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Two-Phase Cooling System with a Jet Pump for Spacecraft

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A two-phase cooling system with a jet pump is proposed for the thermal control of spacecraft. The system does not require an external source of energy, the pumping of the working fluid is performed by the work that is produced in a thermodynamic cycle being carried out in the heat transport loop. The cooling system has no moving parts or control devices, with the exception of a mechanical pump and an actuated valve, that are used only for the startup sequence. This article reports on the results of the theoretical and experimental studies of the two-phase thermal control system with a jet pump for spacecraft application. A mathematical model for the steady-state analysis of the proposed system was developed. The model was applied to predict overall performance characteristics and operating range for a specific spacecraft two-phase cooling system. The possible reasons for the fluid loop operation failures were identified. The influence of the accumulator volume on the system characteristics was also investigated. Three jet pumps of various configurations were tested and stable operational regimes of the cooling system were obtained under different heat-load and heat-rejection conditions on a ground experimental facility.

Nomenclature

A_n = flow area of two-phase nozzle
A_r = radiator area
A_s = total flow area of the cold liquid sprayers
A_t = flow area of throat
c_p = specific heat of the liquid phase
d = diameter
h = enthalpy
k_{mc} = mixing chamber velocity coefficient
k_n = nozzle velocity coefficient
L = length
M_Σ = total mass of the coolant in the loop
m = mass flow rate
p = pressure
p_n = design pressure at the exit of the nozzle
Q = heat load
T = temperature
V = velocity
V_a = accumulator volume
v = specific volume
x = quality
α = angle between the axis of the subcooled liquid sprayer and the axis of the mixing chamber
ε = absorptivity
μ = viscosity
ρ = density
σ = Stefan–Boltzmann constant
Φ^2_{lLM} = two-phase multiplier

Subscripts
f = liquid
fg = difference between liquid and vapor properties
g = vapor
in = inlet
m = two-phase mixture
out = outlet

Superscripts
$'$ = heat acquisition loop
$''$ = heat rejection loop

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Introduction

SPACECRAFT have previously used passive cooling or pumped liquid loops for temperature control of onboard instruments. Single-phase thermal control systems with a mechanical pump proved to be quite reliable for spacecraft with moderate heat load (10^2–10^3 W). The principal disadvantage of these single-phase systems is that fluid temperature is variable within the fluid loop. Temperature variations can be reduced by an increase in the fluid flow rate. However, this results in an increase in pump power, larger pipe diameters, and consequently, an increase in the weight of the entire system. A growth in heat load in future spacecraft, with a simultaneous increase of their overall dimensions, favors the development of a thermal control system that uses a boiling fluid as a working substance. In this kind of system the heat is transferred using the latent heat of evaporation of a two-phase coolant. It allows the transfer of more waste heat per unit of fluid flow rate than in the single phase systems, because the latent heat of evaporation is several orders of magnitude higher than the characteristic values of heat capacity of liquids. Therefore, the required mass flow rate of the two-phase working fluid, pump power, and hence, weight of the entire system can be less than for a single-phase loop. Moreover, the use of boiling heat transfer allows one to provide almost isothermal conditions for cooled equipment under variable heat loads and external heat flux. The heat transfer processes with the phase change have higher heat transfer coefficients than under conditions of convective heat transfer in liquids. This will allow a decrease in the dimensions and weight of heat exchangers. The above-mentioned advantages of the two-phase thermal control system over a single phase one become much more significant with an increase in the waste heat to be rejected.[1,2]

Two-phase thermal control systems have been intensively developed by NASA for the Freedom Space Station since the end of the 1970s. Several types of systems have been proposed, but the highest degree of development has been achieved in the mechanically and the capillary pumped two-phase loops.[3–5] Bland et al.[6] suggested the use of a two-phase loop with an original element, the rotary fluid management device that incorporates the pitot pump. Work on the development of two-phase thermal control systems has been started by the European Space Agency (ESA) for future large-scale space applications, such as Europe's Columbus and Polar Platform Projects.[7] The NPO "Energia" Scientific–Industrial Aerospace Enterprise (in Russia) has also been developing mechanically-pumped two-phase loops for the last 5 years in connection with the Universal Space Platform and the Mir 2 Space Station Projects.[8]

