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STEADY STATE AND TRANSIENT PERFORMANCE

OF HOT RESERVOIR GAS CONTROLLED HEAT PIPES

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INTRODUCTION

In recent years there has been an enormous effort expended in heat pipe research and development. As a consequence, the technology has reached the point where heat pipes are being practically applied to solve a multitude of thermal control and heat transfer problems.

Most of these applications involve conventional heat pipes consisting of a sealed, wicked vessel containing only an appropriate quantity of working fluid and possessing no moving parts. Such a heat pipe does not have any particular operating temperature. Instead, it automatically adjusts its temperature to match the heat source and heat sink conditions so as to maintain conservation of energy. There are many applications, however, in which a specific operating temperature range is desired at either the evaporator or condenser portion, in spite of variations in the source and sink conditions. In those cases, it becomes necessary to actively or passively control the heat pipe so that it maintains the desired temperature range.

There are three fundamental approaches to controlling heat pipes: (1) control of the vapor flow, (2) control of the liquid flow, and (3) control of the heat transfer into or out of the device. All of these approaches have been explored with varying degrees of success [1, 2, 3, 4].

This paper is concerned with one of the techniques in the latter category; that is, the use of inert gases to passively control the heat output from the condenser. In particular, the paper deals with the effects of the presence of working fluid, in either liquid or vapor state, within the reservoir of "hot reservoir" systems.

PRINCIPLE OF GAS CONTROLLED HEAT PIPES

During operation of a heat pipe, the vapor always flows from the evaporator to the condenser region. As a consequence, any non-condensible gas present in the vapor is swept along and accumulates at the condenser end, forming a gas plug. This gas plug acts as a diffusion barrier to the flowing vapor and effectively "shuts off" that portion of the condenser which it fills. Consequently, by varying the length of this gas plug one varies the active condenser area and, hence, the heat transfer from the system.

The basic heat pipe, shown in Figure 1a, accomplishes this variation in condenser area passively. By introducing a fixed mass of gas into the system shown, a certain portion of the condenser section is filled with gas and is effectively "shut off", depending on the operating temperature of the pipe's active region. If the operating temperature increases, the vapor pressure of the working fluid increases. This compresses the non-condensible gas into a smaller volume, thus providing a greater active condenser area. On the other hand, if the operating temperature falls, the vapor pressure of the working fluid falls and the fixed mass of gas expands to a greater volume, thus blocking a larger portion of the condenser. The net effect is to provide a passively controlled variable condenser area which increases or decreases with the heat pipe temperature. As a consequence, this reduces the temperature response to variations in the heat input rate or sink conditions.

Bienert [3] has recently published an excellent paper analyzing the steady state performance of gas controlled heat pipes. In this paper, he has shown that control is improved by minimizing the percentage change in gas volume with movement of the gas front, thus demonstrating the desirability of large gas reservoirs (Figure 1b). He has also shown the advantage of thermally coupling the reservoir to the evaporator to minimize gas temperature fluctuations, as this also limits controllability. Such "hot reservoirs" are achieved by placing the reservoir near or actually within the evaporator (Figures 1c and 1d).

Bienert's analysis, however, assumed a complete separation of gas and vapor such that the gas pressure in the reservoir and inactive portion of the condenser equaled the vapor pressure in the active portion. This, however, can never be the case. There must always be a partial pressure of vapor in the gas region, and this has considerable implications as regards the steady state and, particularly, the transient performance of "hot reservoir" gas controlled heat pipes.

Hot Reservoir Heat Pipe Dynamics

The effects of working fluid in hot reservoirs become clear when one realizes that a hot reservoir must not be wicked. If it were, it would always contain liquid and there would exist within it a vapor pressure corresponding to its temperature. If the reservoir temperature equals the evaporator temperature (as in Figure 1d) the vapor pressures in each are equal and, since the total pressure is uniform throughout the system, there could be no gas in the reservoir. In other words, the reservoir would act as a heat pipe itself, displacing all of the gas into the condenser section, and thus would not be available to improve control. The same principle holds true for hot reservoirs which are not quite as hot as the evaporator (as in Figure 1c) although the gas displacement is not as complete.

However, it is not necessarily true that a hot reservoir which contains no wick also contains no fluid or vapor. Most heat pipes contain a slight amount of excess fluid and it is quite possible that circumstances could place this liquid within the reservoir (e.g., vibration during launch). Under such conditions, when the heat pipe is started up, the liquid in the reservoir will vaporize, forcing the gas out. Since the evaporator is acting similarly, the gas will be compressed to a relatively small volume in the condenser end, resulting in an operating pressure and temperature considerably higher than design conditions. This condition will prevail as long as there remains liquid or vapor in the reservoir.

Unfortunately, it is generally the case in zero-g applications, and frequently so on earth, that the only mechanism by which fluid in the reservoir is eliminated is diffusion, and this is a rather slow process.

Transient Experiments

Several laboratory experiments were performed to demonstrate the phenomenon described above. Two internal reservoir heat pipes were fabricated as shown in Figure 2. The design details of the pipes are presented in Table 1. Both heat pipes were identically fabricated and processed except that pipe no. 2 had a perforated Teflon* plug blocking the opening of the reservoir. The purpose of the Teflon plug was to prevent liquid from entering the reservoir while permitting the gas to pass through. The pipes were instrumented with twelve thermocouples along their length and a pressure transducer attached to the fill tube. Heat was supplied by resistance wire wrapped around the pipe and insulated. The pipes dissipated heat by natural convection and radiation to ambient.

Liquid Penetration Experiments:

Figure 3 shows the results of several transient experiments with pipe no. 1 (without the Teflon plug). Two pressure-time histories are shown - one for a start-up with the reservoir free of working fluid and one for

*Reg. Trademark, E.I. Dupont de Nemours

which the pipe was vigorously shaken to assure its presence. (Vapor pressure is used as the ordinate rather than temperature, for it is a much more sensitive variable). The dry-reservoir run was made in the heat pipe mode (evaporator elevated) and its start-up transient came directly to operating conditions and remained there (slight variations with time represent changes in ambient temperature). The wet reservoir run was initially started in the reflux mode (condenser elevated) to prevent the liquid from running out by gravity. In this case the start-up transient came up to a pressure (and temperature) considerably above design conditions and remained there. If undisturbed, it would have remained at this condition until all the liquid in the reservoir vaporized and diffused out. Instead, after a period of time the pipe was simply tipped to the heat pipe mode allowing the liquid to run out by gravity. After a short pulse due to excess fluid moving from the hot to the cold end, the pipe exhibited a slow recovery transient as vapor remaining in the reservoir diffused out. Ultimately, the pipe reached equilibrium at conditions similar to those reached directly in the dry-reservoir start-up. Once at this condition, subsequently tipping the pipe horizontal, then to the reflux mode and then back to the heat pipe mode had no effect other than to generate the short thermal transients caused by the slight excess fluid moving from the cold to the hot end and vice versa.

The recovery by diffusion is actually a rather complex process and difficult to predict, for the diffusion length and vapor pressure difference which drive it are constantly changing. However, if one assumes a simple first-order process it can be shown that the characteristic relaxation time constant is given by:

$$\tau_{\mathcal{D}} = \frac{L_{\mathcal{D}} V_r}{A_{\mathcal{D}} \mathcal{D}_{vg}} \qquad (1)$$

where $\mathcal{D}_{vg}$ - diffusivity of gas-vapor pair

$L_{\mathcal{D}}$ - characteristic diffusion length

$A_{\mathcal{D}}$ - area of diffusion duct

V_r - reservoir volume

For these heat pipes, $\tau_{\mathcal{D}} \approx 250$ minutes, which, although not precise, does give order of magnitude agreement with the data.

When heat pipe no. 2 was tested in a similar way, no such temperature or pressure excesses were observed (Figure 4). The Teflon plug apparently prevented liquid from entering the reservoir when the pipe was shaken, for both start-up transients came directly to design operating conditions, and tipping the pipe back and forth between the reflux and heat pipe modes had no effect other than that due to excess fluid.

