Thermal Control Systems for Spacecraft Instrumentation

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Thermal management for instrumentation on shuttle pallets or proposed space platforms requires a system capable of accepting or rejecting heat to various instruments in different physical locations. This paper presents a review of the current, two-phase pumping mechanisms being considered. Techniques for the determination of the required pumping power needed for the removal of a predetermined quantity of heat using a two-phase system are presented and evaluated along with a discussion of the flow regimes occurring in pumped two-phase fluids in micro-*g* environments. Six different concepts for achieving the required pumping power are presented and compared based upon factors such as power requirements, control, life expectancy, system complexity, and technology readiness.

Nomenclature

A = area
C_f = friction factor
d = pipe diameter
D = hydraulic diameter
G = mass flow rate per unit area
h = latent heat of vaporization
K = correction factor
$\dot{m}$ = mass flow rate
ΔP = pressure drop
$\dot{q}$ = rate of heat removal
s = liquid layer thickness
V = velocity
y = mass flow rate vapor fraction
η = pump efficiency
ρ = density
τ = shear stress
μ = viscosity
ϕ = void fraction

Subscripts

ℓ = liquid
v = vapor
ω = wall
z = longitudinal pipe axis

Introduction

THE principal methods of spacecraft thermal management employed since the Gemini program use pumped single phase systems. These conventional fluid loop systems, when combined with pumps (to provide fluid movement) and radiators (for heat rejection), have proven to be quite reliable for moderate heat loads with fairly constant heat rejection rates.[1] These types of systems, however, have two significant shortcomings.

First, the use of a fluid's sensible heat for thermal control results in a large temperature variation within the loop. This variation can be reduced by increasing the fluid flow rate, but this results in an increase in the required pumping power and, hence, in an increase in the total system weight. Second, these types of single phase systems are designed for a specific combination of instrumentation with a predetermined thermal load. Therefore, in order to accommodate even small changes in the heat load, significant design modifications would be required.

Future space platforms and shuttle pallets, similar to the one shown in Fig. 1, will require heat rejection systems that are capable of transporting heat from a widely diversified group of multidisciplinary instrumentation and maintaining it within a narrow isothermal bandwidth. This instrumentation will have a broad spectrum of heat loads, whose combined load will fluctuate as the usage varies. In addition, the physical location of the instruments themselves, which may vary by as much as fifty feet, may change over the life of the system.

The interactive effect of instrumentation with varying heat loads sharing a common liquid coolant loop can be reduced significantly through the use of a two-phase system. This type of system uses not only the fluid's sensible heat but also its latent heat in providing thermal control. As a result, two-phase systems are capable of removing much larger quantities of heat per unit mass of fluid. In addition, they are capable of maintaining a narrow isothermal bandwidth close to the saturation temperature with a minimal flow rate. This small flow rate not only reduces the total weight of the system; it also decreases the total pumping power required.

The total system, or "thermal utility" as it has come to be known, would be composed of three principal elements. The first is a group of interfacing devices or cold plates. The function of these devices is to collect excess heat from the various electronic components by providing a means for the vaporization of the working fluid. The second is a system referred to as a thermal bus, capable of transporting excess heat to the third element, the radiators, where it can be rejected. Work is currently being done on all three elements of the system. Preliminary work on both the cold plate and radiator design has been completed, and investigations into the thermal bus concept is presently underway.

The thermal bus, as it is currently configured, will serve as the transport system between the cold plates and the radiator. It will consist of a two-phase fluid loop and some type of pumping mechanism. The pump size will be determined by the power required to overcome the total frictional pressure drop in the system. As shown in Fig. 2, the fluid leaves the pump as

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a saturated liquid and proceeds to the cold plates. In the cold plates, the quality of the single phase fluid begins to change as it absorbs heat from the instrumentation and is vaporized. From this point the fluid continues through the remaining cold plates, undergoing phase changes as necessary, while the instruments are maintained at an almost constant temperature.