In this article a two-phase cooling system with a jet pump (injector-condenser) is proposed for the thermal control of future spacecraft. A similar system was suggested and tested for water-lifting by Alad'ev et al.[9] The system does not require an external energy source to pump the working fluid. Fluid circulation is carried out by the use of the waste heat in the same way as in the capillary-pumped systems. The jet pump is being used in many fields for different purposes because of its simple construction and easy operation. The pump has no moving parts and can be designed with minimum weight. In addition, it has a very good resistance to cavitation compared to other types of pumps. Thus, jet pumping may be an attractive method for waste heat transport in new generations of spacecraft. Many theoretical and experimental studies on single and two-phase jet pumps are reported in Refs. 10–16.

The major objective of this work is to investigate the possibility of an application of a two-phase cooling system with a jet pump for spacecraft thermal control. An earlier report on the work was presented in Refs. 17 and 18. Three two-phase jet pumps were designed, built, and tested. An experimental study on the entire system was performed on a ground test stand. An overall performance prediction technique was developed. Predictions of the system operation for application of the proposed thermal control technique in a spacecraft were made. Possible reasons for the fluid loop operation failures were identified, and the influence of the accumulator volume on the system characteristics was investigated. Stable operational regimes of the cooling system were obtained under different heat-load and heat-rejection conditions on the ground experimental facility.

Two-Phase Cooling System with a Jet Pump

A schematic of a two-phase cooling system with a jet pump is shown in Fig. 1. The system consists of a heat acquisition loop and a heat rejection loop. A liquid pumped by the jet pump splits into two paths downstream of the valve. The liquid passing through the cold plates is partially or totally vaporized and enters the two-phase nozzle (Laval nozzle) of the jet pump shown in Fig. 2. The fluid passing along the heat rejection loop is cooled at the radiator and is supplied to the subcooled liquid sprayers of the jet pump. The two-phase mixture accelerates in the expansion process at the nozzle under the action of the pressure difference between its inlet and outlet sections. A high-velocity mixture with high vapor content (droplet flow) mixes with the subcooled liquid entering the jet pump through the liquid sprayers (nozzles) in the conical mixing chamber. At this point, both flows exchange mass, momentum, and energy; as a result, the vapor is condensed. The complete condensation terminates in a two-phase "shock," which takes place in the throat of the jet pump, i.e., in the area of constant cross section between the mixing chamber and the diffuser, or in the expansion part of the diffuser. In the latter, a further rise in static pressure occurs, due to the slowdown of the flow. As a result, total pressure at the diffuser outlet exceeds its values at the inlet of the two-phase nozzle and of the subcooled liquid sprayers. This makes it possible to provide pumping of working fluid

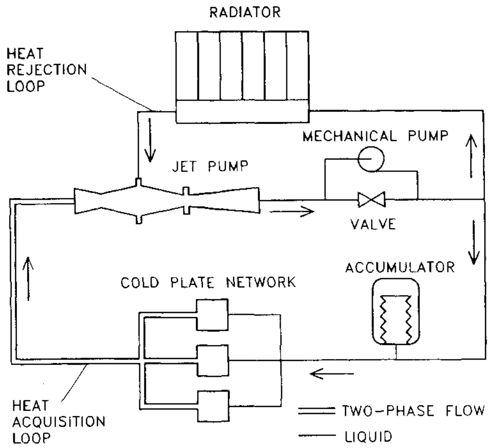

Fig. 1 Two-phase cooling system schematic.