These results clearly demonstrated that the presence of liquid in the reservoir gave rise to high operating pressures and temperatures, as hypothesized, and that a perforated, non-wetting plug is apparently an effective method of impeding the mechanical insertion of liquid into the reservoir.

Vapor Penetration Experiments:

Although the perforated Teflon plug apparently prevented the passage of liquid, it certainly does not prevent the flow of vapor. This can also be a serious problem, for if conditions are temporarily such that vapor is driven into the reservoir, a subsequent return to normal operating conditions will be characterized by a relatively long diffusion transient as the vapor comes back out. This is clearly seen in Figure 5 which shows three pressure and temperature histories measured with heat pipe no. 2.

The top curves show the result of overdriving the pipe, that is, raising the power beyond the control range so that it operates as an ordinary heat pipe. The gas is thus compressed to a volume smaller than the reservoir allowing vapor penetration. Given sufficient time under such circumstances, vapor will continue to diffuse into the reservoir, displacing gas, until the vapor pressure in the reservoir equals that at its entrance. When the power was returned to its starting value, the pressure and temperature did too, but with the slow transient characteristic of the diffusion process necessary to rid the reservoir of the vapor which had occupied it.

The second history shows the result of operating the pipe backwards for a period of time. This was accomplished by placing a heater at the condenser end as well as the evaporator end and simply reversing heaters while maintaining power constant. Once again, when the heat supply was returned to normal, the pressure and temperature recovery of the heat pipe showed the now familiar diffusion characteristic.

The third history is shown for comparison and represents a simple step change in power while the pipe was functioning within its control range. In this case the transient is an ordinary thermal one involving no diffusion and consequently is much more rapid.

<u>Hot Reservoir Heat Pipes - Steady State Performance</u>

The preceding discussion and experiments dealt with the effect of working fluid (vapor or liquid) in the reservoir on the transient response of hot reservoir heat pipes. It is also important to consider what the observed results mean in terms of steady-state operation.

It was demonstrated that working fluid in the reservoir will eventually diffuse out. However, it will never all diffuse out, for the diffusion potential is the vapor pressure difference between the hot portion of the reservoir and its entrance. Thus, the vapor pressure in the reservoir will only fall to that corresponding to the temperature at the reservoir entrance. On the other hand, should the temperature at this point increase due to a power increase or change in sink conditions, vapor will diffuse into the reservoir until the vapor pressure corresponds to this new temperature.

Since it is true that the total pressure in the system is essentially uniform, and thus that any vapor in the reservoir must displace its equivalent in gas, it follows that variations in the condenser temperature at the reservoir entrance must affect the operating temperature in a more profound way than that due to changes in the gas temperature.

Analysis:

The effect of reservoir vapor pressure can be easily demonstrated with a simple analysis. Consider the model shown in Figure 6, and allow the following assumptions:

a. The interface between the active and shut-off portions of the pipe is very sharp.

b. Axial conduction can be neglected.

c. For small operating temperature variations, heat transfer from the vapor to the sink at the condenser is proportional to the temperature difference between the vapor and the sink.

d. The total pressure is uniform throughout the pipe.

With these assumptions, one can write the Newton equation for heat transfer from the condenser:

$$Q = h\pi D\, L_a\, (T_{va} - T_s) \tag{2}$$

where: Q - heat transfer rate

h - heat transfer coefficient

D - pipe diameter

L_a - active length of condenser

T_{va} - vapor temperature in active zone

T_s - sink temperature

To determine the active length, one sums the molar inventory of non-condensible gas in the pipe and reservoir using the perfect gas law.

$$n = \sum \frac{P_g\, V_g}{R_u\, T_g} \tag{3}$$

where: n - number of moles of gas

P_g - partial pressure of gas

V_g - volume occupied by gas

T_g - temperature of gas

R_u - universal gas constant

Within the inactive portion of the condenser and reservoir:

$$\left.\begin{aligned} P_g &= P_{va} - P_{vs} \\ T_g &= T_s \end{aligned}\right\} \qquad (4)$$

where: P_{va} - vapor pressure at T_{va} (total pressure)

P_{vs} - vapor pressure at T_s

Within the active portion of the reservoir:

$$\left.\begin{aligned} P_g &= P_{va} - P_{vs} \\ T_g &= T_{va} \end{aligned}\right\} \qquad (5)$$

Note that the vapor pressure in the reservoir has been set equal to that at sink conditions. This is based on the assumption of a sharp vapor-gas front, a completely shut-off inactive portion (no axial conduction) and diffusion equilibrium between the reservoir and conditions at its entrance.

Performing the summation in Equation (3) using the conditions of Equations (4) and (5), and solving for L_a yields:

$$L_a = \frac{\dfrac{V_p}{T_s} - \dfrac{n\,R_u}{(P_{va}-P_{vs})}}{\dfrac{A_p}{T_s} - \dfrac{A_r}{T_{va}}} - L_e \qquad (6)$$

where: V_p - total void volume of pipe

L_e - length of insulated evaporator section

A_p - internal cross-sectional area of pipe envelope

A_r - Internal cross-sectional area of reservoir

Now, substituting Equation (6) into Equation (2), and performing some algebra, one obtains:

$$Q = h\pi D\,(T_{va} - T_{vs})\,L_c \left[\frac{\dfrac{V_p}{T_s} - \dfrac{n\,R_u}{(P_{va} - P_{vs})}}{\dfrac{L_c\,A_p}{T_s} - \dfrac{L_c\,A_r}{T_{va}}} - \frac{L_e}{L_c} \right] \qquad (7)$$

Equation (7) is the characteristic operating equation for these heat pipes. Using this equation, one can parametrically study the variation in any one variable with any other.

Of particular interest is the term in brackets, for this represents the fraction of the available condenser length which is active. Within this bracket, the term $nR_u/(P_{va} - P_{vs})$ reflects the effect of reservoir vapor pressure on the heat pipe. If P_{vs} is neglected, the sink temperature influences performance only through T_s. Clearly, unless P_{vs} is very much smaller than P_{va}, this is not a negligible effect.

Steady State Experiments:

To demonstrate this effect, steady state data were collected for heat pipe no. 2 at various power levels. The resulting temperature distributions are shown in Figure 7.

To use these data as a test of Equation (7), it is best to normalize it with respect to the "full-on" heat dissipation rate: $h\pi D\,(T_{va} - T_s)\,L_c$. The resulting terms, which represent the fraction of active condenser should theoretically vary between zero and one:

$$0 \leq \frac{Q}{h\pi D\,(T_{va}-T_s)\,L_c} = \frac{L_a}{L_c} = \left[\frac{\dfrac{V_p}{T_s} - \dfrac{nR_u}{(P_{va}-P_{vs})}}{\dfrac{L_c\,A_p}{T_s} - \dfrac{L_c\,A_r}{T_{va}}} - \frac{L_e}{L_c} \right] \leq 1 \quad (8)$$

One can now compare the results for L_a/L_c with the term in brackets, all of which are accurately known or measured. The results of this comparison are shown in Figure 8 (solid lines), which presents predicted (Equation 8) and measured operating pressure and temperature as a function of L_a/L_c. Apparently, the model is quite good at low values of L_a/L_c but gets progressively worse as L_a/L_c approaches unity.

If one examines Figure 7, it is clear that major drawbacks in the theoretical model are the assumptions of a sharp front between the vapor and gas and the neglect of axial conduction. The fact that the transition length from T_{va} to T_s is substantial means that the values used for the gas temperature are somewhat in error. More important is the fact that axial conduction causes the temperature at the reservoir entrance to rise well before the full-on region reaches it. Consequently, the vapor pressure in the reservoir becomes substantially higher than that corresponding to T_s, causing a displacement of gas, and thus a rise in pressure and temperature of the system above those predicted by Equation (8).

That this is actually the dominant effect is also shown in Figure 8, by the dotted lines. These curves also represent Equation (8) but the measured wall temperature at the reservoir entrance has been substituted for T_s and (P_{vs}) within the brackets. With this substitution, the theory much more closely agrees with the data. Precise agreement is not expected since these measured wall temperatures do not necessarily equal the liquid-vapor interface temperatures.