From the cold plates the liquid/vapor mixture travels to the radiators where it is condensed; the waste heat is rejected and the single phase liquid returns to the pumping device. Essentially, the entire system resembles the operation of a heat pipe, with the cold plates serving as the evaporator and the radiator as the condenser, except that in this type of system the liquid and vapor both flow in the same direction.

Almgren, et al.[2] have identified thirty-eight existing and proposed instruments for use as a data base, and, from this information, design criteria for a system of this type have been established. These criteria, shown in Table 1, can be used to determine the total heat rejection capabilities of the system and to assist in the determination of the power requirements.

Power Requirements

In the design and formulation of a two-phase heat removal system such as this one, it is necessary to understand the basic principles involved. First and foremost is the determination of the required mass flow rate, $\dot{m}$, necessary to accomplish the removal of a predetermined quantity of heat. The theoretical flow rate required for the removal of a specified quantity of heat from single or multiple sources is primarily a function of the fluid's latent heat of vaporization. Given the fluid properties and the quantity of heat that needs to be removed, the mass flow rate can be determined from

$$\dot{q} = \dot{m}_\ell h_{\ell v} \tag{1}$$

Table 1 Summary of the input design parameters for a thermal utility

Parameter	Design value range
System length	50 m end-to-end
Isothermal bandwidth	5°C
Thermal resistance at instrument interface	0.002°C
Radiator mass/area	5 kg/m^2
Cold plate mass/area	30 kg/m^2

where $h_{\ell v}$ is the latent heat of vaporization, $\dot{q}$ is the rate of heat removal, and $\dot{m}_\ell$ is the mass flow rate of the liquid (assuming that the fluid temperature is equal to the saturation temperature).

The energy transfer that occurs when a substance undergoes a change of state is very large when compared to the energy transfer occurring in conduction, convection, or radiation. For example, the vaporization of one kilogram of water at atmospheric pressure requires the transfer of approximately 2,200 kJ/kg of energy. Because of the magnitude of this energy transfer, the liquid mass flow rate required in this type of system is quite small. Hence, both the weight and the pumping power required are small. This pumping power, although small, is a major consideration in the design and selection of the overall system. The required pumping power can be determined through an energy balance.

$$\text{Pumping power} = \Delta P \dot{m}_\ell (1/\eta) \tag{2}$$

where ΔP is the total pressure drop in the system, $\dot{m}_\ell$ is the mass flow rate of the liquid, and η is the pump efficiency.

An equation that relates the required pumping power to the desired heat removal rate is obtained by combining Eqs. (1) and (2).

$$\text{Pumping power} = \frac{\dot{q}\Delta P}{\eta h_{\ell v}} \tag{3}$$

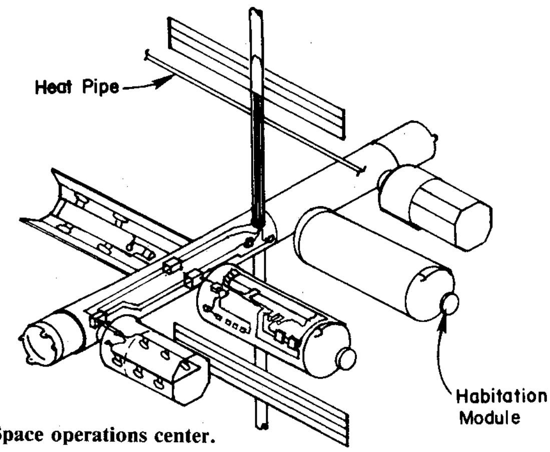

Fig. 1 Space operations center.

Table 2 Comparative summary of the pumping mechanisms

Criteria	Monogroove heat pipe	Capillary pump	Vapor compressor	Mechanical pump	Osmotic pump	Biomorph pump
External power required	None	None	Large	Small	Small	Small
Priming limitations	Severe	Severe	None	None	Minimal	Minimal
Control necessary	Unknown	Minimal	Considerable	Considerable[a] Moderate[b]	Unknown	Unknown
Accept and reject heat	No	No	Yes	Yes	Yes	Yes
Life expectancy	Unknown	Excellent	Fair	Good	Poor[c]	Poor[c]
Technology level	Unknown	Adequate	Good	Excellent	Poor[c]	Poor[c]
System complexity	Not feasible	Minimal	Complex	Fair	Complex[c]	Complex[c]

[a]Pump assisted. [b]Pumped two-phase. [c]At current state-of-the-art.