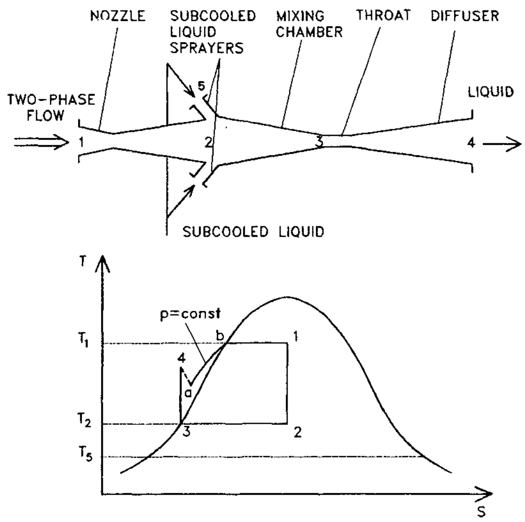

Fig. 2 Two-phase jet pump and the thermodynamic cycle being carried out at the heat acquisition loop.

along both loops of the cooling system. In order to start the jet pump it is necessary to create an initial flow of the working fluid as in any internal combustion engine. The mechanical pump can be used for this purpose, as shown in Fig. 1. The valve is initially closed. After the starting of the jet pump the mechanical pump is turned off and the valve is opened. The task of the accumulator is to control the system pressure and cold plate temperature by either adding or removing fluid from the system.

The working fluid undergoes cyclic changes in the heat acquisition loop so that a Rankine cycle is carried out. This is shown in Fig. 2 on the *T-s* plane. Process 1-2 is the adiabatic expansion of the vapor in the active nozzle of the jet pump. Process 2-3 is the isobaric vapor condensation in the mixing chamber due to mixing with the subcooled liquid. Thus, the subcooled liquid serves as a heat sink. Process 3-4 is the adiabatic pressure rise at the diffuser. Process 4-a is the pressure drop in the heat acquisition loop because of friction. And, finally, process a-b-1 is the isobaric heating of the subcooled liquid and its evaporation at the cold plates. As can be seen, the jet pump, together with the heat acquisition loop, is a thermal engine. A part of the heat to be rejected is converted into mechanical work, which is used for working fluid pumping along both loops. Because of frictional losses, the energy of the flow is dissipated as heat and is finally rejected by the radiator. Therefore, in the steady-state conditions, the amount of the heat transferred to the system always equals the amount of the rejected heat. The jet pump simultaneously performs the functions of a pump, a direct-contact heat exchanger, and a condenser. Unlike a mechanical pump, it is not sensitive to the presence of noncondensable gases. The main disadvantage of this thermal control technique compared with the other two-phase systems consists of a temperature difference (of the order of a few tens of degrees Kelvin) that has to be maintained between the heat source and the heat sink. This results in a lower radiation temperature and, consequently, a heavier radiator. But it should be pointed out that the two-phase working fluid is used only in the heat acquisition loop. Condensation occurs at the jet pump. In the situations in which the use of the condensing heat exchanger is not desirable, this is a positive feature of this heat transport loop. It is known that the condensation in a two-phase heat exchanger (condenser) in a closed fluid loop may be accompanied by noise and vibrations. The use of a two-phase jet pump avoids these undesirable phenomena. The proposed system does not require an external energy source. Components with moving parts are not used, with the exception of the starting pump and the valve. Therefore, a high reliability level and a long operation lifetime of the system are expected.

Modeling the System

A mathematical model of the two-phase cooling system with a jet pump was developed for predicting its overall performance characteristics. The standard SINDA modeling technique was not applied. Instead of breaking each component into many nodes and links, the simulation scheme for pumped two-phase thermal control systems developed by Ollendorf and Costello[19] was used. The mathematical modeling was performed in two stages: first, the models of the components were developed, and then they were united in a whole system model by means of the technique described below. During the modeling process, the following main assumptions were made. The system was considered at steady state, and homogeneous two-phase flow was assumed. The physical properties of the two-phase mixture were evaluated assuming a saturated system, i.e., liquid and vapor in equilibrium. Several other assumptions specific to the individual component models were also made. The component models were designed as independent modules (performance subroutines). Each model requires either the inlet or outlet conditions (pressure, quality, and mass flow rate), and a set of physical parameters (pipe diameter and length, area of a radiator, etc.), specific to the model. A brief description of the major functions and features of each component model, including some of the major equations, is given next.

