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Transient Response of a Gas-Controlled Heat Pipe

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The paper presents a combined analytical and experimental investigation of the transient response of a gas-controlled heat pipe with an absorption gas reservoir. A mathematical model which considers the heat conduction in the condenser wall, the overall vapor core temperature, the diffusion of the vapor-gas interface, and the absorption of the control gas in the working fluid/wick structure matrix of the reservoir is developed for analysis. Experimental evaluation of the heat-pipe performance is made with methanol as a working fluid and ammonia and nitrogen as control gases. The combination of ammonia and methanol is considered for the soluble gas absorption reservoir, while nitrogen and methanol is considered for the standard gas reservoir. The performance of the two is compared. It has been observed that, to a limited extent, the performance can be improved by the absorption gas reservoir. Both theoretical and experimental results exhibit the same features of the performance of the heat pipe.

Nomenclature

A = surface area of reservoir
A_x = vapor mole fraction
A_{x_i} = vapor mole fraction at interface
C = concentration distribution
C^0 = initially dissolved gas concentration in reservoir
C^* = gas concentration at absorption surface
D = coefficient of diffusivity
D_{ab} = diffusion coefficient $a \rightarrow b$
D_o = outer diameter of heat pipe
D_i = internal diameter of heat pipe
D_v = diameter of vapor space
h = heat-transfer coefficient
K_i = thermal conductance of liquid-saturated wick
K_w = wall conductance
l = arbitrary length
l_c = length of condenser
l_g = length of gas block
M_b = solvent molecular weight
p = pressure distribution
Q = heat input
R = physical absorption
R_G = universal gas constant
T = temperature of solvent
T_0 = sink temperature
T_i = temperature at interface
T_v = vapor temperature
T_w = wall temperature
T_∞ = vapor temperature in adiabatic section
t = time interval
U = vapor velocity
X = axial coordinate in reservoir
Z = axial coordinate in condenser
ρ = density
λ = latent heat of evaporation
ψ = association parameter
μ_b = solvent viscosity
σ_d = molecular diameter

Introduction

A GAS-controlled heat pipe is a device that generally employs a reservoir and maintains a relatively stable heat-source temperature by a stagnant gas leg in the condenser which moves in response to varying heat-source or sink conditions. Basically the technique involves the addition of a fixed quantity of gas which is swept to the end of the condenser and into the reservoir volume, thereby effectively shutting off the portion of the condenser it fills. The changes in the heat-source or sink condition cause a change in the length of the gas plug and consequently vary the surface area of active condenser for heat transfer (to the surroundings). The temperature of the heat source is controlled: 1) by placing the reservoir at the end of the condenser (cold reservoir) so that its temperature depends on the sink temperature and fluctuates with variations in thermal environment and 2) by coupling the reservoir thermally with the evaporator (hot reservoir) so that its temperature range corresponds to the control range of the heat pipe. The first case requires a large-volume gas reservoir. The volume of the reservoir, however, can be reduced by replacing it by a gas absorption reservoir as proposed by Saaski.[1] In this technique, the solubility of the control gas in the working fluid of the heat pipe is utilized in order to reduce the volume of the gas reservoir. The present study is directed toward characterizing the transient response of a gas-controlled heat pipe with absorption.

Inherent in the design of a gas-controlled heat pipe with an absorption reservoir is the necessity of understanding the interaction between the working fluid vapor and the control gas. An adequate design implies the ability to predict the location of the interface between the vapor and gas as a function of heat input, source or sink temperature, and any other variable parameter. Further, knowledge of the solubility data for the control gas and working fluid is essential for an appropriate combination of the two.

Saaski's analytical model assumes a sharp interface between the vapor and gas and neglects both the wall conduction and the mass diffusion between them. Although these assumptions do not have much influence on qualitative prediction of the heat-pipe performance, the quantitative results differ significantly from experimental observations.

Edward and Marcus[2] have presented an improved analytical model to predict the performance of gas-controlled heat pipes. Their model includes the wall conduction as well as the diffusion between the working fluid and control gas. Sun and Tien[3] neglected the diffusion process and developed a simple conduction model to predict the wall temperature. Their analysis also includes the phenomenon of phase change in the evaporator wick. The predictions based on these theories seem to agree well with the experimental results; however, they are restricted to the steady state only.

From a system design point of view, it is of utmost importance to have a knowledge of the transient response of gas-controlled heat pipes. The transient response of the wicked-reservoir heat pipe is primarily controlled by sensible heat

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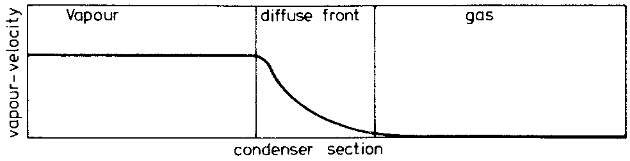

Fig. 1 Typical vapor velocity profile.

transfer from its various elements and vapor/gas diffusion effects. Further, the movement of the vapor/gas front depends mainly on the vapor dynamics. Thus, the vapor dynamics also plays a significant role in the transient performances.