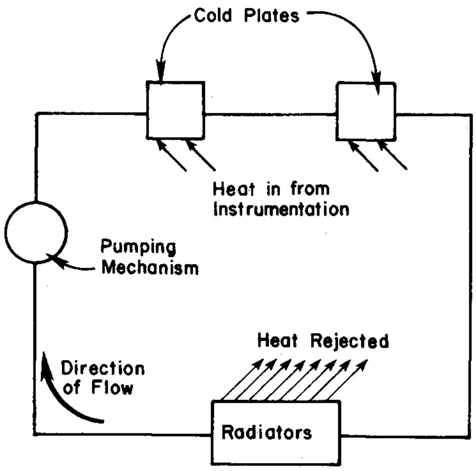

Fig. 2 Thermal utility schematic.

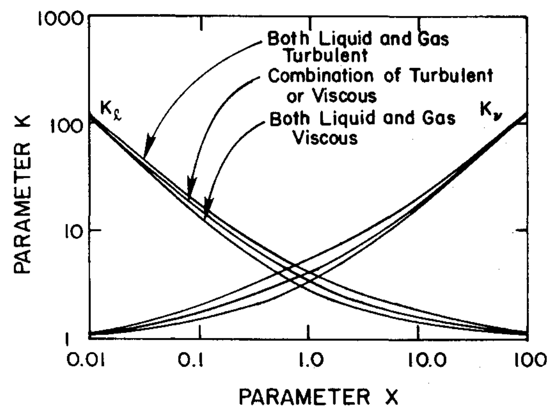

Fig. 3 Parameters for pressure drop in liquid and gas flow.[10]

Having found the required mass flow rate from Eq. (1), and using conventional data for the pump efficiency, only the pressure drop remains to be determined. Dukler et al.[3] have summarized and classified the various methods available for determining the pressure drop that results from the flow of a two-phase fluid through a circular duct. They have classified these into three categories: empirical correlations, equations of motion, and mathematical analysis of simplified physical models, with equations that relate the variables.

Of these methods, the most common are those utilizing empirical correlations. These can be further categorized into two types; those derived on the basis of horizontal flow,[4,5] and those resulting from vertical flow data.[6,7] The reliability of these relationships has been proven when they are used within the range over which the data were collected. However, when they are used beyond that range, the reliability is poor.[8] Levy[9] has presented a solution using the equations of motion, but the most accurate solutions are those resulting from the mathematical analysis of a simplified physical model.[10,11]

Knowledge of the pressure drop resulting from the phase interactions is based primarily on experimental evidence and is largely dependent upon the void fraction and hence the flow regime. At present, there is some question regarding the actual type of flow regime that will occur in a two-phase system in zero-g. In a one-g environment, it has been shown that as the fluid progresses through this series of cold plates, absorbing heat through evaporation or rejecting it through condensation, the mode of flow will change slowly from bubble flow to annular flow to slug flow and, if the heat load is high enough, to total vapor flow.[12] In a low-g or zero-g environment, however, the type of flow would not necessarily follow this

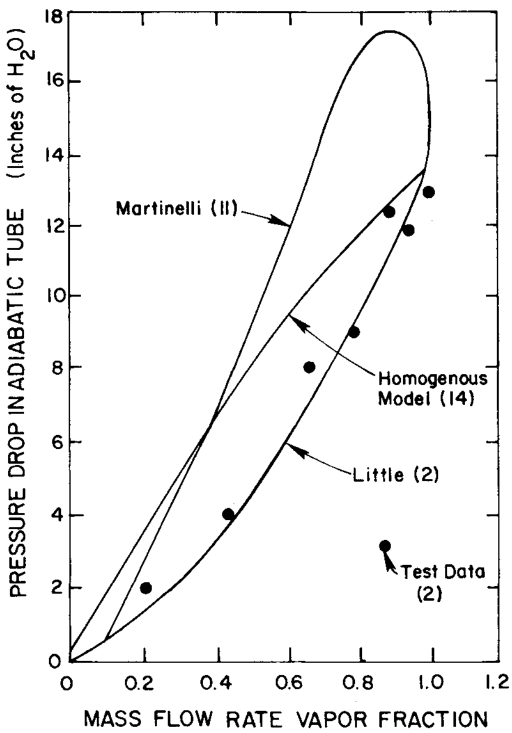

Fig. 4 Comparison of predicted and experimental pressure losses.[2]