Cold Plate Model

The pressure drop in the cold plate was assumed to be negligible with respect to the pressure drop in the pipelines. Knowing the inlet conditions and heat flux, the outlet parameters were determined with the energy balance equation, namely,

$$h_{\text{out}} = h_{\text{in}} + Q/m \tag{1}$$

Radiator Model

A simple mathematical model of a radiator was used. The temperature difference between the radiator wall and the fluid was ignored. The external heat flux was assumed to be zero for simplicity. Thus, the energy balance equation for the infinitesimal radiating surface area was written as follows:

$$mc_p \, \mathrm{d}T = -\varepsilon\sigma T^4 \, \mathrm{d}A_r \tag{2}$$

Eq. (2) has an analytical solution in the form of

$$T_{\text{out}} = [T_{\text{in}}^{-3} + 3\varepsilon\sigma A_r/(mc_p)]^{-1/3} \tag{3}$$

Eq. (3) expresses the dependence of outlet temperature on inlet temperature, heat capacity of the coolant, mass flow rate, radiating surface properties, and radiator area.

Pipe Model

The objective of the pipe model was to predict friction pressure drop along the pipe. The single-phase pressure drop was determined using the Darcy–Weisbach equation with the friction factor for a smooth tube wall. The two-phase pressure drop was determined based on the homogenous flow model[20]

$$p_{\text{in}} - p_{\text{out}} = \frac{\Phi^2_{l1M} L m^2}{(2\rho_f d A^2)} \tag{4}$$

where

$$\Phi^2_{l1M} = (1 + x v_{fg}/v_f)(1 + x\mu_{fg}/\mu_g)^{-1/4} \tag{5}$$

Jet Pump Model

The two-phase jet pump consists of the following elements: the nozzle, the cold liquid sprayers, the mixing chamber, the throat, and the diffuser. The equations of mass, momentum, and energy conservation in an integral form were applied to each of the elements of the jet pump.

A homogeneous equilibrium (isentropic-homogeneous expansion) model described by Starkman et al.[21] was employed to calculate the critical flow rate. The model was extended to take into account the influence of the back pressure on the flow parameters at the nozzle outlet. In case of underexpansion, the pressure at the nozzle outlet was set equal to the nozzle design pressure p_n. Since the mixing chamber pressure was lower than p_n in all the experiments, the overexpansion phenomenon in the nozzle was not considered. The relationships suggested by Badr et al.[22] were used to evaluate the Freon 113 properties.

The following assumptions were made in the mixing chamber model:

1) The process of condensation is isobaric, i.e., pressure is constant along the mixing chamber. The increase in the two-phase mixture density due to the vapor condensation is compensated by the decrease in the flow velocity.

2) Complete flow mixing takes place and an equilibrium two-phase mixture enters the throat.

The mass, momentum and energy balance equations for the mixing chamber are (see Fig. 2 for the nomenclature)

$$m_1 + m_5 = m_3 \tag{6}$$

$$\begin{aligned} -m_{m2}V_{m2} - m_{f2}V_{f2}\cos\alpha + m_3V_3 \\ = A_np_n + (A_s\cos\alpha - A_t)p_2 \end{aligned} \tag{7}$$

$$m_1(h_1 + V_1^2/2) + m_5(h_5 + V_5^2/2) = m_3(h_3 + V_3^2/2) \tag{8}$$

Bernoulli's equation was used to relate the inlet and outlet parameters of the cold liquid sprayers

$$p_5 - p_2 = \tfrac{1}{2}(\rho_f V_{f2}^2) - \tfrac{1}{2}(\rho_f V_5^2) \tag{9}$$