The present analysis is performed for a gas-loaded heat pipe with an absorption reservoir and includes vapor dynamics, axial conduction in the walls, vapor/gas diffusion, and gas absorption in the reservoir. The effect of a change in the heat inputs on the performance is studied further. A comparison is made on the overall performance of the gas-absorption reservoir with those of the standard gas reservoir.

Analysis

Vapor Dynamics

A typical velocity profile of the working liquid vapor is shown in Fig. 1. The velocity of the vapor in the condenser section is found by considering the equations of motion and continuity in that region. Treating the vapor as an incompressible fluid and considering the flow in a channel, we can write the equation of motion

$$\frac{\partial U}{\partial t}+U\frac{\partial U}{\partial Z}=-\frac{1}{\rho}\frac{\partial p}{\partial Z} \tag{1}$$

and the equation of continuity

$$\partial U/\partial Z=0 \tag{2}$$

If l denotes the length of the gas plug at time t, we have

$$U=-\dot{l} \tag{3}$$

Substituting Eqs. (3) and (2) into Eq. (1), we obtain

$$-\ddot{l}=-\frac{1}{\rho}\frac{\partial p}{\partial Z} \tag{4}$$

which after integration with respect to Z gives

$$l\ddot{l}=\frac{1}{\rho}\left[p(T)-p(T_\infty)\right] \tag{5}$$

or,

$$l\ddot{l}=\frac{p(T_\infty)}{\rho}\left[\frac{p(T)}{p(T_\infty)}-1\right]=-Ax_iR_GT_\infty \tag{6}$$

Equation (6) is solved with help of the initial conditions

$$\dot{l}(0)=0,\quad l(0)=0 \tag{7}$$

The vapor mole fraction at the interface and the interface temperature T_i are then related by the Clausius-Clapeyron equation in the following way:

$$A_{x_i}=1-\exp\left[-\frac{\lambda}{RT_\infty}\left(\frac{T_\infty}{T_i}-1\right)\right] \tag{8}$$

At any time the heat transferred to the heat pipes must be accounted for by the heat transfer from the condenser and by a temperature increase of the thermal mass. For ease of modeling, the thermal mass is always assumed to be in equilibrium with the vapor core temperature $T_v(t)$. The heat-transfer equation for the vapor is given by

$$(\rho C_p)_v\left(\frac{\partial T_v}{\partial t}+U\frac{\partial T_v}{\partial Z}\right)=Q-\frac{4}{D_0}h[T_w-T_0] \tag{9}$$

The initial and boundary conditions for T_v are written as

$$T_v(Z,0)=T_\infty \tag{10}$$

$$\partial T_v(0,t)/\partial Z=0 \tag{11}$$

Wall Conduction

The effect of axial wall conduction in the evaporator and adiabatic sections of heat pipes are of little importance because of negligible temperature drop in these sections. The control gas causes a sudden temperature drop in the condenser section and thereby the axial wall conduction becomes important in this section. For modeling purposes, the condenser section of the pipe is assumed to reject heat to the surroundings by convection. Accounting for these factors, we can write the heat-balance equation in the condenser section as

$$\rho C_p\frac{\partial T_w}{\partial t}=K_w\frac{\partial^2 T_w}{\partial Z^2}+8K_i\frac{(T_v-T_w)}{(D_o^2-D_i^2)\log(D_i/D_v)} -4\frac{D_oh(T_w-T_0)}{(D_o^2-D_i^2)} \tag{12}$$

The temperature of the wick is assumed to be very near the temperature of the vapor. The second term on the right-hand side of Eq. (12) vanishes because there is no flow of heat across the wick in the gas region. The heat balance equation in the gas region then becomes

$$\rho C_p\frac{\partial T_w}{\partial t}=K_w\frac{\partial^2 T_w}{\partial Z^2}-4\frac{D_oh(T_w-T_0)}{(D_o^2-D_i^2)} \tag{13}$$

Equations (12) and (13) are subjected to the following initial and boundary conditions

$$T_w(Z,0)=T_{w0} \tag{14}$$

and

$$\frac{\partial T_w(0,t)}{\partial Z}=\frac{\partial T_w(l_c,t)}{\partial Z}=0 \tag{15}$$

In addition, the continuity conditions at the interface are assumed to hold.

Vapor/Gas Diffusion

When the heat pipe operates, the vapor penetrates into the condenser. Subsequently, a pressure drop occurs at the interface of the control gas and the working liquid vapor. This causes diffusion of vapor into the gas. To describe the diffusion process at the vapor/gas interface in the condenser section, a quasisteady process at constant temperature and pressure for each instant of time is considered. Applying Fick's first law of diffusion, we have

$$(1-A_x)U=D\frac{dA_x}{dZ} \tag{16}$$

with the condition

$$Z=0, \qquad A_x=0 \tag{17}$$

Integrating Eq. (16) and using Eqs. (17), we obtain

$$A_x=1-\exp(-(U/D)Z) \tag{18}$$

The average mole fraction for a condenser length l is

$$\frac{1}{l}\int_0^l A_x \mathrm{d}Z=1+\frac{D}{Ul}\left[\exp\left(-\frac{U}{D}l\right)-1\right] \tag{19}$$

Thus the total mole fraction diffused is obtained by multiplying Eq. (19) by the vapor space volume in the condenser section.