pattern. In a micro-g environment, the surface tension, a force which is largely neglected in one-g, becomes one of the dominant forces, and, hence, in the absence of gravity, launch loads, or on orbit perturbations, annular flow is the dominant regime and the pressure drop can be most accurately predicted using one of the annular flow models.

Inertial forces through bends, etc. could result in changes in the flow pattern, but with the extremely small flow rate necessary in this type of system, these forces are small and therefore should not be expected to have a significant influence on the flow regime.

Recent work has examined, evaluated, and compared three different methods used to determine the total liquid pressure drop that occurs in pumped two-phase systems.[2] These methods include the homogeneous model, the Lockhart-Martinelli correlation, and a method that was developed under NASA contract. The first two are well known classical methods and have previously been described in the literature, while the third is discussed in detail in Ref. 2.

The first of these three methods, the homogeneous model, starts from the basic premise that a two-phase fluid can be replaced by a single phase compressible fluid. The assumption is made that, because of the dispersion of the vapor phase, the momentum and energy transfers are rapid enough that the time-averaged velocities and temperatures of the individual phases will be equal.[13] This assumption can be combined with the three balance equations, mass, momentum, and energy, which, for steady flow, yields

$$\frac{\mathrm{d}p}{\mathrm{d}z}=\frac{4\tau_\omega}{d}=\frac{2C_f\rho V_z^2}{d} \tag{4}$$

where τ_ω is the shear stress at the wall, d is the pipe diameter, and C_f is the friction factor and is defined as

$$C_f=\frac{2\tau_\omega}{\rho V_z^2} \tag{5}$$

Because of the initial assumption that the velocities and temperatures of the two phases are equal, an alternate method

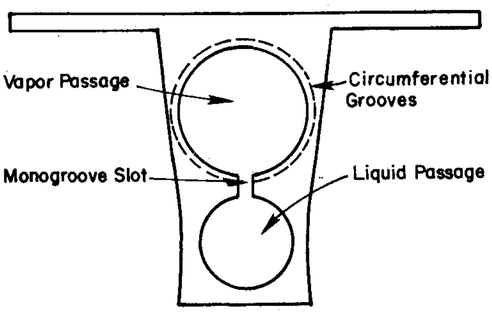

Fig. 5 Monogroove heat pipe.

must be used for cases such as counter-current flow or flows in which there are rapid variations in the flow parameters.

A second method used to determine the pressure gradient occurring in two-phase annular flow is one developed by Lockhart and Martinelli.[10] In this procedure, the total two-phase pressure gradient $(dp/dz)_{tp}$ is assumed to be a function of either the single-phase liquid $(dp/dz)_\ell$ or the single-phase vapor $(dp/dz)_v$ pressure gradient, multiplied by a correction factor K_ℓ for the liquid, or K_v for the vapor. These expressions can be written as

$$\left(\frac{dp}{dz}\right)_{tp} = K_\ell\left(\frac{dp}{dz}\right)_\ell \tag{6}$$

for the liquid, or

$$\left(\frac{dp}{dz}\right)_{tp} = K_v\left(\frac{dp}{dz}\right)_v \tag{7}$$

for the vapor, where

$$K_\ell = f_1(x) \qquad K_v = f_2(x) \tag{8}$$

and

$$x = \left[\left(\frac{dp}{dz}\right)_\ell \Big/ \left(\frac{dp}{dz}\right)_v\right]^{1/2} \tag{9}$$

The individual pressure gradients given in Eq. (9) are found in the normal manner using the Fanning equation, assuming that each phase is flowing alone in the pipe.[10] These velocities are based upon the full cross-sectional area of the pipe and are

$$V_\ell = \frac{\dot{m}_\ell}{\rho_\ell A} \tag{10}$$

and

$$V_v = \frac{\dot{m}_v}{\rho_v A} \tag{11}$$

where $\dot{m}_\ell$ is the mass flow rate of the liquid, $\dot{m}_v$ is the mass flow rate of the vapor, and A is the cross-sectional area of the pipe.