The Nature of the Vapor-Gas Front

The previous section shows the relative merits and inadequacies of a gas-controlled heat pipe model based on the assumption of a sharp vapor-gas front and a neglect of axial conductivity. To bring the theory more closely into line with behavior, one must account for the actual temperature and pressure profiles at the vapor-gas interface. This is a fairly complex problem involving the internal gas dynamics coupled with conduction in the wall and heat transfer to the sink, and is beyond the scope of this paper. However, for completeness, it should be mentioned that this problem has been studied numerically and that certain general conclusions are available. In particular, the sharpness of the front depends most strongly on:

1. the working fluid-gas combination
2. the heat dissipation rate per unit length of pipe
3. the pipe diameter
4. the axial conductivity of the pipe wall and any attached fins

The last factor, axial conductivity, is particularly important. As an example, Figure 9 shows predicted temperature profiles both neglecting and including conduction in the wall as compared with measured data for a water heat pipe operating at 210°F and dissipating heat to ambient. The effect of axial conduction in spreading the front is obvious.

Conclusions and Design Implications

The results of the experiments and analysis reported in this paper lead to several important conclusions:

1. In order for hot reservoir heat pipes to operate at design conditions, it is necessary to keep liquid out of the reservoir. If liquid is present the pipe will operate at a substantial overpressure and temperature at start-up, and will relax only by diffusion of vapor out of the reservoir - a very slow process. For cases in which normal operating pressure is substantial, this pressure pulse should be accounted for in the structural design of the pipe.

2. A perforated non-wetting plug placed in the reservoir entrance has been found effective in impeding mechanical insertion of liquid.

3. Even if liquid is not present in the reservoir, diffusion plays a large role in the heat pipe dynamics. Thus, one should design the pipe so as to minimize the diffusion time constant.

4. An excellent way to minimize diffusion-dominated transients is to assure that the vapor pressure at the reservoir entrance never exceeds that corresponding to sink conditions. This can be accomplished by making the condenser somewhat longer than necessary and/or by including a "cold finger" region at this point.

5. The sharp front - no axial conduction model is fairly good at predicting performance as long as the actual front is fully developed at the entrance to the reservoir. However, it is necessary to include the reservoir vapor pressure in this model.

6. The actual front length is strongly dependent on axial conductivity in the walls. To obtain sharp fronts and avoid the need for excessively

long condensers, one should minimize this by using thin walled tubes of low conductivity materials and exercising care in the design of fins.

7. Finally, the vapor pressure in the reservoir limits the degree of control possible on both hot (non-wicked) and cold (wicked) reservoir gas controlled heat pipes. For close control, one must minimize the variation in this vapor pressure as a percentage of the operating pressure. This means that either (a) the sink conditions must not vary widely, or (b) the vapor pressure at maximum sink temperature must be small compared with the total pressure.

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NOMENCLATURE

$A_{\mathcal{D}}$ - Area of diffusion duct

A_p - Internal cross-sectional area of heat pipe

A_r - Internal cross-sectional area of reservoir

D - Heat pipe outside diameter

$\mathcal{D}_{vg}$ - Diffusivity of vapor-gas pair

h - Coefficient of heat transfer

$L_{\mathcal{D}}$ - Characteristic diffusion length

L_a - Length of active portion of condenser section

L_e - Length of insulated evaporator section

n - Number of moles of non-condensible gas

P_g - Partial pressure of non-condensible gas

P_{va} - Vapor pressure of working fluid at T_{va}

P_{vs} - Vapor pressure of working fluid at T_s

Q - Heat transfer rate

R_u - Universal gas constant

T_g - Temperature of non-condensible gas

T_s - Sink temperature

T_{va} - Vapor temperature in active zone

$\tau_{\mathcal{D}}$ - Characteristic diffusion time constant

V_g - Volume occupied by non-condensible gas

V_p - Total void volume of heat pipe

V_r - Volume of gas reservoir

L_c - Length of condenser section

Table I

HEAT PIPE DESIGN DETAILS

Working Fluid: Methanol

Inert Gas: Nitrogen at 10 psia

Pipe:

Material:	Stainless steel
Inside Diameter:	0.800 in.
Outside Diameter:	0.8750 in.
Length:	23 in.

End Plates:

Material:	Stainless steel
Thickness:	0.125 in.

Gas Reservoir:

Material:	Stainless steel
Inside Diameter:	0.490 in.
Outside Diameter:	0.5625 in.
Length:	21.875 in.

Wicking:

Material:	94 mesh 304 stainless steel screen
Description:	2 wraps on heat pipe I.D. 1 wrap on gas reservoir O.D. Multi-wrap transfer wick at end of evaporator

---Nominal Diffusion Time Constant: $\tau_D \simeq 250$ minutes

CAPTIONS

Fig. 1 Schematic Diagram of Various Gas-Controlled Heat Pipe Configurations

Fig. 2 Schematic Diagram of Experimental Heat Pipes

Fig. 3 Transient Start up Test Results for Internal Reservoir Gas Controlled Heat Pipe Without Teflon Plug (Pipe No. 1)

Fig. 4 Transient Start up Test Results for Internal Reservoir Gas Controlled Heat Pipe With Teflon Plug (Pipe No. 2)

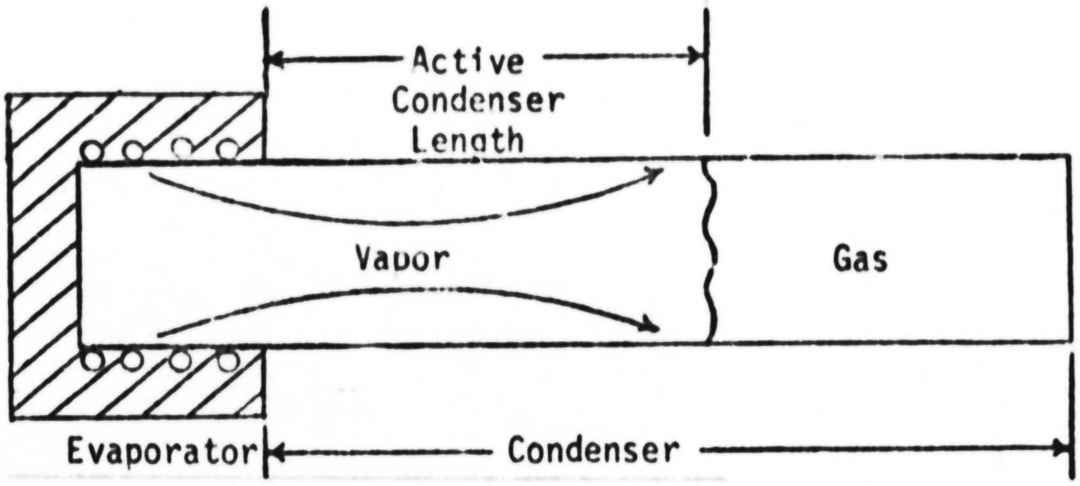
Fig. 5 Transient Test Results of Vapor Penetration Experiments (Heat Pipe No. 2)

Fig. 6 Definition of Analytical Model

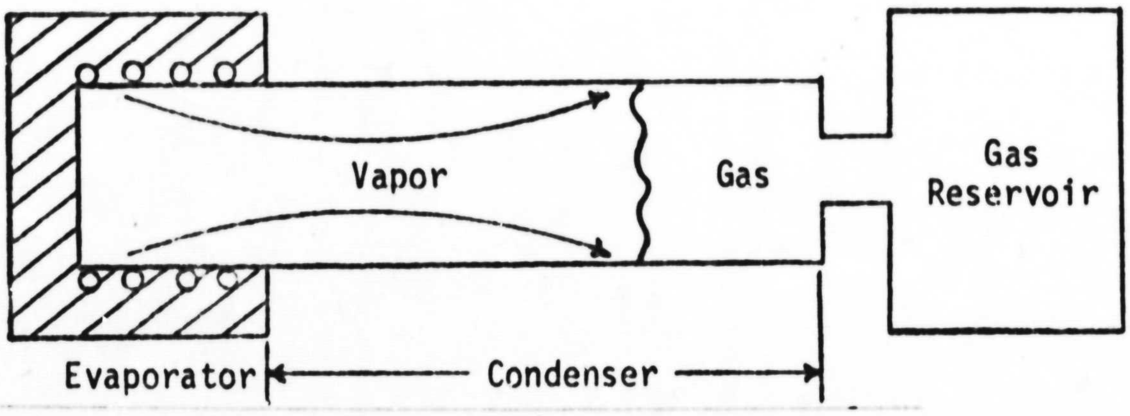
Fig. 7 Steady State Temperature Distributions (Heat Pipe No. 2)

Fig. 8 Steady State Evaporator Temperature and Pressure vs. Active Condenser Length. Comparison of Experimental Data with Theory.