Applying Eq. (9) to the diffuser yields

$$p_4 - p_3 = \tfrac{1}{2}(\rho_f V_3^2) - \tfrac{1}{2}(\rho_f V_4^2) \tag{10}$$

In order to account for the friction in the elements of the jet pump, the calculated ideal (isentropic) velocity at the outlet of each element was multiplied by a corresponding velocity coefficient. The velocity coefficient for the nozzle was calculated using the equation suggested by Tsiklauri et al.[23]:

$$k_n = 0.96 - 0.115\log(v_{g1}/v_{f1})(1 - x_1) \tag{11}$$

Values for the nozzle velocity coefficients and for the diffuser pressure rise coefficients have been established by various authors and are summarized in the form of the loss coefficients in Ref. 10. In this study for k_n and k_d, values of 0.95 and 0.80, respectively, were used. To take into account frictional losses in the mixing chamber, a mixing chamber velocity coefficient k_{mc} was used. It was defined as the ratio of the actual velocity at the mixing chamber outlet to the outlet velocity for isentropic flow calculated for the same inlet state by Eqs. (6–8). The value of k_{mc} was determined on the basis of the experimental data and depended on the flow parameters, geometric configurations of the mixing chamber, and the subcooled liquid sprayers. For the jet pumps tested in this study, values of k_{mc} from 0.45 to 0.60 were found.

The calculation process was carried out in the following sequence. In the general case, at least five parameters were necessary to define steady-state flow in the jet pump: two parameters at each inlet section and one at the outlet section, e.g., T_1, x_1, m_5, T_5, and p_4. The first guess in p_2 was made and mass flow rates m_1 and m_5 were computed. Then, m_3 and V_3 were calculated by Eqs. (6) and (7). Next, v_3 was determined by the continuity equation

$$v_3 = V_3 A_t / m_3 \tag{12}$$

Having computed v_3, x_3 and h_3 were determined using the equations of state for the two-phase mixture $x = f(v, p)$ and $h_m = f(p, x)$. If the energy balance Eq. (8) did not agree, p_2 was reassumed. This procedure was repeated until the energy balance was satisfied with an acceptable accuracy (10^{-4}). Next, the location of the shock wave in the throat or diffuser was determined on the basis of the given back pressure p_4 and the relationships for the normal shock wave, assuming that the complete condensation occurs in the shock wave. Finally, the other flow parameters at the jet pump inlets and outlet were calculated.

Network Algorithm

The model of the cooling system was broken down into submodels of the jet pump, and the heat acquisition and rejection loops at the places of their junctions. The submodels of both loops were constructed by the incorporation of the component models with the help of the conservation law equations for thermal/hydraulic networks (generalized Kirchoff's laws). At the break points the values of the flow parameters, calculated with the models of the jet pump and associated loop, must be equal. For example, the temperature T_1 that is a parameter of the jet pump must be equal to the temperature T_1', determined by the heat acquisition loop submodel. Thus, the mathematical model of the cooling system has been reduced to the following system of nonlinear equations:

$$T_5 - T_5'(T_1, x_1, m_5, T_5) = 0 \tag{13}$$

$$x_1 - x_1'(T_1, x_1, m_5, T_5) = 0 \tag{14}$$

$$P_5'(T_1, x_1, m_5, T_5) - P_5''(T_1, x_1, m_5, T_5) = 0 \tag{15}$$

$$m_1 - m_1'(T_1, x_1, m_5, T_5) = 0 \tag{16}$$

$$M_\Sigma(T_1, x_1, m_5, T_5, m_1) = \text{const} \tag{17}$$

Eq. (17) was used for the closure of the equation set (13–16). The heat transport loop is a closed thermodynamic system, therefore, in order to define a system steady state besides the heat load and the heat sink conditions, it is necessary to give the total mass of the working fluid. The Newton–Raphson method was used for solving Eqs. (13–17).

Experimental Facility and Program

The purposes of the experimental study were 1) demonstration of the system operation under different heat-load and heat sink conditions; 2) determination of the experimental jet pump performance characteristics and operating limits; 3) to obtain the experimental data required for the jet pump mathematical model; and 4) implementation of the startup of the jet pump under different initial conditions.

