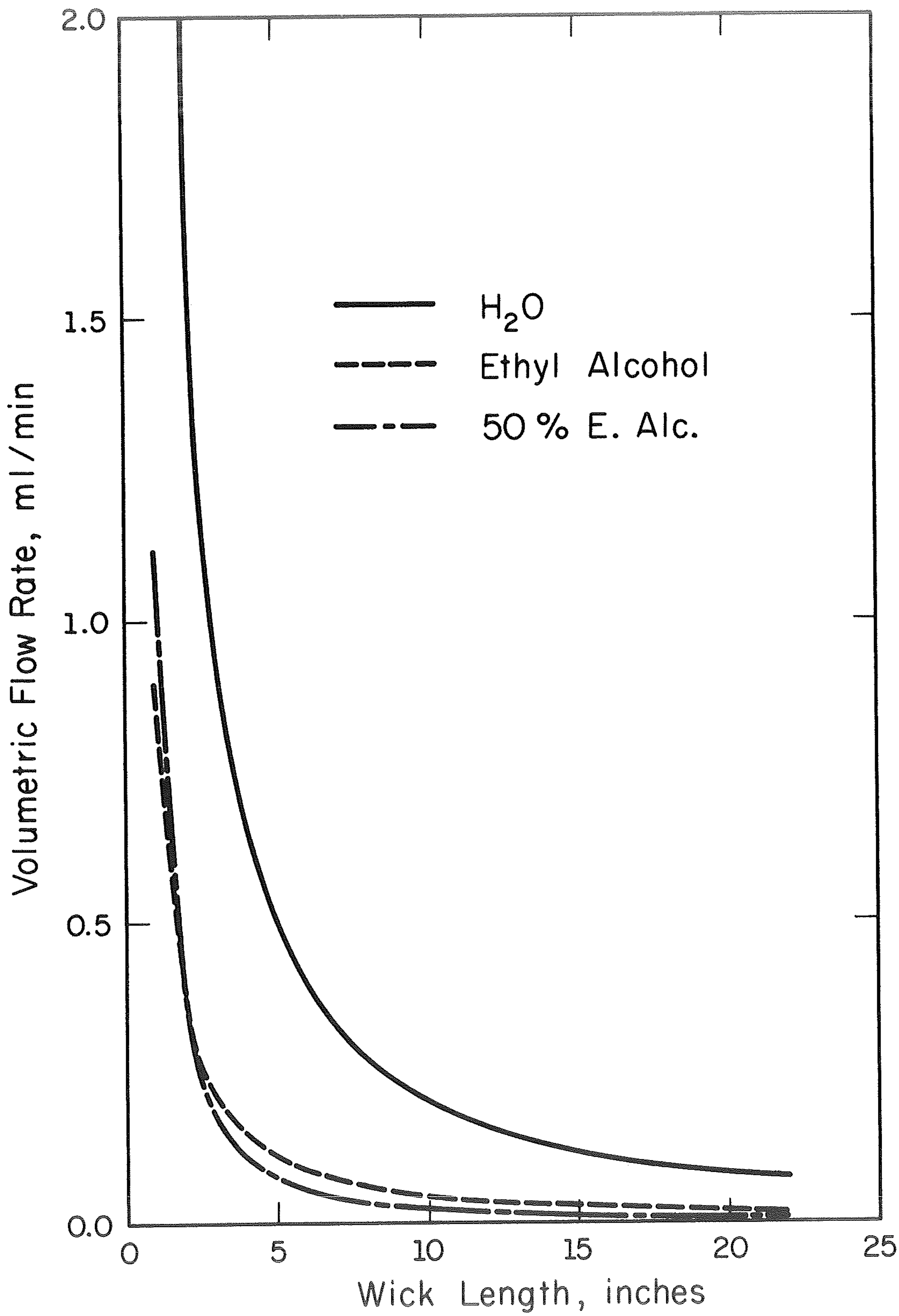


Figure 8. Volumetric flow vs wick length graph
Wick No. 2. Single layer at 0°

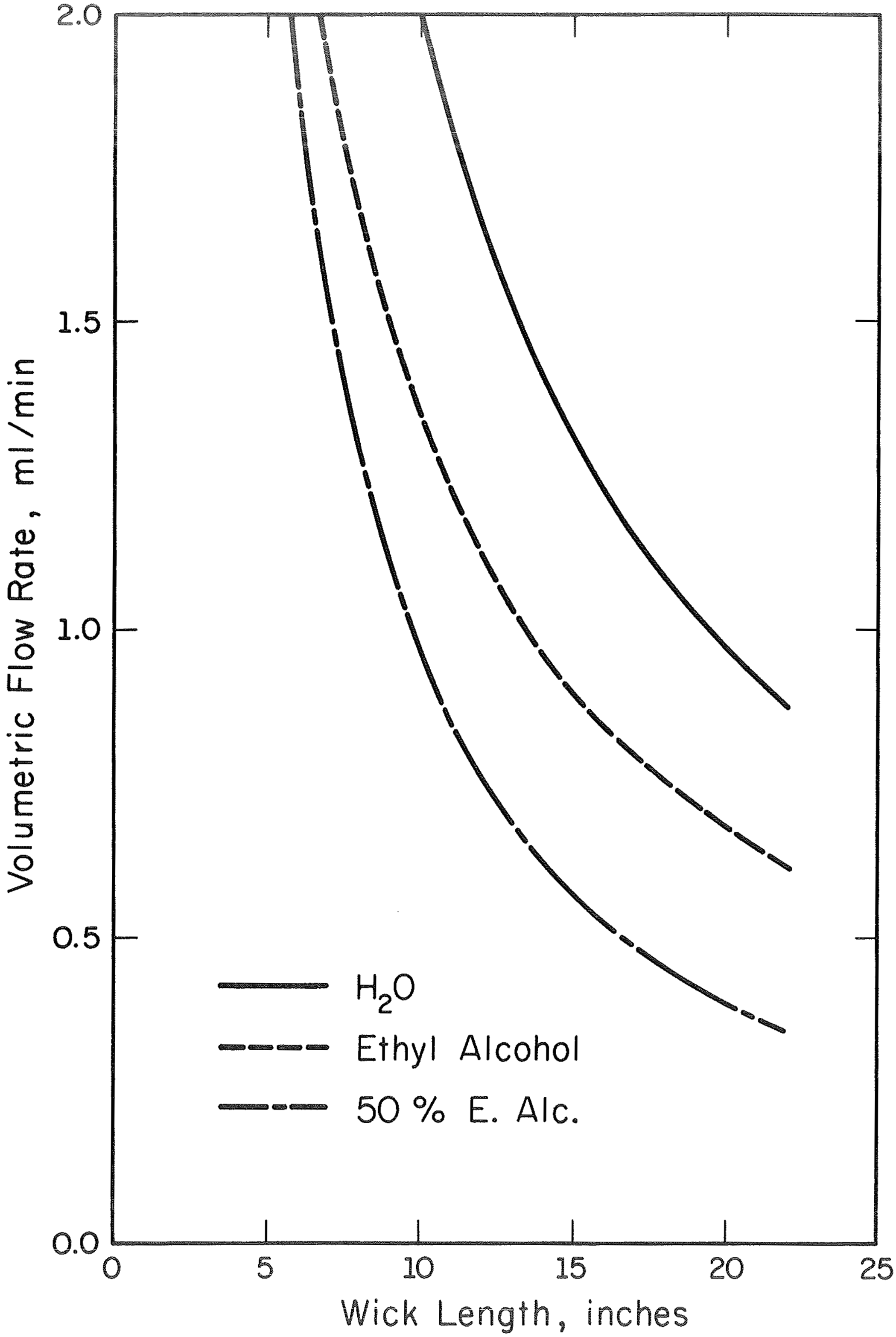