Absorption

The absorption gas reservoir is used to store volumetrically more gas as dissolved solute than as dispersed gas in the conventional reservoir. When the control gas is forced out toward the reservoir, it is absorbed by the liquid available in the reservoir wick. The absorbed part of the gas in the reservoir is obtained by solving the diffusion equation in the reservoir

$$\frac{\partial C}{\partial t}=D\frac{\partial^2 C}{\partial x^2} \tag{20}$$

with the initial and boundary conditions

$$C(x,0)=C^0, \quad C(0,t)=C^* \tag{21}$$

Equation (20) is solved with the help of the initial and boundary conditions [Eqs. (21)]. The solution yields

$$\frac{C-C^*}{C^0-C^*}=\mathrm{erfc}\left(\frac{Dx}{2\sqrt{\pi t}}\right) \tag{22}$$

The physical absorption[4] is calculated as

$$R=-AD\frac{\partial C}{\partial x}\bigg|_{x=0}=2(C^*-C^0)A\sqrt{D/\pi t} \tag{23}$$

Mathematical Treatment and Numerical Solution

We have to solve: 1) the equation of motion (6) for l and l where the mole fraction A_{x_i} is obtained by Eq. (8); 2) Eq. (9) for the vapor/gas temperature under the initial and boundary conditions of Eqs. (10) and (11), respectively; and 3) Eqs. (12) and (13) for the wall temperature under the initial and boundary conditions of Eqs. (14) and (15), respectively.

As Eqs. (6), (9), (12), and (13) are interconnected, we proceed as follows. Start with the initial values of u and T_w. The vapor temperature is calculated from Eq. (9) at a time step Δt and, with this vapor temperature, the condenser wall temperature at the same time step is calculated from Eqs. (12) and (13). When the vapor/gas temperature is known, we estimate the vapor mole fraction A_{x_i} from Eq. (8) and then l and $\dot{l}$ from Eq. (6). From this l the total mole fraction diffused is found from Eq. (19). At the same time the physical absorption is calculated from Eq. (23). On this basis, the total gas mole in the system is found and a comparison is made with the total gas mole filled in the heat pipe at the starting time. At each new time step, T_v and T_w are iterated until the difference between the calculated and actual values of the gas moles does not change more than an arbitrary ϵ per iteration. With the consistent set of variables, the integration in T_v and T_w proceeds to the next step.

Table 1 Design details of heat pipe

Working fluid	Methanol
Control gas	Ammonia, nitrogen
Pipe	
Material	Stainless steel
Dimensions, mm	
Inside diameter	10.5
Outside diameter	12.3
Evaporator length	150
Condenser length	350
Adiabatic length	100
Reservoir length	150

Table 2 Design details of heat-pipe wicks (Fig. 8)

	Mesh size	Layers of screen, mm	Screen wire diameter, mm	Screen wire spacing, mm
Outer layer	150	1	0.065	0.104
Layer 1	150	1	0.065	0.104
Layer 2	65	1	0.11	0.270
Webs	150	1	0.065	0.104

In practice, Eq. (6) must be transformed into a convenient form for integration. The second-order differential equation (6) is replaced by the system of two differential equations:

$$\dot{l}=y \qquad \dot{y}=-A_{x_i}R_G T_\infty/l \tag{24}$$

The above system of equations is solved by the trapezoidal rule.[5] Equations (9), (12), and (13) are partial differential equations of a parabolic type with initial and boundary conditions. A convenient method of solving these equations is one of finite difference. An explicit finite-difference scheme is worked out to solve the set of these equations. The predicted results can be compared with measurements.

Analytical Summary

A stainless-steel heat-pipe assembly composed of an evaporator, condenser, and adiabatic sections, together with a wicked reservoir, is considered for sample calculation to study the startup mechanism and the temperature control capability of a gas absorption reservoir heat pipe. The geometrical dimensions used for the calculations are specified in Tables 1 and 2. All calculations are performed using methanol as the working fluid. Ammonia, having a higher absorptivity coefficient in methanol, is considered as a control gas for the study. Nitrogen with an almost negligible absorptivity coefficient is considered as a reference to establish the control capability of a soluble gas absorption reservoir over a standard reservoir. The coefficients of diffusivity of these gases in methanol are calculated by the expression[6]

$$D_{ab}=\frac{9.76(10^{-8})(M_b\psi)^{1/2}T}{\mu_b\sigma_a^{1.8}}$$

The thermal conductance of the liquid-saturated wick K_i and the heat-transfer coefficient at the sink conditions are unknown properties and can be determined only by experiment. However, due to lack of sufficient experimental data, the reference values of $H=6.0\times10^{-3}$ W/cm$^2\cdot$s and $K_i=0.04$ W/cm^2 are used. The other thermophysical properties used in the calculations are taken from Refs. 6 and 7.