The values for K_ℓ and K_v can be found in Fig. 3. Note that different curves are required for each of the different flow combinations, i.e., both liquid and vapor in viscous (laminar) flow, etc.

Experimental work performed indicates that neither of these two methods adequately predicted the pressure drop occurring in pumped two-phase systems operating in a low-g or micro-g environments, thus leading to the development of a third method (presented in Ref. 2). This method starts from the momentum equation

$$\tau_{\ell v}A_{\ell v} - A\tau_{\ell\omega}A_{\ell\omega} = -\Delta P_v A_\ell \tag{12}$$

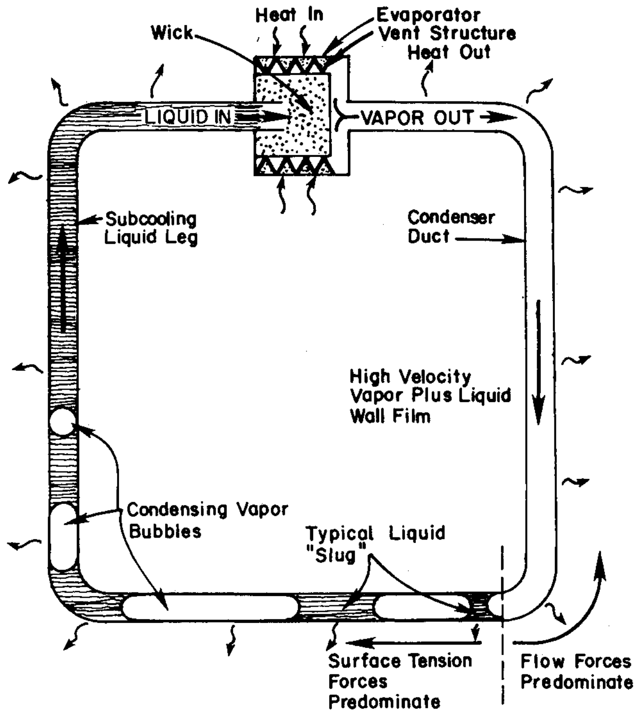

Fig. 6 Stenger's capillary pumped loop.[16]

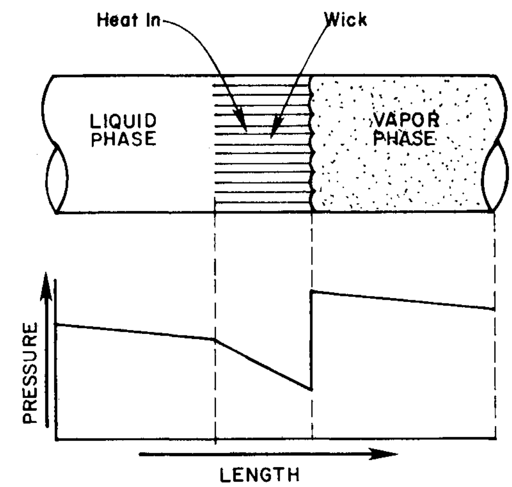

Fig. 7 Capillary pump.