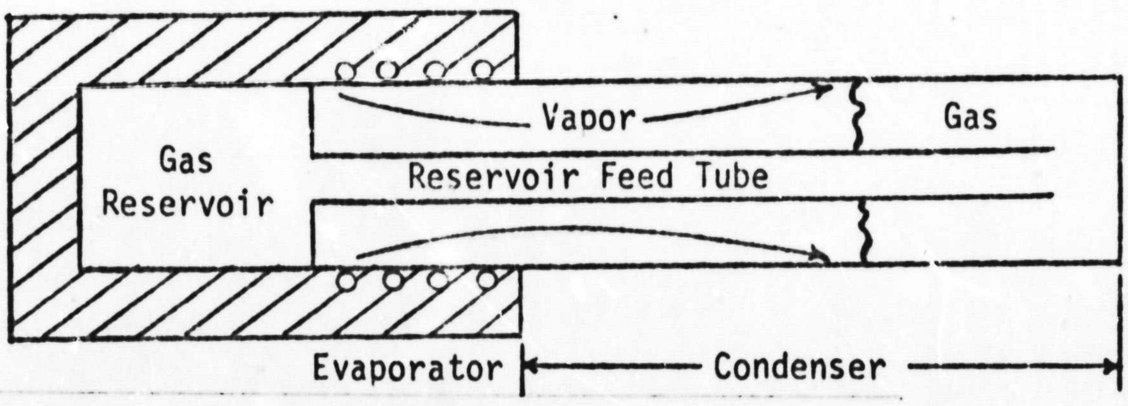
Fig. 9 Comparison of Predicted and Measured Temperature Distributions Across the Vapor-Gas Front of a Laboratory Heat Pipe



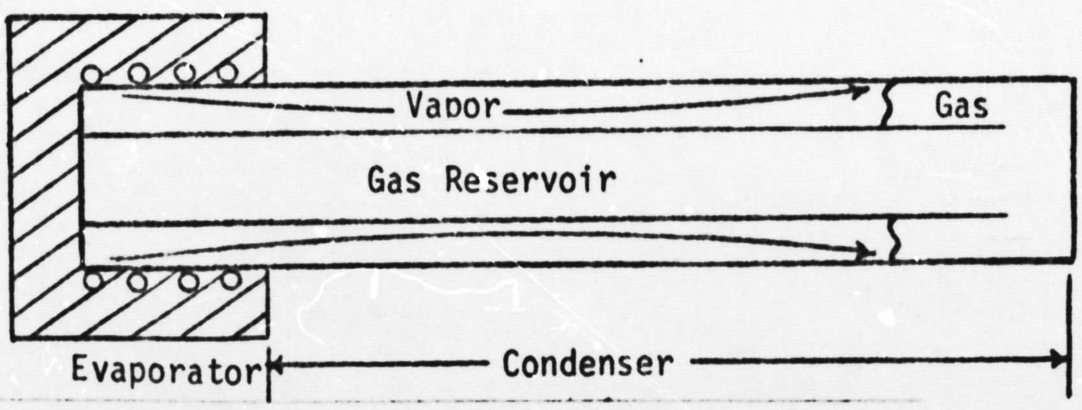
(a) Basic Heat Pipe with Inert Gas



(b) Heat Pipe with External Cold Gas Reservoir



(c) Heat Pipe with External Hot Gas Reservoir



(d) Heat Pipe with Internal Hot Gas Reservoir

FIGURE 1

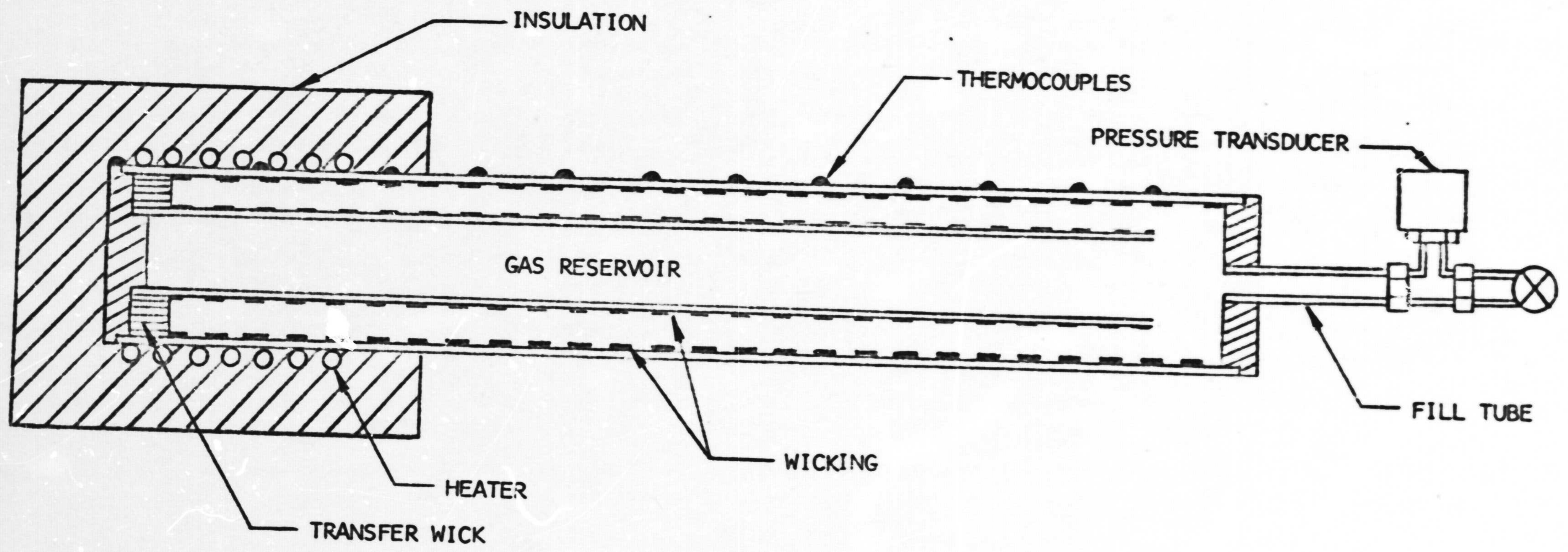
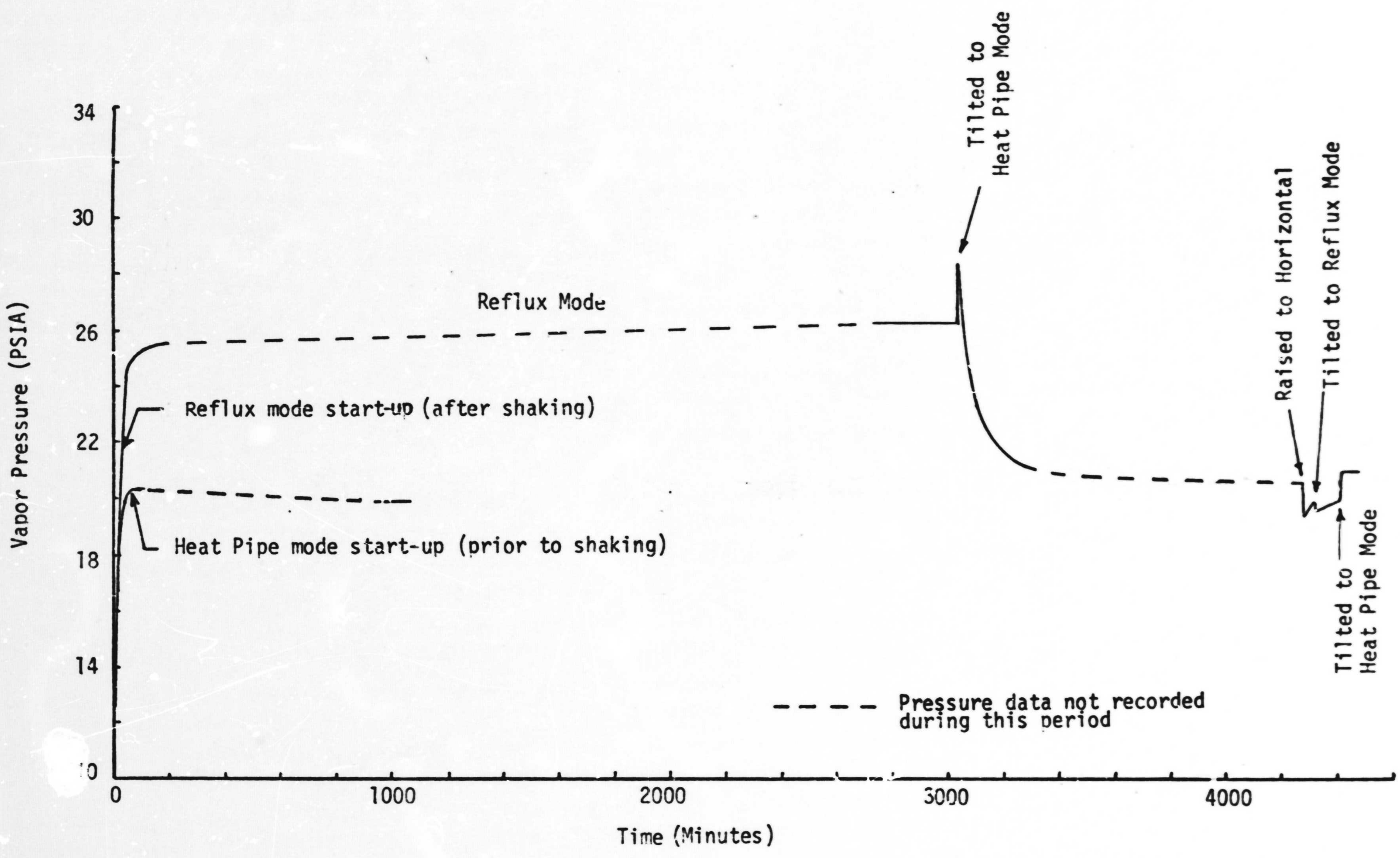