The vapor temperature transient given in Eq. (9) is a function of the heat-pipe geometry, the heat-transfer coefficient from the wall to the sink being at the condenser and the

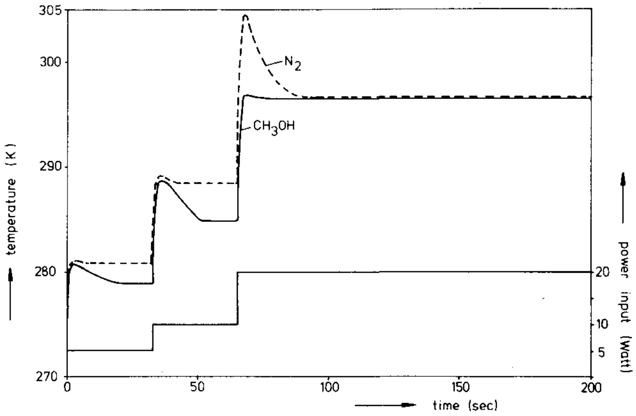

Fig. 2 Vapor temperature transients for a step change in power.

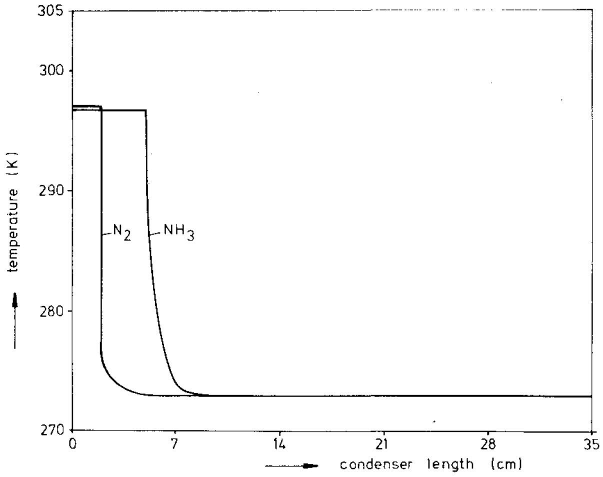

Fig. 3 Vapor-gas temperature distribution along condenser section.

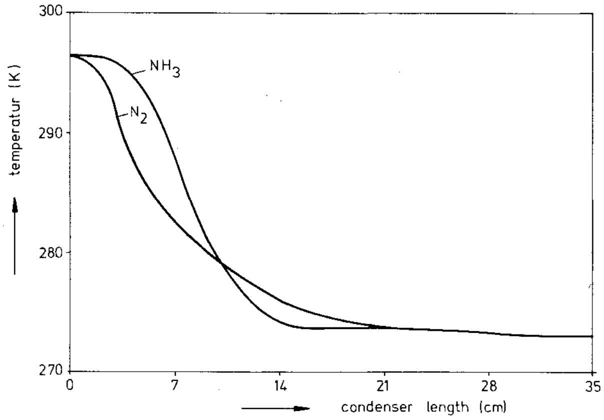

Fig. 4 Wall temperature along condenser section.

mole fraction of the control gas at the interface. Figure 2 represents these transients for various step changes in power input. A smaller temperature overshoot with a step change in power input is observed with ammonia than with nitrogen. This behavior is in quite good agreement with the physical process. Ammonia with higher absorptivity in methanol activates the condenser at a faster rate than nitrogen and hence minimizes the overshooting or undershooting of the vapor temperature and pressure. The role of conduction and diffusion in flattening the interface can also be seen from Figs. 3 and 4, where the profiles for the vapor/gas temperature and the wall temperature are presented along the condenser length for steady-state conditions. For various step changes in heat input, the vapor temperature is allowed to attain steady state. When this temperature does not change in subsequent time iteration for a while, a steady state is assumed. The heat-transfer rates corresponding to vapor temperatures in such cases are plotted in Fig. 5 to predict the performance of a gas absorption reservoir. It is observed that the control capability with absorption gas reservoir can be improved for a limited range of power input up to 15 W for the heat-pipe geometry under consideration. For larger power input, no improvement in the control capability of the gas control heat pipe with gas absorption reservoir is observed.

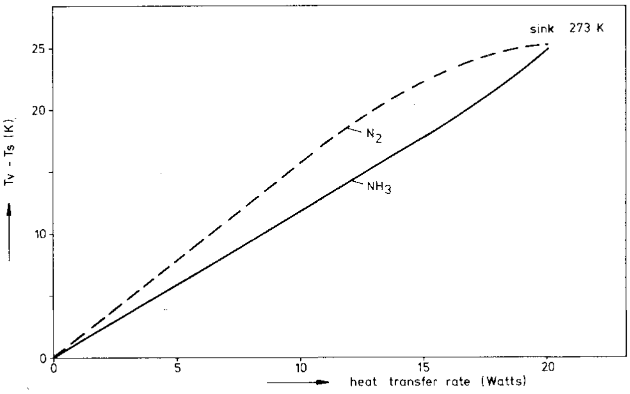

Fig. 5 Heat-transfer limitations of VCHP.