where $\tau_{\ell v}$ is the shear stress at the liquid-vapor interface, $\tau_{\ell\omega}$ is the shear stress at the wall, $A_{\ell v}$ is the area of the liquid-vapor interface, $A_{\ell\omega}$ is the area of the liquid wall interface, A_ℓ is the cross-sectional area of the liquid, and ΔP_v is the total vapor pressure drop. Assuming that the velocity distribution in the liquid can be represented as a quadratic yields

$$V_\ell = \int_0^s V_{\ell y}\,dy \Big/ s = \frac{bs}{2} + \frac{cs^2}{3} \tag{13}$$

where s is the thickness of the liquid layer and both b and c are dummy variables. A further assumption that the velocity of the liquid is much smaller than the velocity of the vapor yields values for b and c of

$$b = \frac{C_{fv}\rho_v V_v^2}{2g_c\mu_\ell}\left(\frac{D_\ell}{D_v} + \frac{A_{\ell v}}{A_{\ell\omega}}\right) \tag{14}$$

and

$$c = -\frac{c_f \rho_v V_v^2}{4 g_c \mu_\ell s}\left(\frac{D_\ell}{D_v} - 1 + \frac{A_{\ell v}}{A_{\ell \omega}}\right) \tag{15}$$

where D_ℓ and D_v are the hydraulic diameters of the liquid and vapor spaces, respectively.

If the liquid velocity and vapor velocity are taken to be

$$V_\ell = \frac{C_{fv} \rho_v V_v^2 s}{2 g_c \mu_\ell}\left(\frac{1}{3}\frac{D_\ell}{D_v} + \frac{1}{3}\frac{A_{\ell v}}{A_{\ell \omega}} + \frac{1}{6}\right) \tag{16}$$

and

$$V_v = \frac{\dot{m} y}{\rho_v A \phi} = \frac{G y}{\rho_v \phi} \tag{17}$$

where ϕ is the void fraction, G is the mass flow rate per unit area, and all other symbols are as previously defined, then, by definition of the void factor and using an average value of s, a value of y, the mass flow rate vapor fraction, can be developed.

$$y = \frac{-1 + \sqrt{1 + (4BD/C)}}{2BD/C} \tag{18}$$

where

$$B = \frac{C_{fv} G}{2 g_c \mu_\ell}\left(\frac{1}{3}\frac{D_\ell}{D_v} + \frac{1}{3}\frac{A_{\ell v}}{A_{\ell \omega}} + \frac{1}{6}\right) \tag{19}$$

and

$$C = \frac{\rho_v \phi^2 d}{\rho_\ell (1 - \phi) s} \tag{20}$$

Having determined y, the mass flow rate vapor fraction, V_v can be found and substituted into the expression

$$\frac{\mathrm{d}p}{\mathrm{d}z} = 2 C_f \frac{Z}{D_v} \frac{\rho_v V_v^2}{g_c} \tag{21}$$

to find the frictional pressure drop.

The predicted pressure drop as determined using the three aforementioned techniques is shown in Fig. 4. The curves plotted were determined using Freon 11 as the working fluid, an overall tube length of 10 m, and a diameter of 0.0124 m.

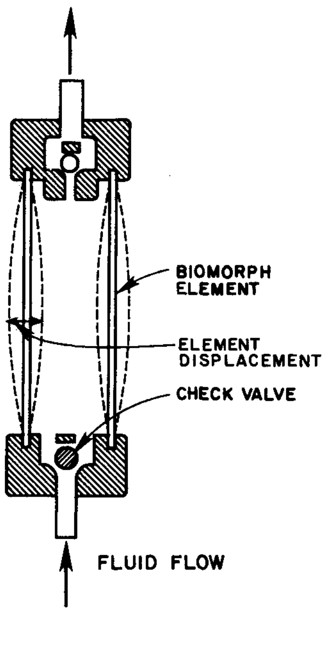

Fig. 8 Bimorph pump.

As shown in Fig. 4, the method developed in Ref. 2 more closely matches the experimental data. It has been shown that the accuracy of the homogeneous model increases as the pressure and velocity increase.[13] Therefore, it is logical that in a pumped two-phase system, where the pressure and velocity are low, this method would be inappropriate. The Lockhart and Martinelli method, since it is based largely on empirical techniques for unknown flow regimes, may also be inappropriate. Although the experimental results presented in Ref. 2 cannot be considered conclusive, there is a high degree of correlation between the modeling technique proposed and the experimental data shown in Fig. 4.

Once the fundamental questions—How much fluid is necessary to transport a specified quantity of heat? What is the pressure drop? How much power is needed to overcome this pressure drop?—have all been answered, it is necessary to determine how this pumping power should be achieved.