A simplified schematic of the experimental facility is shown in Fig. 3. The facility primary components were the jet pump, the electrically heated evaporator, the heat exchanger, the valves, and the accumulator. During the startup process a circulation of the working fluid was accomplished by the mechanical pump. Downstream of the mechanical pump, the liquid flow splitted into two branches. One of the two flows passed through the evaporator and entered the jet pump nozzle. The other part of the liquid was cooled in the heat exchanger and entered the jet pump through the subcooled liquid sprayers. Heat transport from the heat exchanger to the refrigeration unit was accomplished by an intermediate liquid loop. Both flows merged at the mixing chamber of the jet pump and returned to the mechanical pump. The accumulator accommodated system liquid inventory changes associated with the heat load variation. A soft membrane separated the accumulator into two parts: one part was communicated to the fluid loop, the other was filled with air. The moment of start was determined by a sudden drop in the mixing chamber pressure. Then, the loop was transferred to the working regime, in which the pumping of the coolant was carried out without any additional external energy source. For this purpose the following operations were accomplished consecutively: the bypass line was closed by valve no. 2, valve no. 1 was opened and the mechanical pump was turned off. After that, the working fluid was pumped by the jet pump for as long as the heat load was applied and rejected. Running time for each steady-state operating regime was not less than 60 min. The bypass line was used in the first experiments in order to facilitate the startup of jet pumps of early designs. For the automatic startup of the loop a control unit that performed the above-mentioned functions was developed. The main components of the experimental facility were made of stainless steel. The experimental loop was equipped with a standard set of thermophysical measurement devices and a data acquisition unit.

The dimensions of the jet pumps tested are presented in Fig. 4 and in Table 1.

The subcooled liquid sprayers were conical holes arranged in a staggered order on the circumference of an iron ring in two rows. The outlet diameter d and number n of the sprayers are given in Table 1. Freon 113 was used as the working fluid. The basic criterion for the working fluid choice was the pressure rise in the jet pump, calculated for the design heat load and temperature difference between the heat source and heat sink. Value for the coolant freezing temperature was also taken into account. Based on the jet pump model predictions for the pressure rise for Freon 11, 12, 21, 22, 113, 114, 216, water, and ammonia were made. This study showed that Freon

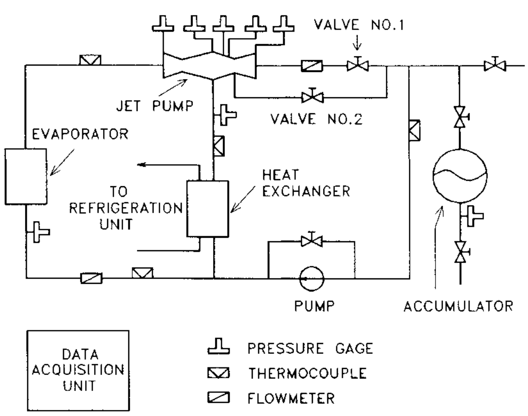

Fig. 3 Experimental facility schematic.

Table 1 Dimensions of jet pumps (mm)

Jet pump	d_1	d_2	d_3	d_4	d_5	d_6	L_1	L_2	L_3	L_4	L_5	Sprayers d	Sprayers n	Sprayers α, deg
IC-EU-2	15	7.7	8.4	14	3.8	12	20	90	75	60	15	0.75	24	50
IC-57-60	32	14	15.9	23	6.6	20	80	191	57	96	20	1.08	60	50
IC-76-60	32	14	15.9	23	6.6	20	80	191	76	96	20	1.08	60	50

Uncertainty in d_1, d_3, d_4, d_6 = ± 0.2 mm, in d_2, d_5 = ± 0.1 mm, in d = ± 0.05 mm, in L_1, L_2, L_3, L_4, L_5 = ± 0.2 mm, and in α = ± 1 deg.