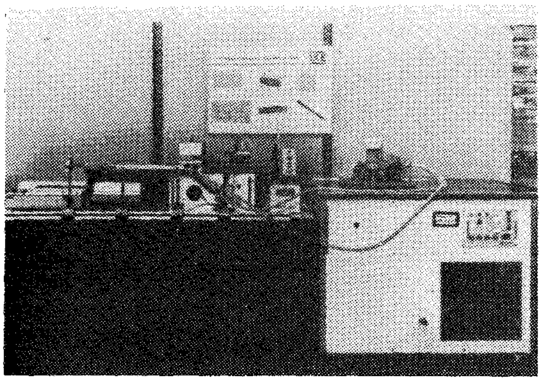

Fig. 6 Heat-pipe assembly.

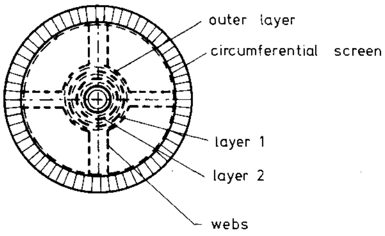

Fig. 7a Cross section of tunnel-type wick.

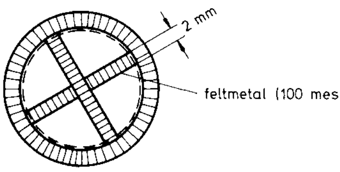

Fig. 7b Cross section of reservoir wick.

Experimental Verification

In the following sections, the experimental setup and the measurement technique are described and the results are interpreted in the light of the model described in the preceding sections.