INSULATION
THERMOCOUPLES
PRESSURE TRANSDUCER
GAS RESERVOIR
FILL TUBE
WICKING
HEATER
TRANSFER WICK

FIGURE 2



Vapor Pressure (PSIA)
34
30
26
22
18
14
10
0
1000
2000
3000
4000
Time (Minutes)
Reflux Mode
Reflux mode start-up (after shaking)
Heat Pipe mode start-up (prior to shaking)
Tilted to Heat Pipe Mode
Raised to Horizontal
Tilted to Reflux Mode
Tilted to Heat Pipe Mode
Pressure data not recorded during this period

FIGURE 3

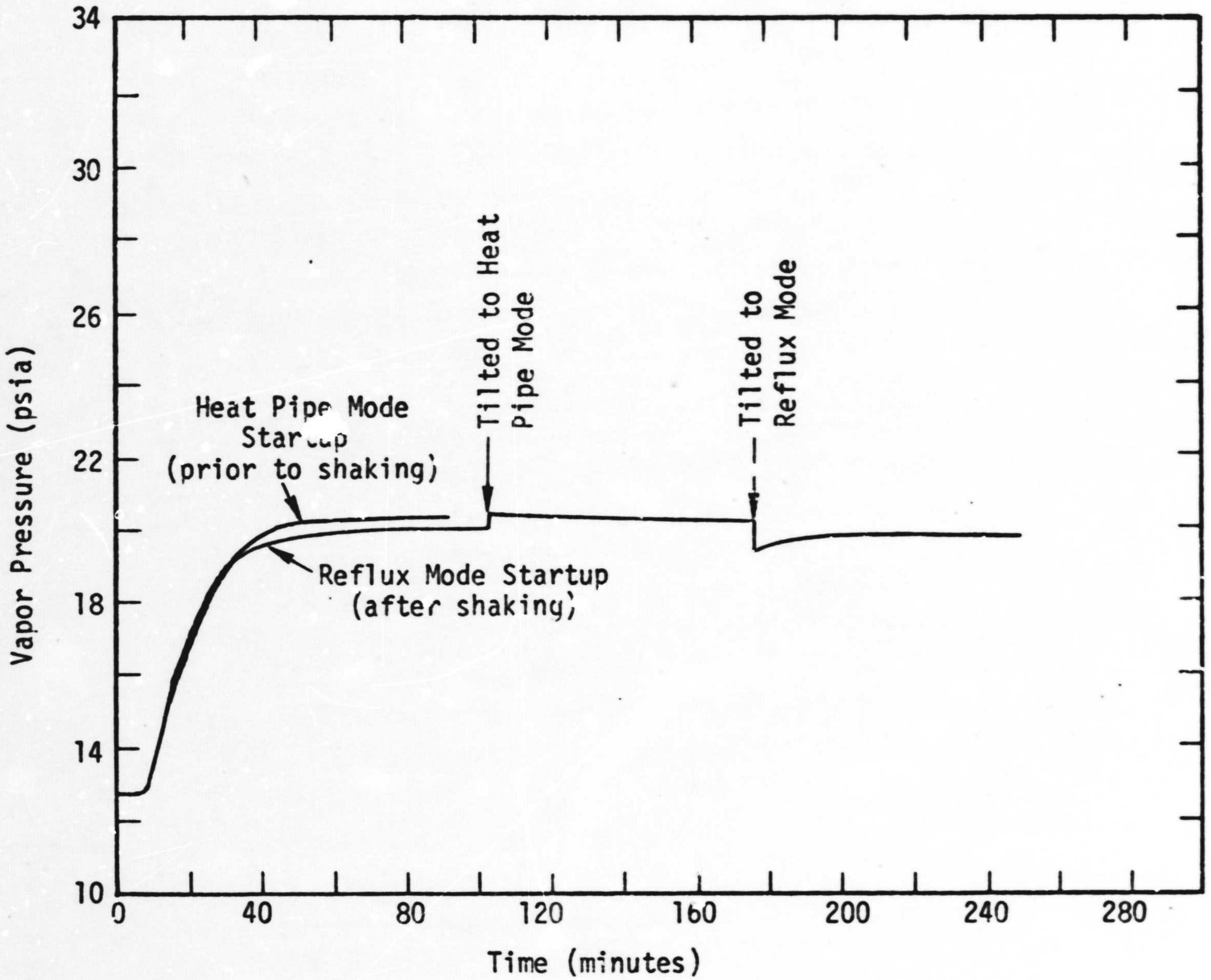
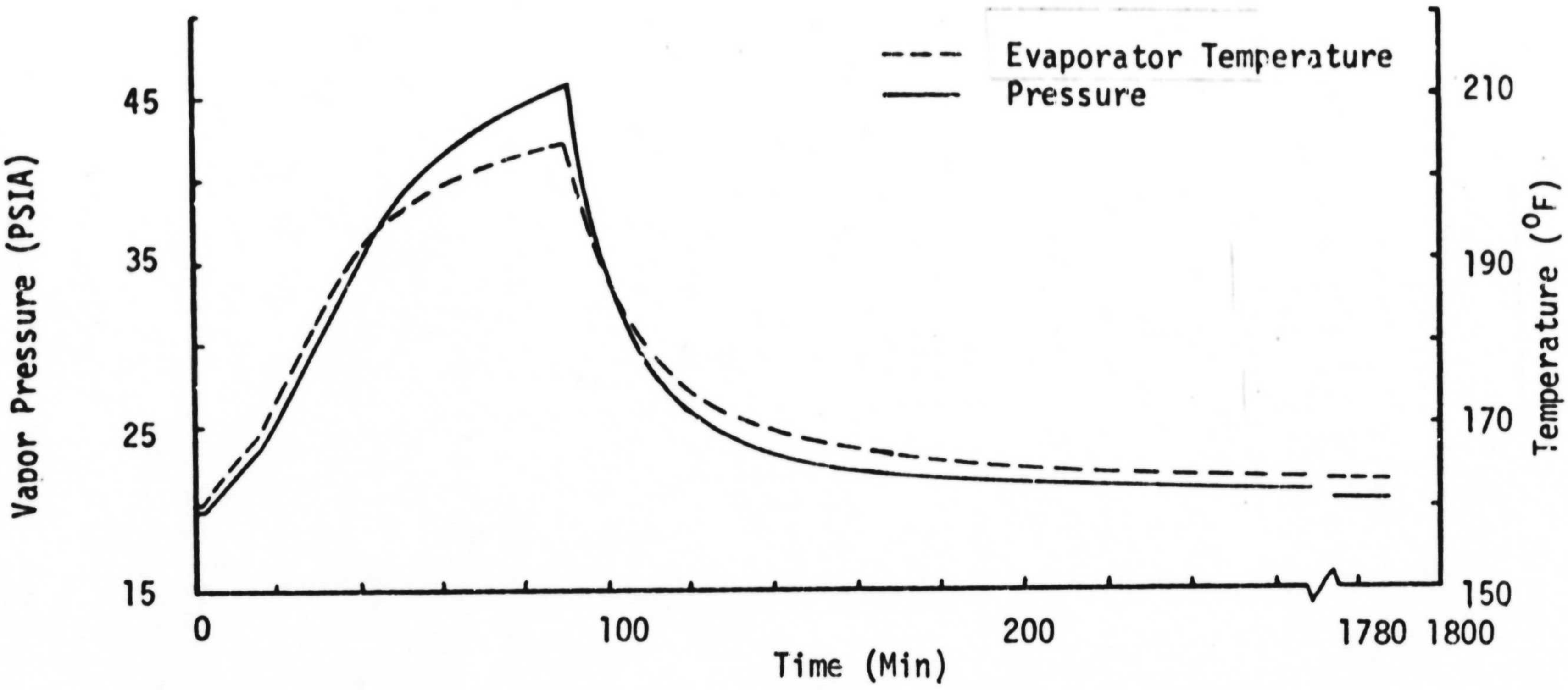
