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Heat Pipe Thermal Control Set Point Shift

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Nomenclature

m	= ratio of reservoir volume to net volume injected during compression into reservoir
P	= total pressure, psia
p	= working fluid partial pressure, psia
$\bar{p}$	= average partial pressure of working fluid injected during compression, psia
$p(T)$	= saturation pressure of working fluid at temperature T psia
S_n	= fraction at final composition after n cycles
T	= temperature °R
$T(p)$	= saturation temperature at pressure p °R
V_{CAP}	= null volume including capillary tube volume, in.^{3}
V_I	= net injected volume; defined in Eq. (4), in.^{3}

Subscripts

c	= condenser
E	= evaporator
f	= final
i	= initial
R	= reservoir
n	= related to number of cycles

THIS Note reports a rapid procedure for estimating heat pipe set point drift behavior when end point conditions are known. Good agreement has been obtained between predictions and experimental data.

Variable conductance heat pipes possess substantial promise for providing spacecraft thermal control and have been studied extensively.[1] Various techniques have been proposed to improve the heat pipe thermal control by specialized design of the control gas reservoir. These have included the so-called wet wicked reservoir in which the heat pipe wicking is extended into the reservoir,[1] and various positive feedback options including electrical reservoir heaters,[2] and pressure actuated bellows.[3]

The dry or non-wicked reservoir is the simplest design. No additional wicking is required, control is less sensitive to reservoir temperature, and the reservoir location is less critical than for a wicked reservoir system. Control accuracy can be affected, however, by diffusion of the working fluid or by mass transfer from the condenser into the reservoir.[4] While diffusion between the condenser and reservoir can be controlled by using a capillary isolation tubing for some applications, sizing based on diffusion is inadequate to prevent the dynamic mass transfer which may accompany equipment or environment mission duty cycles.[5,6] Analytical techniques have been proposed which predict the change in thermal control associated with changes in the external thermal environment, in the heat input to the heat pipe, or with changes such as those which would accompany space radiator degradation.[6] Such analyses are sufficiently detailed, however, that a significant effort may be required to characterize the entire system. The following description defines an approximate technique so that set point drift predictions can be made easily. In addition, this technique permits incorporation of such experimental information as may be available.

The basic mechanism leading to set point drift is the injection of a mixture of working fluid vapor and noncondensable gas whose concentration differs from that already in the control gas reservoir. The set point temperature will increase if the working fluid vapor concentration in the reservoir increases. For situations where the heat pipe operation is periodic some fluid with a concentration corresponding to the condenser end point conditions will be added during each cycle. Each injection of fluid replaces an approximately equal volume expanded out of the reservoir a half cycle earlier. Such a repetitive process can be represented in the following way. Let m represent the ratio of reservoir volume to the volume of mixture injected from the condenser. After one expansion and recompression but before mixing the reservoir contains a fraction by volume of $1/m$ at the injected composition and $(m-1)/m$ at the previous composition. If cycling is continued the fraction at the injected composition after n cycles will be

$$S_n = 1/m[1 + (m-1/m) + (m-1/m)^2 + \ldots (m-1/m)^{n-1}] \tag{1}$$

This expression defines the set point drift path when the initial and final control temperatures are known. In addition, experimental data can be used to project subsequent set point behavior and determine final control temperatures.

The temperature history for the heat pipe evaporator and the condenser end point for a cycle provide the experimental data necessary to predict the final control temperature. They characterize the mean concentration of the injected fluid and the flow into the reservoir. The average working fluid vapor partial pressure is given by

$$\bar{p} = \int_o^{V_I} p(T_c)\mathrm{d}V \Big/ \int_o^{V_I} \mathrm{d}V \tag{2}$$

where $p(T_c)$ is the working fluid saturation pressure at the condenser end point temperature T_c and the denominator is the volume of mixture compressed into the reservoir. The incremental volume change $\mathrm{d}V$ is related to the evaporator temperature T_E by

$$\mathrm{d}V = V\mathrm{d}p(T_E)/p(T_E) \tag{3}$$

Care must be taken in defining the upper limit of integration, V_I. When a long capillary tube is used as a diffusion barrier some gas is retained in the tube both at the end of expansion and compression. This tends to reduce the quantity of gas exchanged. The total volume injected is thus

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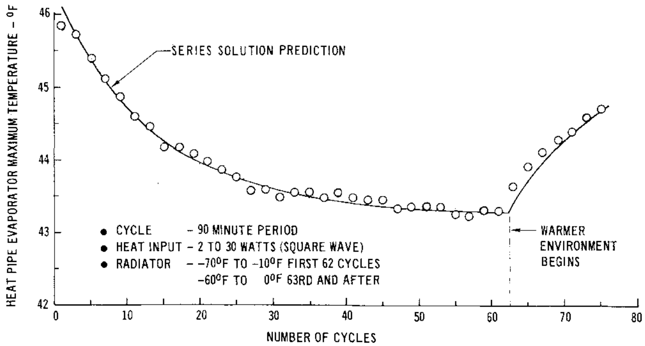

Fig. 1 Series solution—comparison with data.

$$V_I = \int_{T_{MIN}}^{T_{MAX}} \frac{V}{p(T_E)} \frac{\mathrm{d}p(T_E)}{\mathrm{d}T_E} \mathrm{d}T - V_{CAP} \tag{4}$$

where V_{CAP} includes all such null volume including the capillary volume, and T_{MAX} and T_{MIN} represent control temperature extremes during a single cycle. Equation (4) also shows that if V_{CAP} is large enough $V_I = 0$ and set point drift thus can be stopped.

The set point temperature change evolves from a consideration of the change in evaporator pressure associated with the change in partial pressure of the reservoir working fluid. For small changes the evaporator final temperature

$$T_{E_f} \simeq T(P_{R_f}) \tag{5}$$

where $T(P)$ is the saturation temperature at pressure P and the reservoir final pressure,

$$P_{R_f} = p(T_{E_i}) + (\bar{p} - P_{R_i}) \tag{6}$$

differs from the initial pressure by the amount the mean working fluid vapor pressure $\bar{p}$ differs from the original working fluid vapor concentration P_{R_i}.

Some data obtained by the author serve to illustrate the technique for an ammonia-helium variable conductance heat pipe. The control gas reservoir was originally charged with 16.0 psia ammonia vapor and 67.1 psia helium. The total pressure of 83.1 psia corresponds to an evaporator temperature of 46.3°F. After a cycle of operation, the average partial pressure of ammonia entering the reservoir during cycling was determined from experimental data using Eq. (2) to be 11.12 psia. This leads to $P_{R_f} = 78.2$ psia and an expected temperature reduction of about 3°F. For small changes approximating the temperature as a function of cycle gives

$$T_n \simeq T_i + (T_f - T_i) S_n \tag{7}$$

Data are compared with the calculations performed with the approximate methods in Fig. 1. Calculations for the warmer environment beginning with Orbit 62 required a second application of the procedure for the new conditions. Agreement as shown is very good.

This technique is particularly useful for consideration of set point drift when many identical cycles are performed. When cycle changes can be represented by a small number of distinct changes the process can also be applied successively. Another application is in system design studies where parametric analyses of off nominal performances are made. The impact of changes such as the control change caused by a change in radiator emissivity can then be made quickly to determine sensitivities. Where many parameters change simultaneously or repeatedly, however, the methodology of Ref. 6 may be necessary to provide a complete evaluation.

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