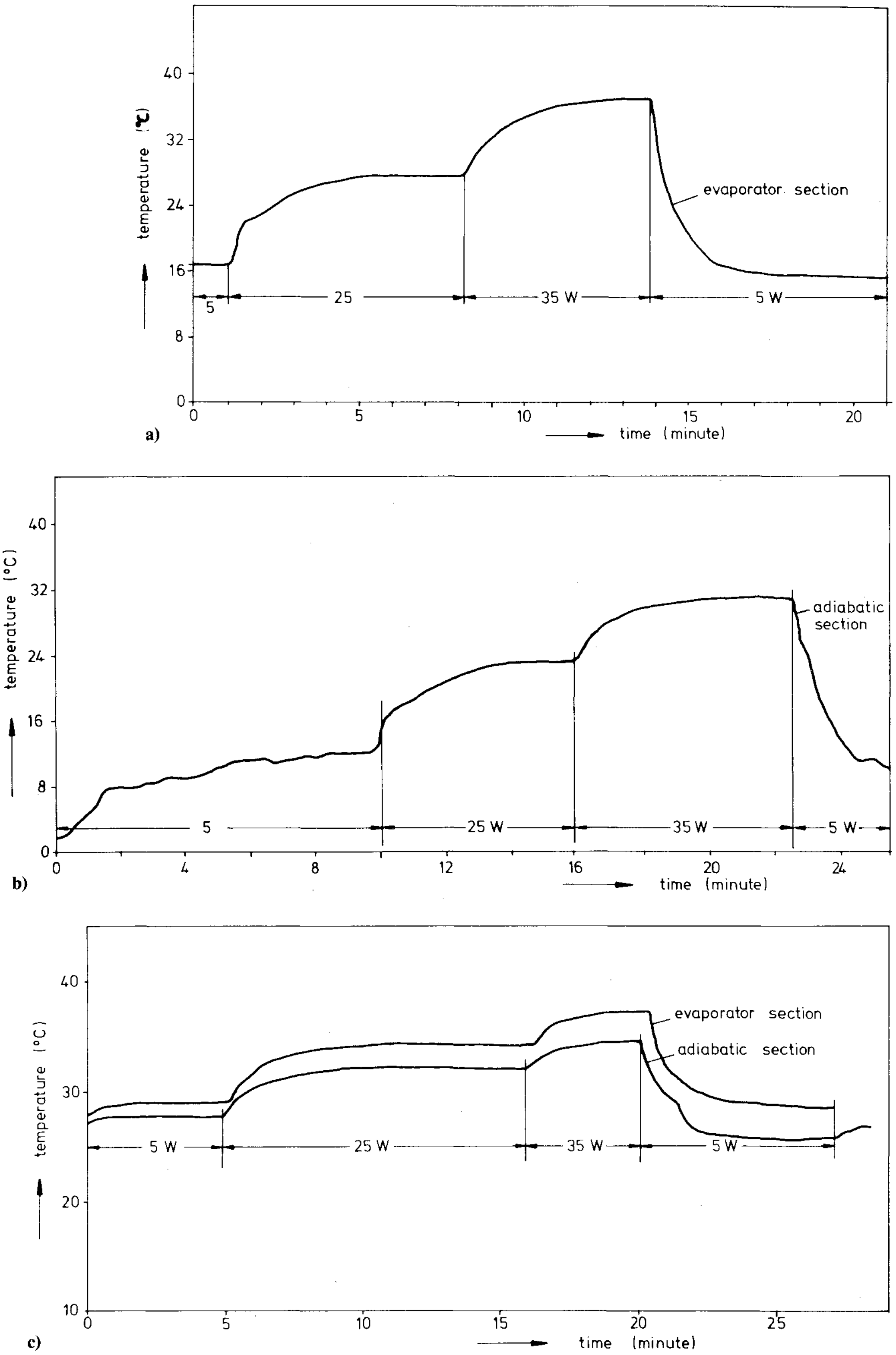

Fig. 8 Result of temporarily overdriving the pipe beyond its control range. a) Control gas: ammonia; b) control gas: ammonia; c) control gas: nitrogen.

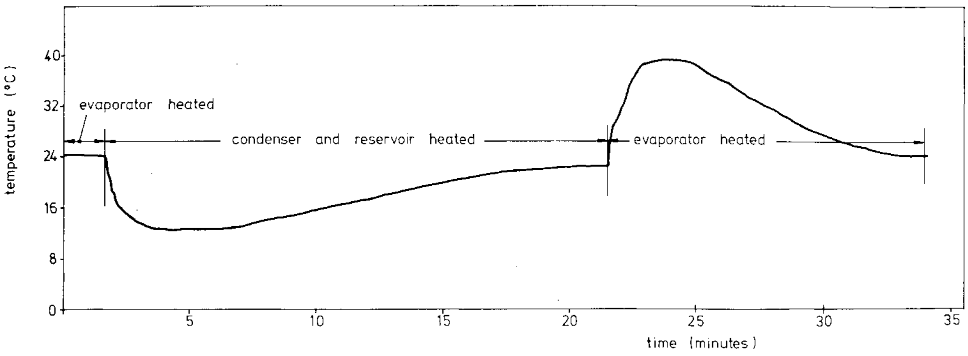

Fig. 9 Result of reversing the heat source between evaporator and condenser ends (evaporator temperature).

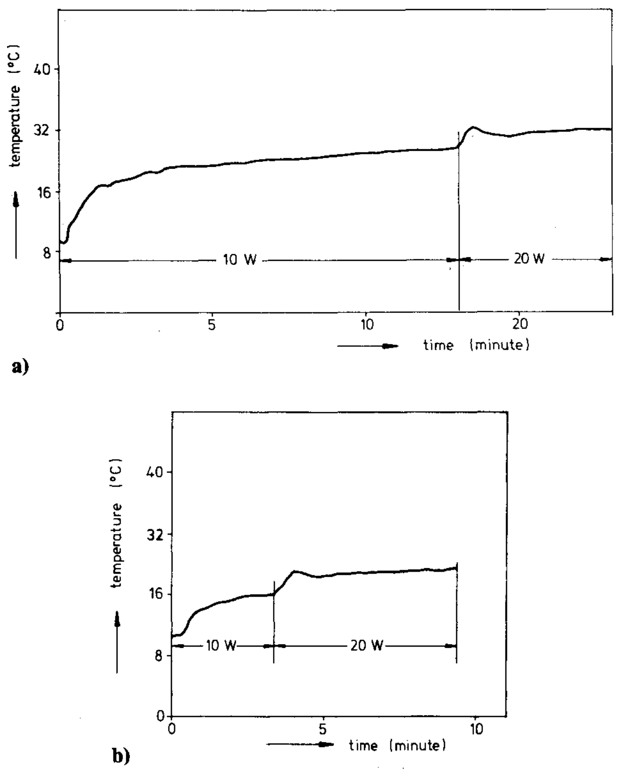

Fig. 10 Transient test results (evaporator temperature). a) Control gas: ammonia; b) control gas: nitrogen.

Experimental Setup

The following experimental arrangement is made to demonstrate the startup mechanism and temperature control capability of a gas absorption reservoir heat pipe. A basic heat-pipe assembly composed of evaporator, condenser, and adiabatic sections is considered for the experiment. The absorption reservoir heat pipe is fabricated as shown in Fig. 6. The structure of the wick is shown in Fig. 7. The design details of the system are presented in Ref. 9.

The heat pipe was instrumented with 12 thermocouples along the wall of the pipe and the reservoir. Heat was supplied by a resistance wire wrapped around the evaporator. The evaporator and the adiabatic section were well insulated against the surroundings. The pipe dissipates heat by convection and radiation to the ambient at the condenser section. The condenser section of the heat pipe is cooled by circulating methanol through a heat exchanger semidiametrically surrounding the condenser end. The methanol temperature can be varied from −80 to 80°C by means of a themostatically controlled cryostat. The reservoir temperature is regulated by another cryostat.

All tests were made using a methanol working fluid. The heat pipe was initially loaded with methanol. After charging the pipe, it was evacuated at elevated temperatures in order to remove any absorbed gas. Several hours of operation without noncondensible gas determined the performance of the heat pipe and the leak tightness. When the desirable performance was achieved, a suitable amount of control gas was injected into the pipe.