Figure 9. Volumetric flow vs wick length graph
Wick No. 5. Double layer at 0°

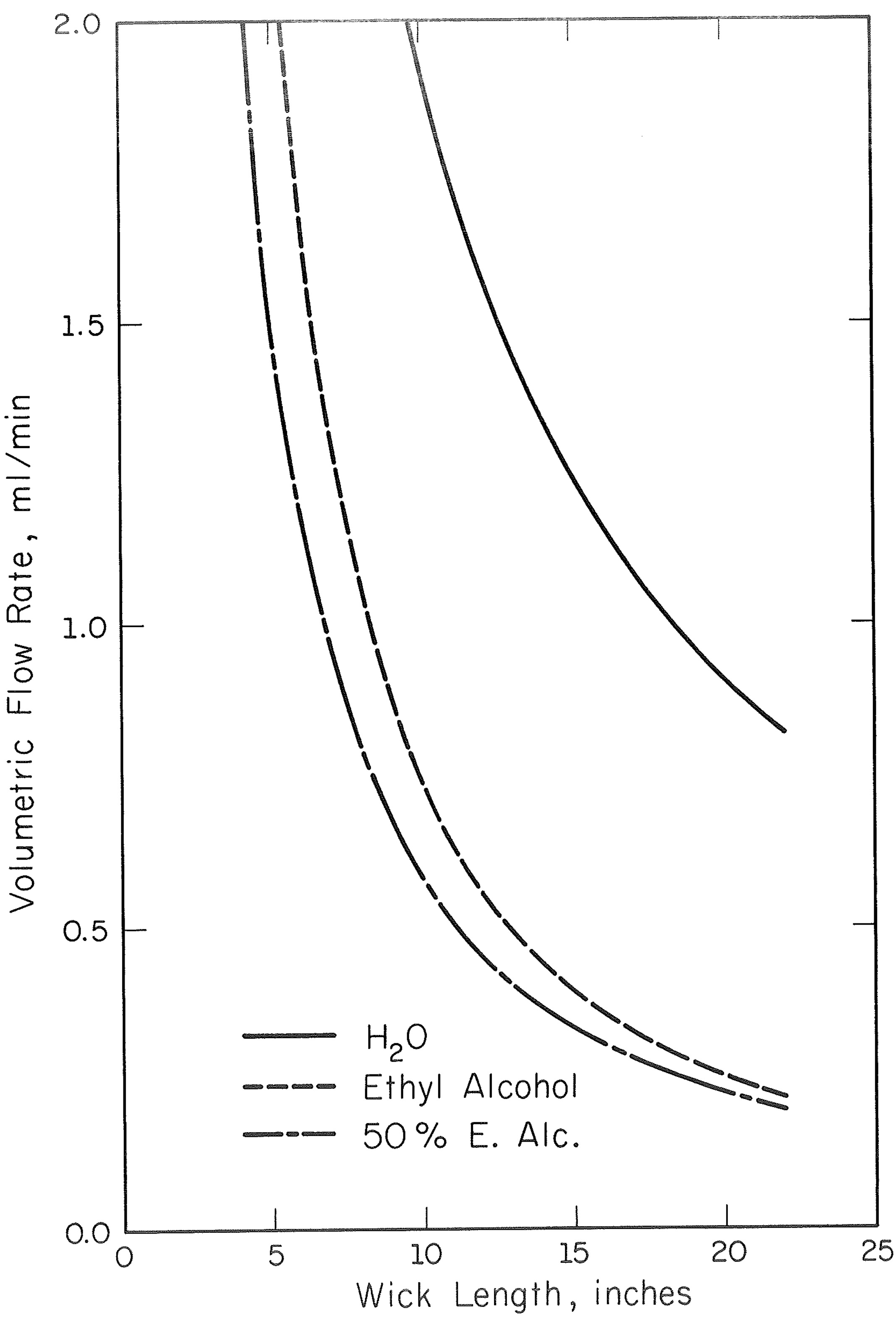


Figure 10. Volumetric flow vs wick length graph