Pumping Mechanisms

A total of six separate pumping systems have been or currently are being considered under various NASA contracts. These include a monogroove heat pipe powered concept, a capillary pumped loop, a vapor compressor driven system, a mechanically pumped loop, a loop powered by an osmotic pump, and a loop with pressure provided by a bimorph pump. These systems have been evaluated previously and can be summarized as follows.[14]

Monogroove Heat Pipe

The developmental work described in Ref. 15 has produced a new heat pipe using what is referred to as a monogroove concept. As shown in Fig. 5, this heat pipe consists of two parallel channels—one for liquid and one for vapor. These two channels are connected by a small monogroove slot. The two separate channels allow the flow of the two phases to occur independently; and thereby reduces the viscous forces and the probability of liquid becoming entrained in the vapor. Preliminary tests indicate that this heat pipe is capable of transporting approximately 25,000 W-m (10^6-in.).

Initial conceptual design had the monogroove heat pipe providing both the heat transport and pumping function. This concept was dropped during preliminary screening due to the wall wicking limitations of the heat pipe in this system's thermal operating range.

Capillary Pump

In 1966, Stenger developed two capillary pumped loops using water as the working fluid.[16] These loops were designed, constructed, and operated successfully, demonstrating the possibility of a capillary pumped loop in one-g. The two loops ranged in power from 248 to 1,000 watts and over a temperature range varying from 100°C (212°F) to 144°C (291°F).

Stenger's original capillary pumped loop is shown in Fig. 6. The pumping power or pressure rise occurs across the highly curved liquid-vapor interfaces in the wicking pores as shown in Fig. 7. This pressure rise is a function of the liquid surface tension and wick pore diameter, and must exceed the total fluid friction pressure drop throughout the entire loop.

The capillary pump has several advantages. It is a totally passive system which requires no external power source to operate. It is somewhat self-regulating in that the fluid velocity is a function of the amount of heat that needs to be removed. Finally, its simple construction will allow a number of small individual loops to be constructed with very little weight penalty. The construction of these individual loops helps prevent total system failure in the event of a micrometeoroid puncture. However, one of the principal disadvantages of a capillary pumped system is that an increased conduit diameter is necessary to reduce the friction losses due to the small pressure gradient developed across the liquid vapor interface.

Ref. 2 describes the extensive work done on the capillary pumped concept and summarizes design parameters for both a centralized and a decentralized loop.

Vapor Compressor

Of the six pumping devices investigated, the compressor driven concept requires the greatest amount of external power.[14] This type of system differs from the others in one major way—the pump, or in this case the compressor location. In this system the compressor is located downstream from the cold plates and compresses the fluid just prior to its entrance to the radiator, thereby providing fluid to the radiators at maximum system pressure and hence maximum temperature.

Mechanical Pump

A dramatic increase in the heat transfer capacity of a passive two-phase loop can be obtained with a correspondingly small amount of mechanical pumping power due to the high latent heat of vaporization of the liquid, as shown in Eq. (1). Although this increase is significant, the addition of a mechanical pump results in an overall reduction in system reliability and an increase in total system weight. More important, however, is the need to take positive steps to provide active control of the fluid flow rate. This is true for all three of the following pumping systems. Mahefkey[17] reports that weight and power penalties associated with pumped loop systems represent a "serious weight problem for near future high power spacecraft." The largest portion of this weight penalty, however, occurs as a result of the large surface area required for the radiators; the additional weight of a mechanical pump represents a small fraction of the total increase in system weight.

Reference 2 describes a series of tests with a breadboard apparatus of the thermal bus using a mechanical pump. These tests demonstrated the thermal control concept. Although the instrumentation used did not provide a high degree of precision, it did provide a map of the system's performance capabilities. The results of these tests have demonstrated the capability of transporting 1 kW over a distance of 10 m with a total pressure drop of 20 in. of H_2O. By extrapolating this test data, a capacity-length of 25,000 W-m (10^6 W-in.) within the 5°C isothermal bandwidth required seems feasible.[16]

The performance of mechanical pumps used in previous programs has proven to be quite satisfactory, with flow rates ranging from 4 liters/min of water at 134 kPa in Space Lab to 14.1 liters/min of Freon 21 at 483 kPa in the current Orbiter.[17]

Osmotic Pump

The use of an osmotic pump to power the two-phase loop offers many of the same advantages as a mechanical pump. The disadvantages are also similar, with a few additions.