Several cycles at 10-35 W were run in order to stabilize the heat-pipe response. After stable response was achieved, the transient tests were made for the stepwise increase in power input. To insure the reproducibility of the heat-pipe performance, tests were also performed with a sudden decrease of the heat input after a test at a higher power level. Tests to predict the transient behavior and startup mechanism were also performed in the case of reversal of the heat input from the evaporator end to the condenser and reservoir end. To study the temperature control mechanism of the heat pipe, measurements were made for the stepwise increase of the heat input and the temperatures were recorded for near-steady-state situations.

Experimental Results

Transient Response of the Heat Pipe

To study the transient behavior of the gas-controlled heat pipe, various runs were made with the heat pipe in a horizontal position with step changes in power input. The results of the temperature distribution as a function of time are presented in Figs. 8-10. Figure 8 shows the result of overdriving the pipe, i.e., the power input is raised beyond control range and in this case the pipe operates as an ordinary heat pipe. Similar to the reasoning presented by Marcus and Fleischman,[8] the active zone of the condenser in this case extends to the entrance of the reservoir and the liquid/vapor diffuses into the reservoir, displacing the control gas. In this way, the control gas is compressed in the condenser region and, as stated earlier, minimizes the liquid transport capacity of the heat pipe and hence increases the temperature along the evaporator section, because the liquid transport capability is inversely proportional to the temperature drop across a wick. In such a case, the pipe does not operate as a constant temperature heat pipe but as an ordinary one. When the power was returned to its starting value, the temperature is also reduced to its original value but more slowly in case of ammonia as compared to nitrogen. The reason is obvious because of the slowness of the diffusion process which requires a longer transient in order to get rid of the liquid/vapor from the reservoir.

To insure the reproducibility of the heat pipe, it was operated for a certain period of time in the backward direction. This process was accomplished by raising the temperature of the reservoir and condenser by means of a thermostatically controlled cryostat and shutting off the heater at the evaporator. After a certain period of time, when the heat-pipe operation was made normal, the temperature of the evaporator returns to its original value and the same transient characteristic of the diffusion process is established. This process is depicted in Fig. 9. The transient behavior of the gas-controlled heat pipe for various changes in power input within its control range is shown in Fig. 10. The experimental results for the temperatures of the evaporator and the adiabatic section (this approximately corresponds to the vapor temperature) show a small overshooting at the step change of the power input in the case of ammonia but does not show any overshooting in that of nitrogen, while the theoretical curve depicted in Fig. 2 shows the opposite. It is quite possible that the heat pipe contained too little control gas nitrogen and the system temperature responded instantaneously to the change in power input. This is clear from the fact that the operating temperature of the pipe with control gas nitrogen is higher than predicted.

Diffusion Effects

Several cycles were run to stabilize the heat-pipe response. When it stabilized, the pipe was allowed to operate on a power input of 5 W and the temperature across the heat pipe was recorded by means of thermocouples in almost steady condition. Afterward the power input was changed to 15 W and the system was allowed to attain the steady state. When the steady state was achieved, the temperature across the length was recorded. This process was continued for the power inputs of 25 and 35 W. The temperature histories along the heat-pipe wall are presented for the various heat inputs in Fig. 11. From the temperature history curves, it is obvious that the actual front of the vapor interface is controlled by the processes of diffusion at the interface and the axial conduction in the pipe wall. This is also in accordance with the predicted results from the theory. The front is spread more with ammonia than with nitrogen because of the comparatively high diffusivity of ammonia in methanol.