Wick No. 6. Double layer at 0°

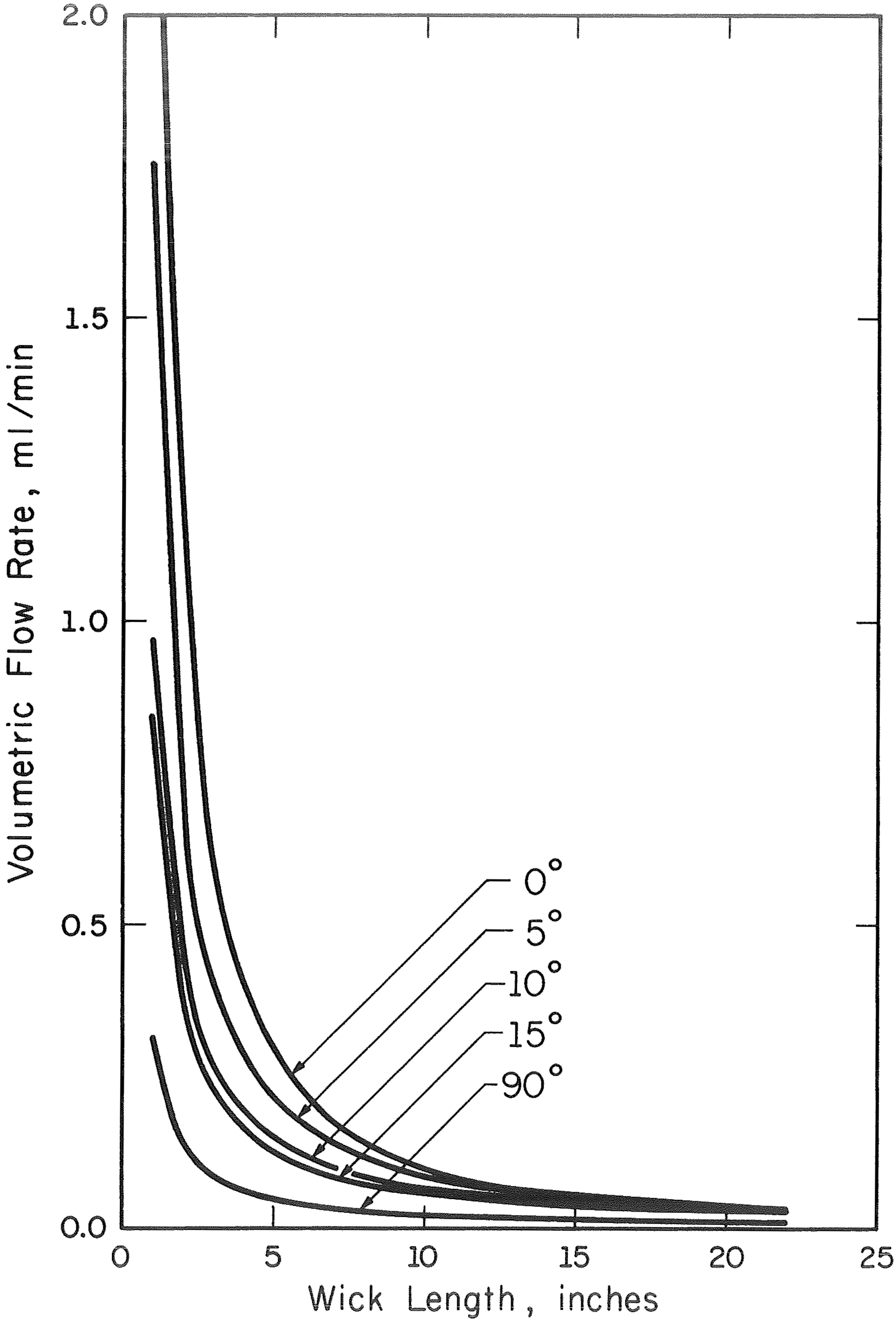


Figure 11. Volumetric flow vs wick length graph
Wick No. 8. Single layer with water