The operation of an osmotic pump results from a pressure differential across a semi-permeable membrane. On one side of the membrane is a solvent-solute mixture; on the other is a mixture of pure solvent. The pumping action occurs as a result of the osmotic forces across this membrane. The solvent is vaporized in the evaporators, leaving behind the solute. This vapor is then used as the working fluid. From the evaporators the vapor travels to the radiators where it is condensed. It then travels through the loop to the other side of the membrane.

Successful operation of the osmotic pump in a zero-*g* environment is dependent upon the development of a solution circulation technique that will direct the concentrated solution at the evaporator back toward the osmotic membrane, promoting an increase in the osmotic pressure. This is a function that is normally carried out by convection in one-*g*.

Fleischmann et al.[19] have developed a concept for maintaining a continuous solvent concentration through the use of a small electric potential across the membrane when an electrolytic solute such as aluminum sulfate is used.

Two major problems exist within the current state of technology in the osmotic pump concept. First, a method of control of the pumping rate must be developed since it is no longer a function of the heat input. Second, almost all of the testing that has been done uses water as the solvent and sucrose as the solute, with a cellulose acetate membrane. The low vapor pressure of water precludes operation at temperatures below 40°C (104°F), while the cellulose acetate membrane is incapable of sustained operation at temperatures greater than 75°C (167°F). Considerable advancements in materials must be made before this type of pump can be considered feasible. However, the pumping power appears to be adequate, and a recently developed prototype of an osmotic heat pipe with a cellulose acetate, spiral-wound membrane achieved an osmotic pressure rise of 2.8×10 N/m (400 psi).[20]

Bimorph Pump

The final concept being considered is that of a bimorph pump. The principal of operation is illustrated in Fig. 8. The pump itself is constructed of two layers of ferroelectric crystals or ceramic material that have been bonded together. The two elements are each separately attached to electrical leads. When a voltage potential is applied across the two leads, the piezoelectric effect results in a bending or flexing of the material. If an alternating current is applied, a repeated flexing occurs which, when combined with check valves etc., results in a pumping action. Narasaki[21] has constructed a pump that is capable of pumping approximately 225 cc/min at 60 Hertz. The flow and pump efficiency, which ranges from 4-10%, is very dependent upon the fluid viscosity. Control on this type of pump can be achieved through a variation in either the voltage or frequency. Unfortunately, this type of pump, like the osmotic pump, is still in the developmental stages and its reliability has not yet been determined.

Summary

It is difficult to compare the various pumping methods discussed without giving considerable attention to the total system in which a particular pumping mechanism exists. Table 2 summarizes some of the important criteria necessary to help determine the optimum pumping device for the thermal bus. Information for this comparison was compiled from the results of feasibility studies and total system concept evaluations, as performed by various contractors. As a result, no quantitative comparisons have been made. Both the bimorph and osmotic pumps have features which make them attractive, but neither of these two methods have been developed to the point where they could be considered reliable enough for use in a thermal utility as it is currently envisioned.

The mechanical pump and vapor compressor offer similar attractive characteristics, although the fundamental system configuration is different. Control of the two systems could be accomplished in approximately the same manner. The mechanical pump has an excellent performance history on previous vehicles and, because of this high degree of proven reliability and current technology readiness, it appears to be the logical choice.

The use of a capillary pump has two important characteristics that make it an attractive alternative. These are: 1) it provides a system that is passive and therefore requires little or no control; and 2) no extra power is required for normal operation. The selection of this type of pump, however, would require additional investigation of the priming capacities in zero-*g* determination of the effect of noncondensible gases on pump operation, and proof of the reliability of a test system in a zero-*g* environment.

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