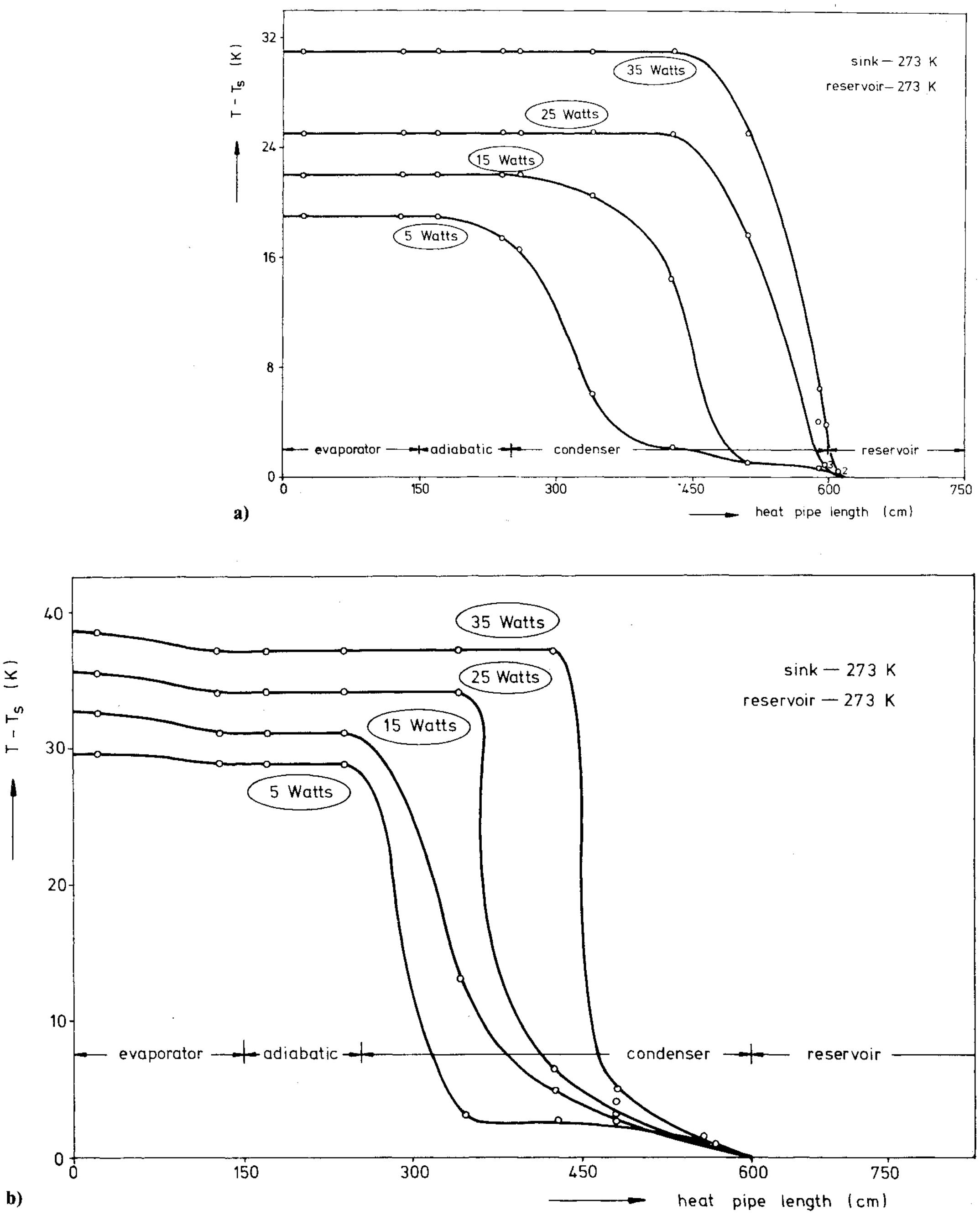

Fig. 11 Wall temperature along straight VCHP. a) Control gas: NH_3; b) control gas: N_2.

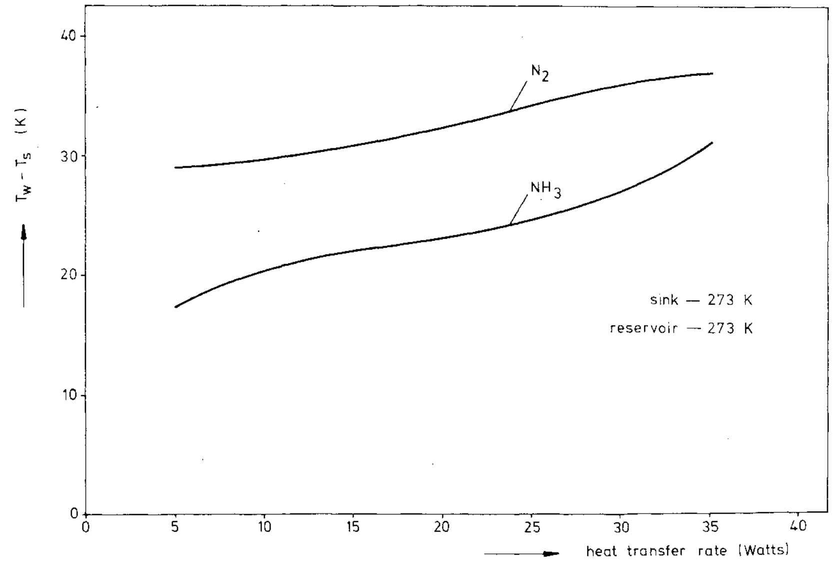

Fig. 12 Temperature difference between VCHP wall (adiabatic section) and condenser heat sink (control gas: ammonia).

Temperature Control

Figure 12 presents the results of the temperature difference between the adiabatic section of the wall of variable conductance heat pipe (VCHP) and the condenser heat sink as a function of the heat-transfer rate for the ammonia and nitrogen control gases. The performance could not be improved by the use of an absorption gas reservoir to the extent reported by Saaski in his experiment.[1] However, the predicted result in Fig. 6 reveals that the performance can be improved to some extent over a small range of power input.

Conclusions

The transient response of a gas-controlled heat pipe has been studied and it has been established that the overshooting or undershooting of the vapor temperature and the pressure can be minimized at the time of startup by the use of a soluble gas absorption reservoir. This demonstrates that stable response can be achieved more quickly with a soluble gas absorption reservoir than that with a conventional reservoir.

Thermal control can be improved by utilizing the absorption gas reservoir for a limited range of power input.

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