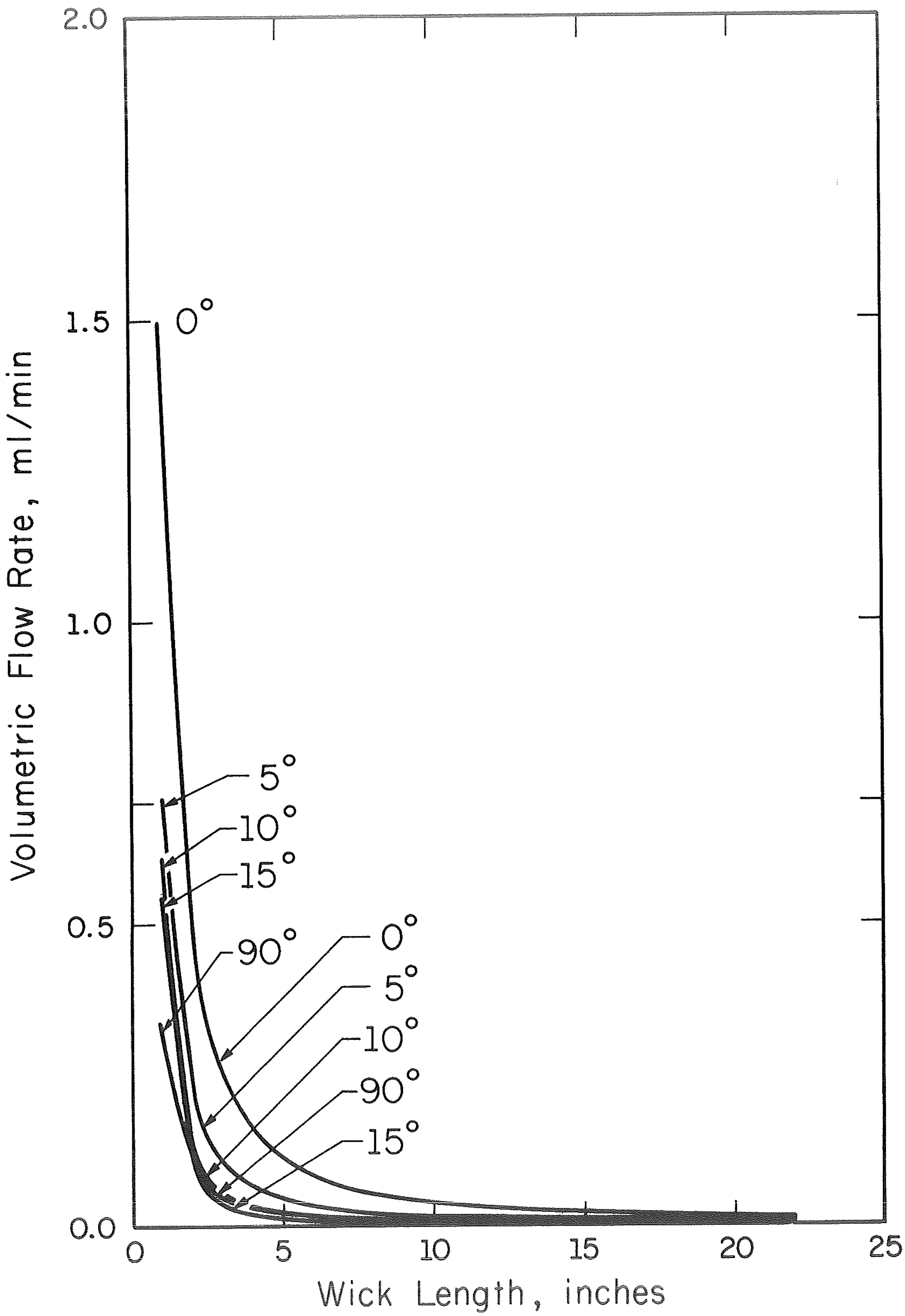


Figure 12. Volumetric flow vs wick length graph
Wick No. 8. Single layer with ethyl alcohol