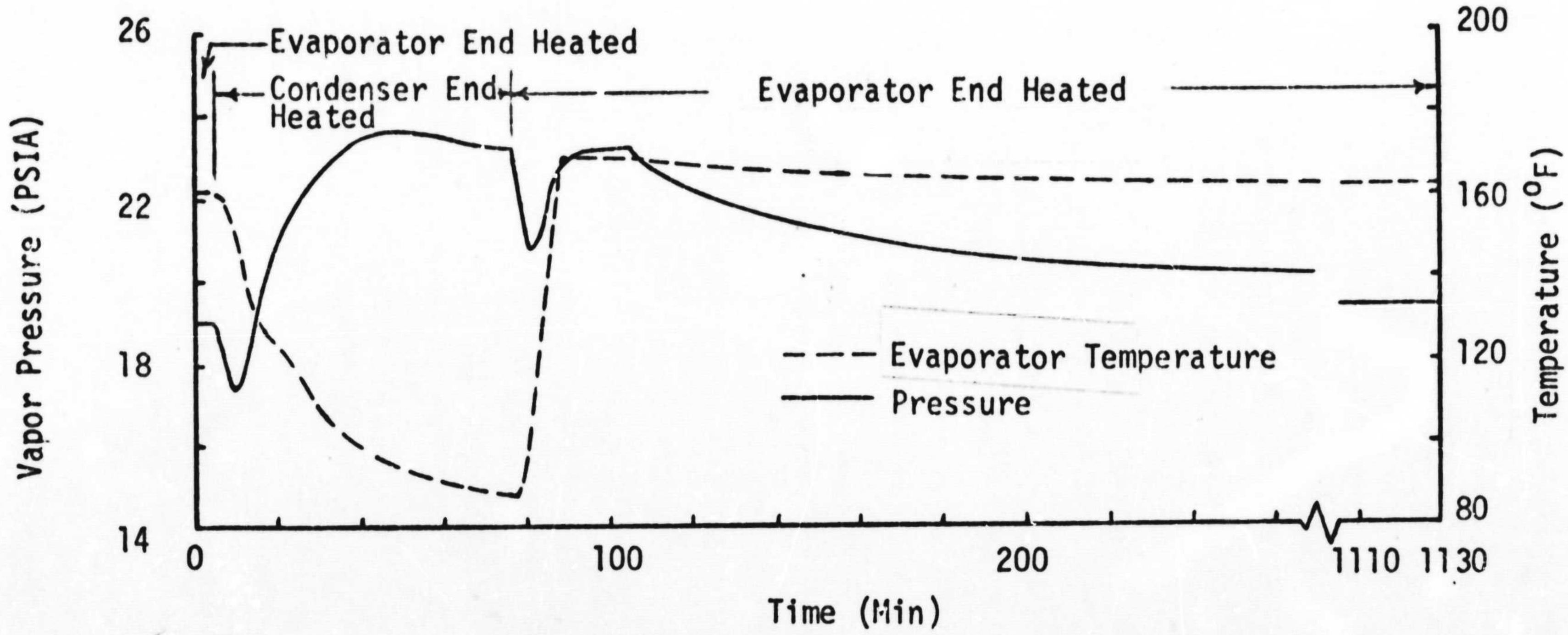

Vapor Pressure (psia)
34
30
26
22
18
14
10
0
40
80
120
160
200
240
280
Time (minutes)
Heat Pipe Mode Startup (prior to shaking)
Reflux Mode Startup (after shaking)
Tilted to Heat Pipe Mode
Tilted to Reflux Mode

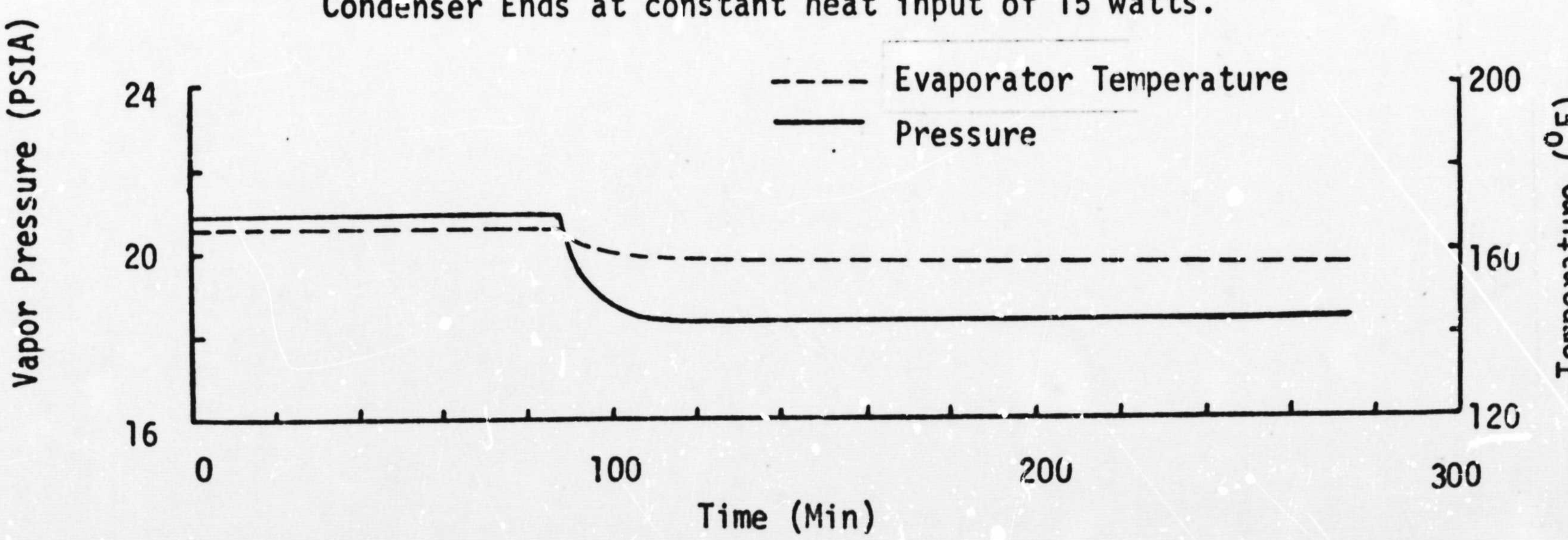
Figure 4



a. Result of overdriving the pipe to a heat input of 30 watts



b. Result of reversing the heat source between the evaporator and Condenser Ends at constant heat input of 15 watts.



c. Step change in power from 15 watts to 10 watts.