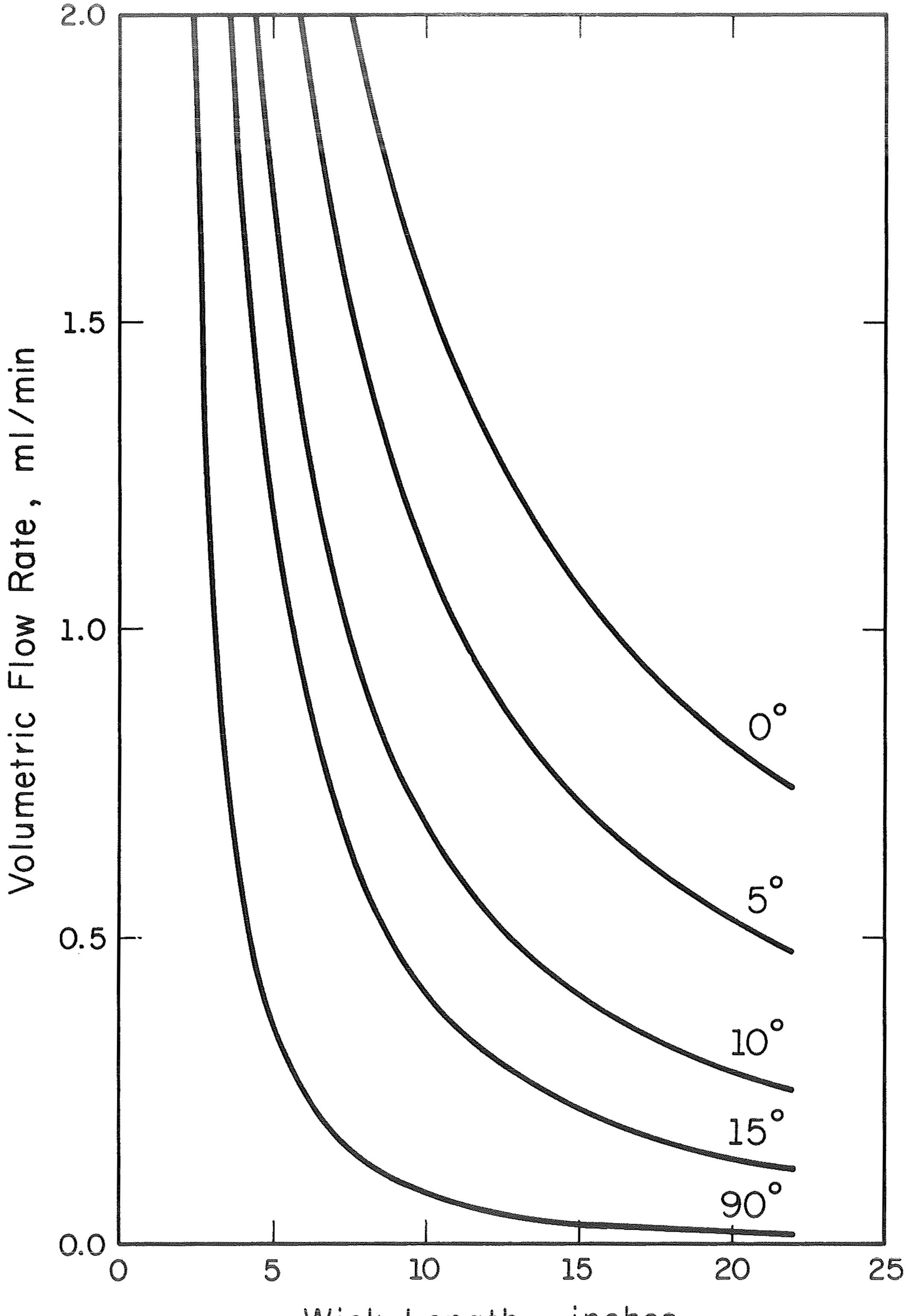


Figure 13. Volumetric flow vs wick length graph
Wick No. 9. Double layer with water

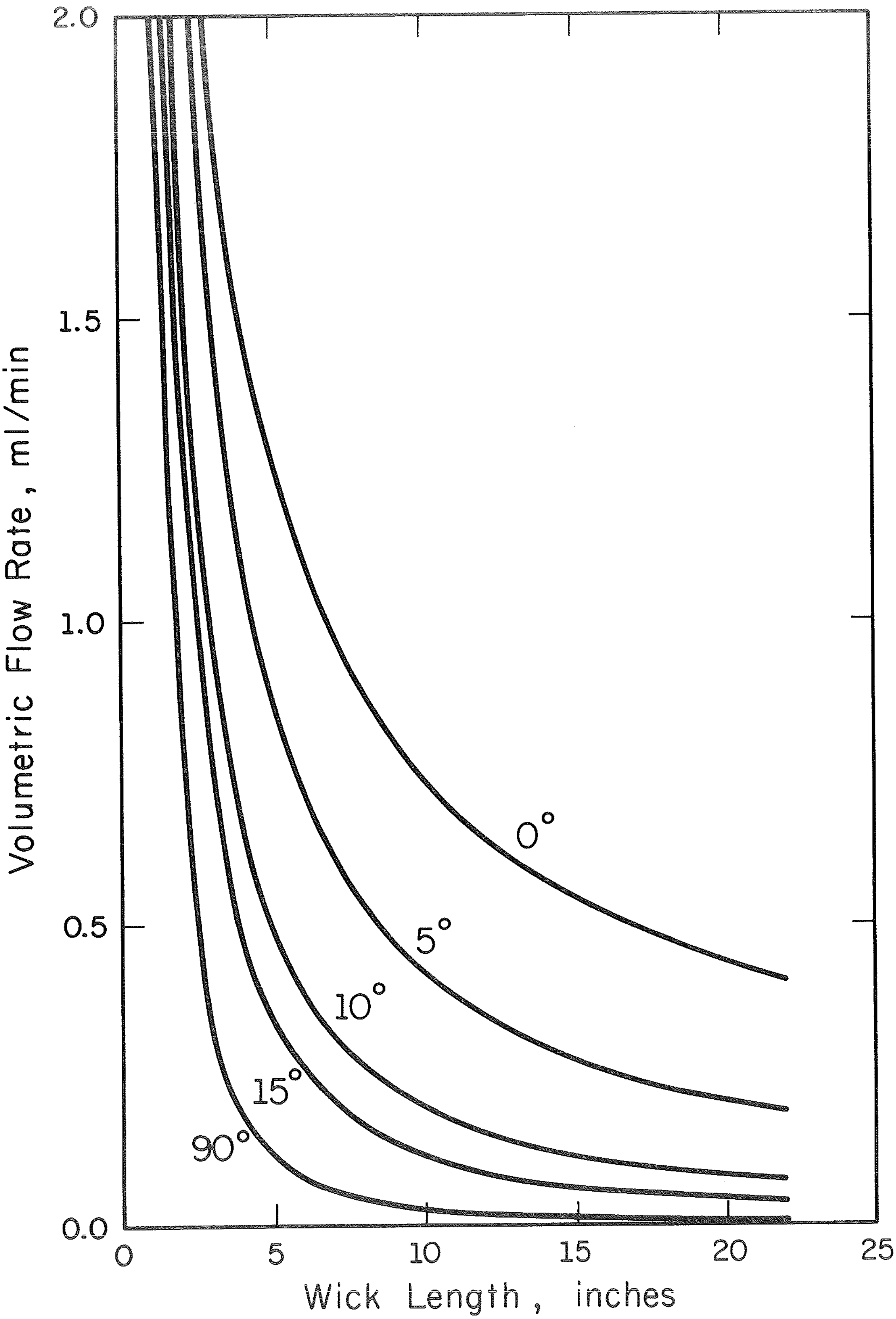


Figure 14. Volumetric flow vs wick length graph
Wick No. 9. Double layer with ethyl alcohol