FIGURE 5

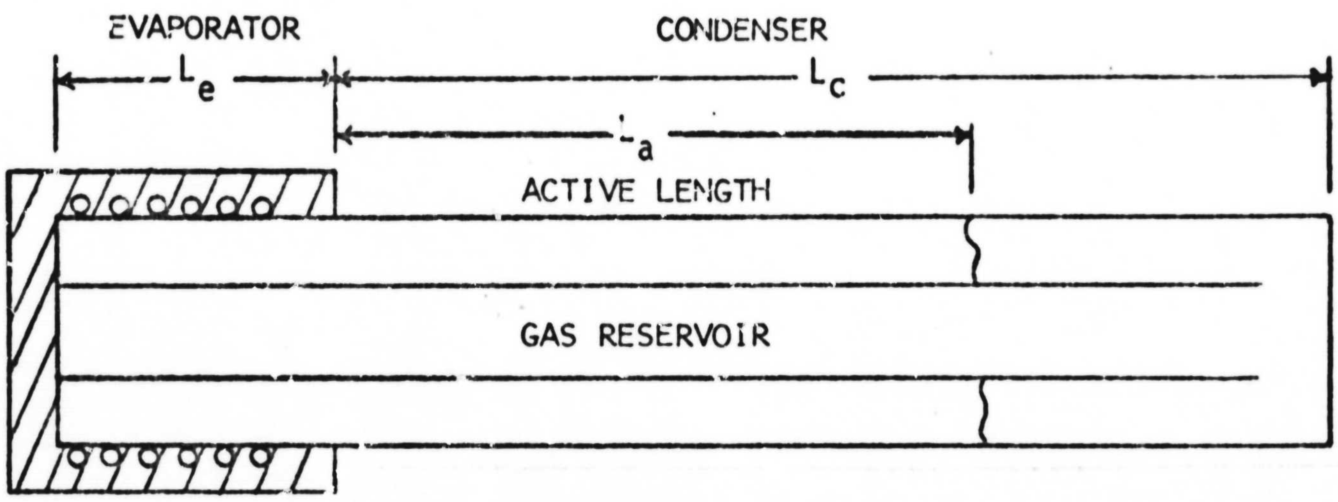
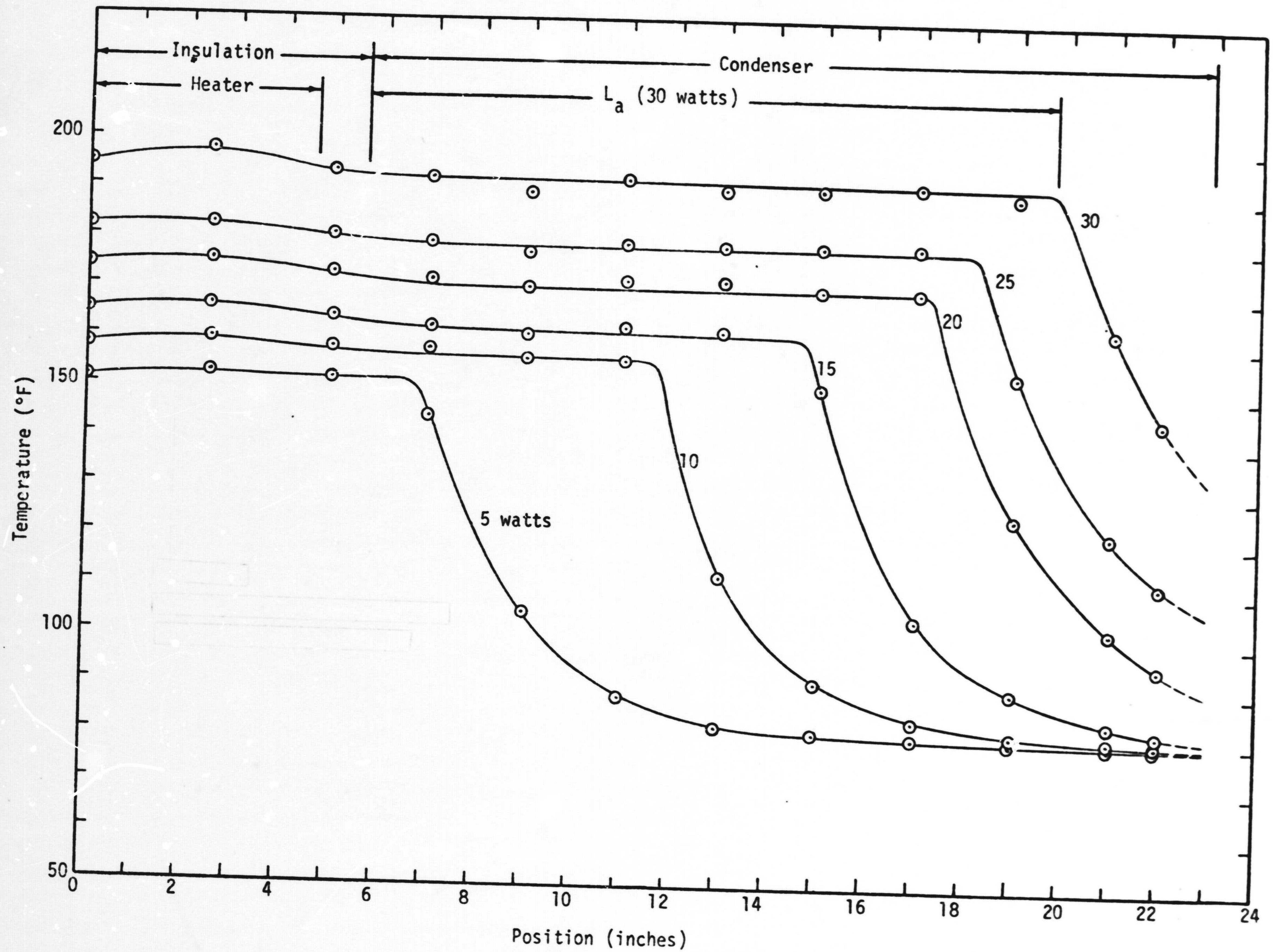
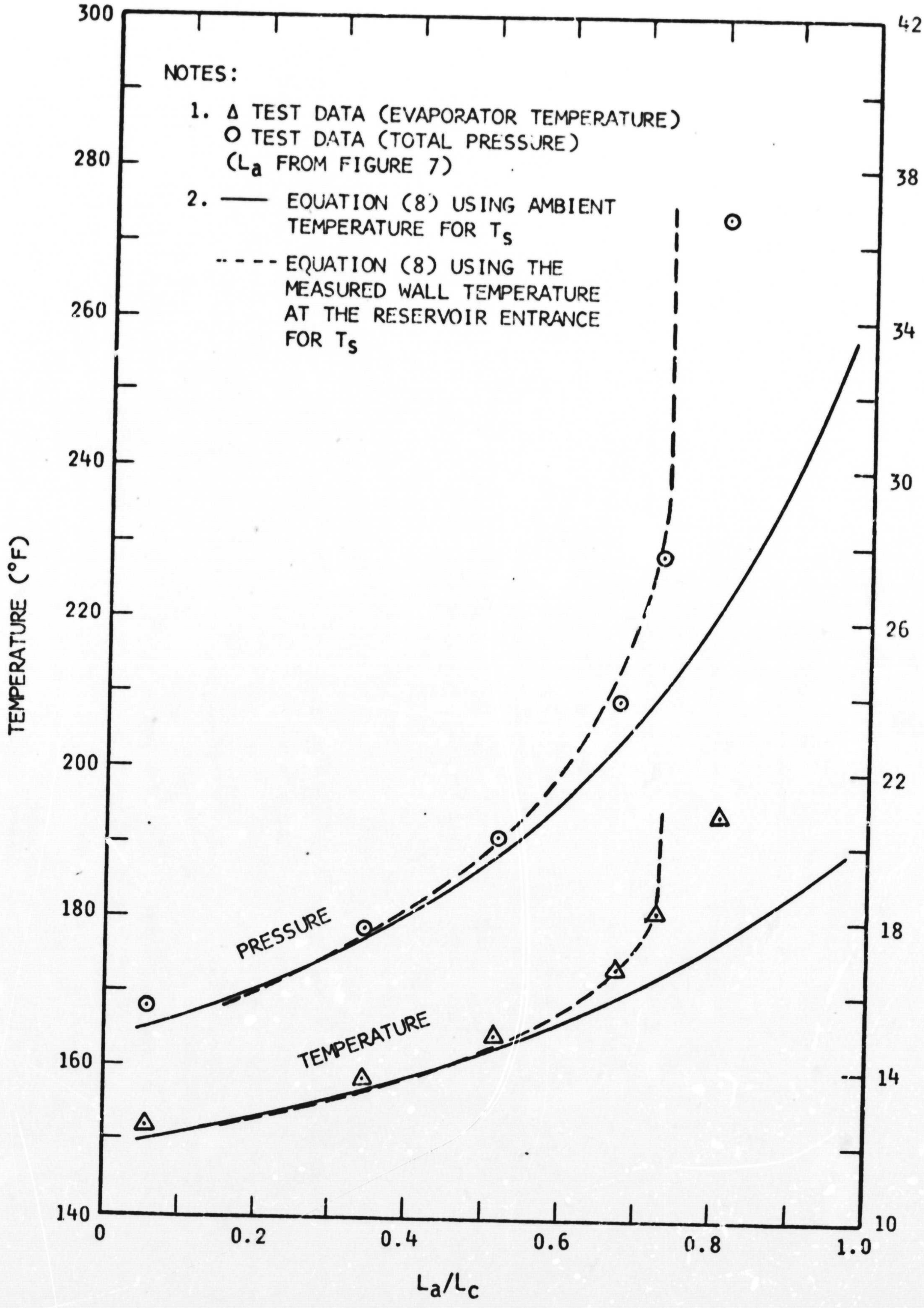


FIGURE 6



Insulation
Heater
Condenser
L_a (30 watts)
200
150
100
50
0
2
4
6
8
10
12
14
16
18
20
22
24
Temperature (°F)
Position (inches)
5 watts
10
15
20
25
30

FIGURE 7



NOTES:
1. Δ TEST DATA (EVAPORATOR TEMPERATURE)
O TEST DATA (TOTAL PRESSURE)
(L_a FROM FIGURE 7)
2. —— EQUATION (8) USING AMBIENT TEMPERATURE FOR T_s
---- EQUATION (8) USING THE MEASURED WALL TEMPERATURE AT THE RESERVOIR ENTRANCE FOR T_s
PRESSURE
TEMPERATURE
TEMPERATURE (°F)
VAPOR PRESSURE (PSIA)
300
280
260
240
220
200
180
160
140
42
38
34
30
26
22
18
14
10
0
0.2
0.4
0.6
0.8
1.0
L_a/L_c

FIGURE 8

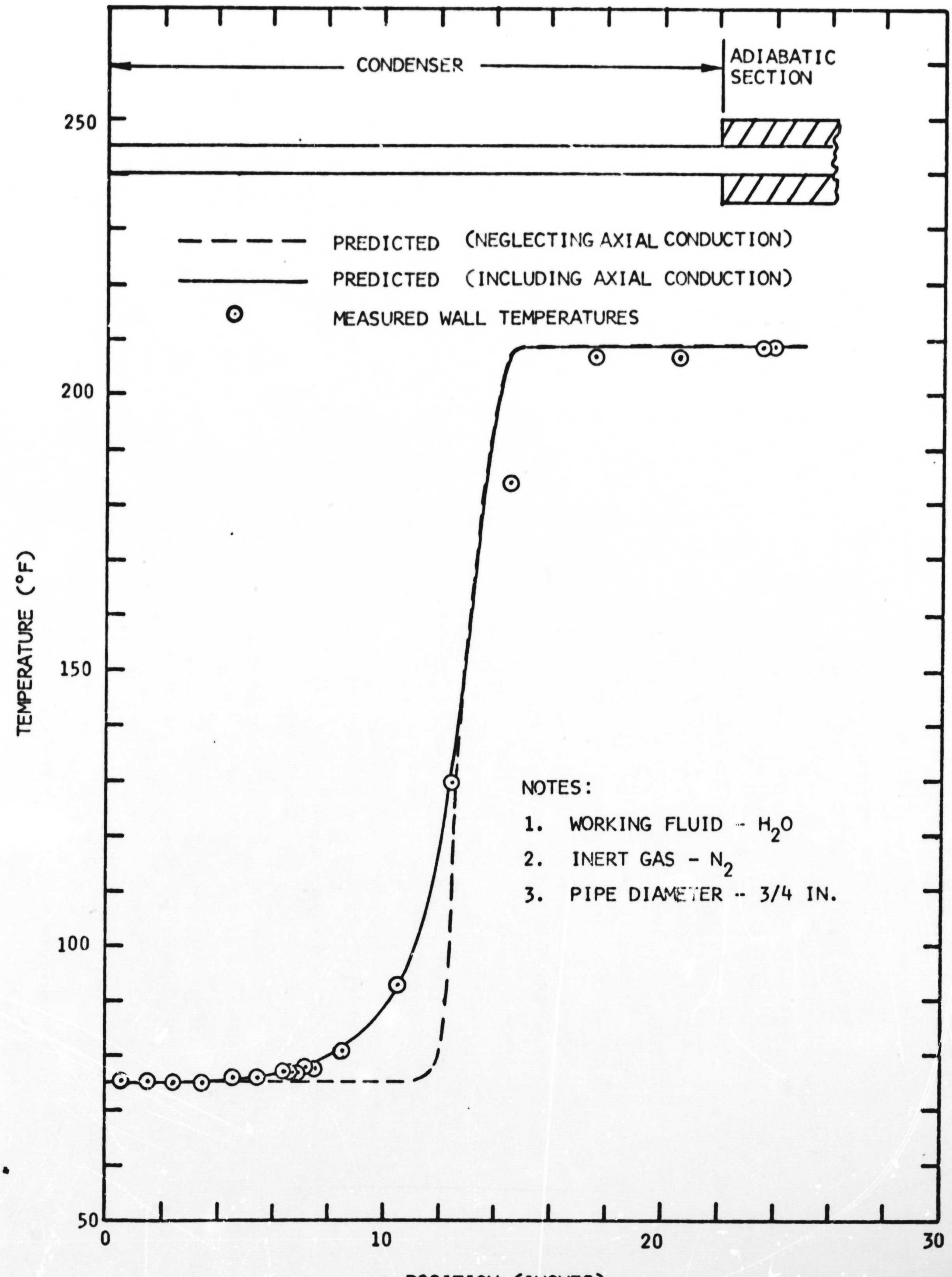

CONDENSER
ADIABATIC SECTION
PREDICTED (NEGLECTING AXIAL CONDUCTION)
PREDICTED (INCLUDING AXIAL CONDUCTION)
MEASURED WALL TEMPERATURES
NOTES:
1. WORKING FLUID - H_2O
2. INERT GAS - N_2
3. PIPE DIAMETER - 3/4 IN.
TEMPERATURE (°F)
250
200
150
100
50
0
10
20
30
POSITION (INCHES)

FIGURE 9