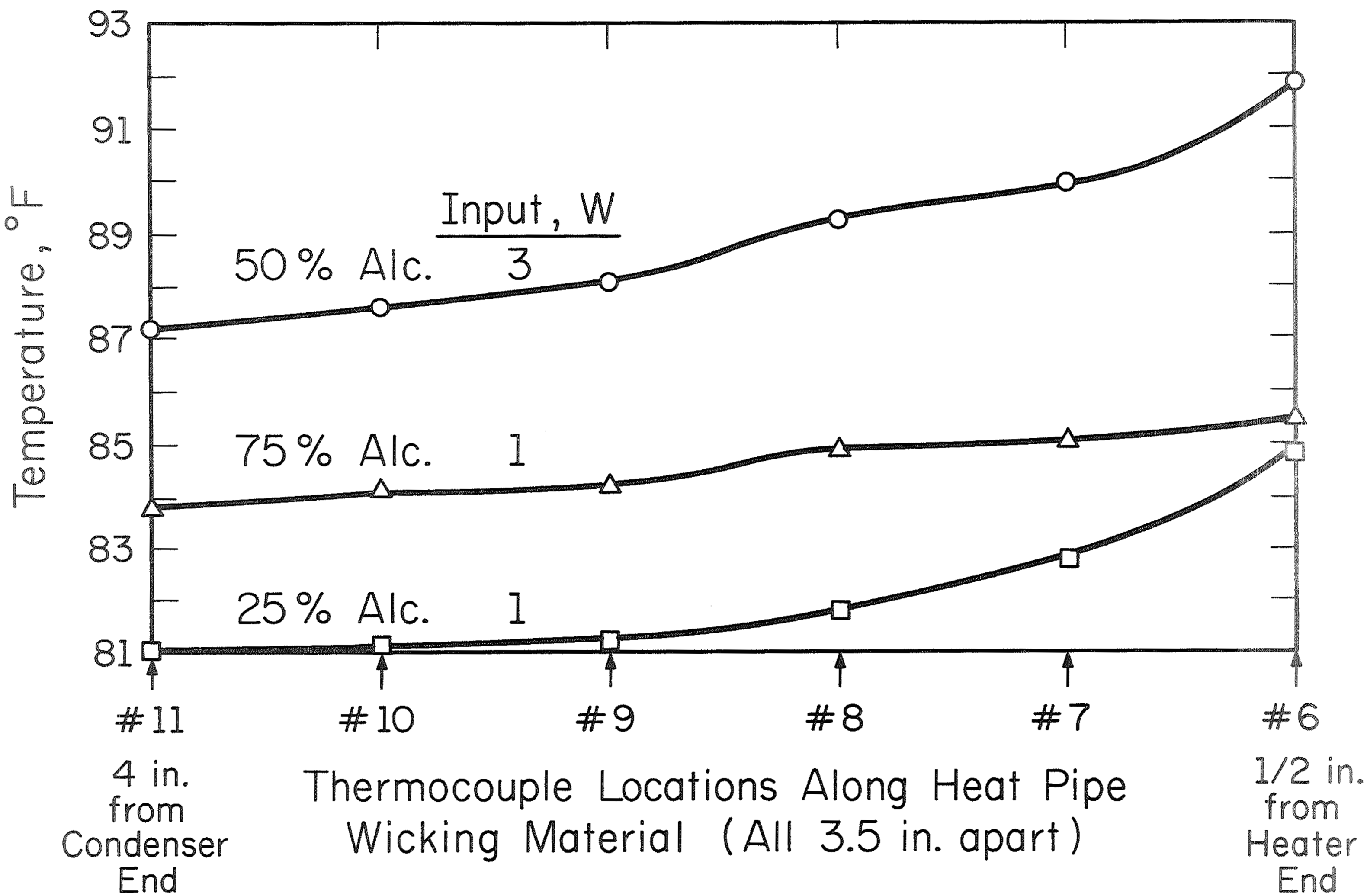
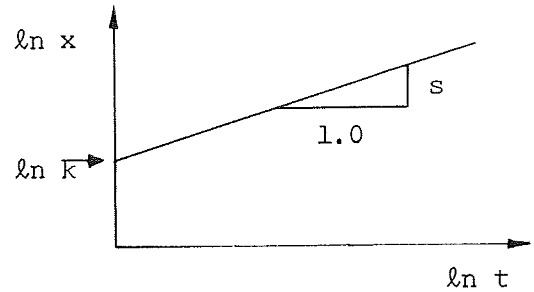


Figure 15. Heat pipe temperature vs distance curves
Single wicks at 0°

APPENDIX

SAMPLE CALCULATIONS

I. Finding Area of Flow



Schematic layout of Figs. 2-6

$$\ln x = s \ln t + \ln k$$

$$x = kt^{s}$$

$$\frac{dx}{dt} = kst^{s-1}$$

$$v_o = AL$$

$v_o \equiv$ total volume in a wick of length L (cm^3 = ml)

$$A = \frac{v_o}{t_o} \frac{1}{L^{(s-1)/s} k^{1/s}}$$

$t_o \equiv$ time at which flow reaches L (min)

$$\dot{v} = A \frac{dx}{dt}$$

$\dot{v} \equiv$ volumetric flow rate at "x" (cm/min) (ml/min)

$$= \frac{v_o}{t_o} \frac{sx^{(s-1)/s}}{L^{(s-1)/s}}$$

$\frac{dx}{dt}$ = flow velocity at "x" (cm/min)

$$= sk^{1/s} x^{(s-1)/s}$$

$\dot{v}$ will be exactly v_o/t_o if $sx^{(s-1)/s} = L^{(s-1)/s}$.

Denoting this "x" by "x_o," we have

$$\left(\frac{x_o}{L}\right)^{(s-1)/s} = \frac{1}{s}$$

$$x_o = L/s^{s/(s-1)} \text{ (cm)}$$

$$A = \dot{v}/(dx/dt)$$

$$A = (v_o/t_o)/(k^{1/s} L^{(s-1)/s}) \text{ (cm}^2\text{)}$$

A. Single Layer (Water at Zero Degrees)

$$s^{s/(s-1)} = (0.437)^{0.437/-0.563}$$

$$= 1.90$$

$$x_o = L/1.90$$

$$A = (v_o/t_o)/k^{1/s} L^{(s-1)/s}$$

$$= \frac{0.614}{(15.0)^{1/0.437}(22.85)^{-0.563/0.437}}$$

$$= 0.070 \text{ cm}^2$$

$$= 0.011 \text{ in.}^2$$

B. Double Layer (Water at Zero Degrees)

$$s^{s/(s-1)} = (0.514)^{0.514/-0.486}$$

$$= 2.02$$

$$x_o = L/2.02$$

$$A = (v_o/t_o)/k^{1/s} L^{(s-1)/s}$$

$$= \frac{4.06}{(18.94)^{1/0.514}(22.85)^{-0.486/0.514}}$$

$$= 0.257 \text{ cm}^2$$

$$= 0.040 \text{ in.}^2$$

II. Finding Predicted Wattages for Single Wick with Fluid at 80°F

A. Water at Zero Degrees

$$
\begin{aligned}
Q_{max} &= \dot{m}_{max} h_{fg} \\
&= \rho \dot{v} h_{fg} \\
&= (1)(0.074)(581)(1/14.35) \\
&= \left[\left(\frac{gm}{cm^3}\right)\left(\frac{cm^3}{\text{min-in. width}}\right)\left(\frac{cal}{gm}\right)\left(\frac{watts}{cal/min}\right)\right] \\
&= 3.0 \text{ watts/in. of width}
\end{aligned}
$$

$$
\begin{aligned}
Q_{heat\ pipe} &= (3.0)(2)\left[\left(2\ \frac{1}{16}\right)+\left(1\ \frac{7}{16}\right)\right] \\
&= [(\text{watts/in. width})\ (\text{in. width})] \\
&= 21.0 \text{ watts}
\end{aligned}
$$

B. Alcohol at Zero Degrees

$$
\begin{aligned}
Q_{heat\ pipe} &= (0.79)(0.011)(214)(1/14.35)(2)\left[\left(2\ \frac{1}{16}\right)+\left(1\ \frac{7}{16}\right)\right] \\
&= \left(\frac{gm}{cm^3}\right)\left(\frac{cm^3}{\text{min-in. width}}\right)\left(\frac{cal}{gm}\right)\left(\frac{watts}{cal/min}\right)(\text{in. width})\Big] \\
&= 0.91 \text{ watts}
\end{aligned}
$$

C. Fifty Percent Ethyl Alcohol at Zero Degrees

$$
\left. Q_{heat\ pipe} \right|_{upper\ limit} = \left. Q_{heat\ pipe} \right|_{h_{fg}\ of\ H_2O} = 1.7 \text{ watts}
$$

$$
\left. Q_{heat\ pipe} \right|_{lower\ limit} = \left. Q_{heat\ pipe} \right|_{h_{fg}\ of\ E.\ Alc.} = 0.5 \text{ watts}
$$