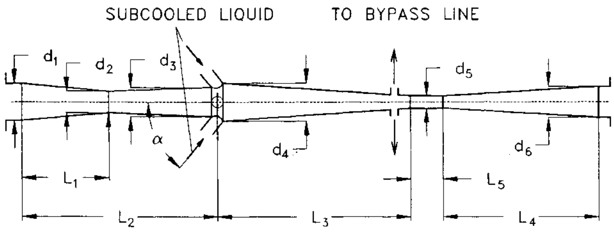

Fig. 4 Configuration and dimensions of the test jet pumps (see also Table 1).

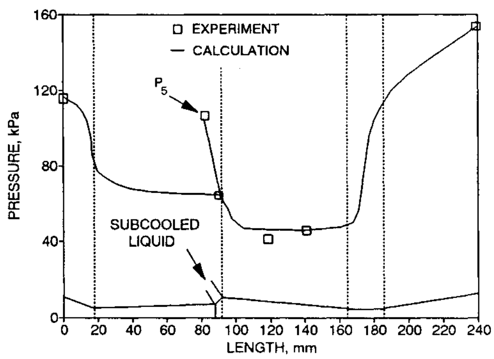

Fig. 5 Static pressure profile for the IC-EU-2 jet pump (m_4 = 0.144 kg/s, φ = 2.36, x_1 = 0.36, T_1 = 324.8 K, T_4 = 300.3 K, T_5 = 264.2 K).

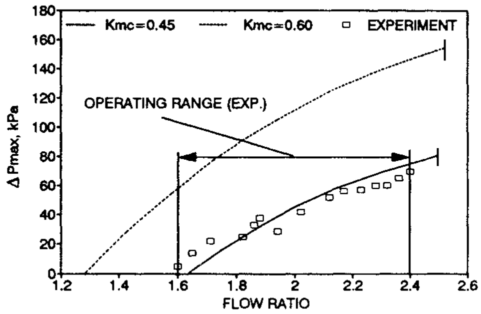

Fig. 6 Experimental and theoretical maximum pressure rise vs flow ratio (T_1 = 325.4 K, T_5 = 265.6 K, x_1 = 0.34, Q = 3.84 kW).

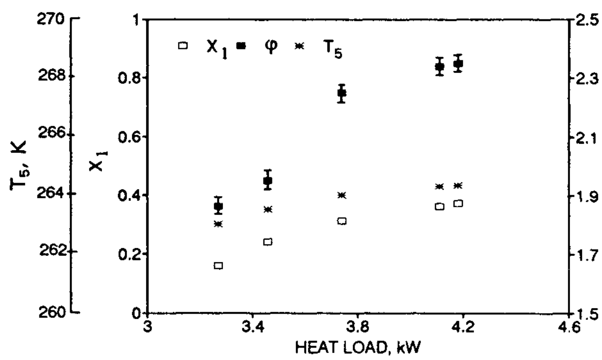

Fig. 7 Experimental overall performance characteristics of the two-phase cooling system (T_1 = 322.4 K).

113 and 114 were the most suitable working fluids for the considered spacecraft system application.

Temperatures were measured with a maximum uncertainty of ± 0.2°C. An estimate of the accuracy of the pressure was $\pm 4.5\%$. The voltage was measured by a digital multimeter with a maximum uncertainty of $\pm 1.3\%$. Current was measured by an a-c ammeter with a maximum uncertainty of $\pm 1.3\%$. This procedure resulted in a maximum uncertainty of $\pm 2.9\%$ in the calculated power inputs. The volumetric flow rate was determined by an in-line turbine flowmeter, which had an uncertainty of $\pm 0.5\%$. The thermophysical properties of Freon 113 were assigned an uncertainty of $\pm 3.0\%$, based on the observed variations in the reported values in Ref. 22. These uncertainties resulted in a maximum uncertainty of the quality measurements of about $\pm 14\%$, using the method of Kline and McClintock.[24]

Results and Discussion

A typical distribution of static pressure along the length of the IC-EU-2 jet pump is shown in Fig. 5. It was observed that the back pressure p_4 below a certain value did not affect the conditions at the jet pump inlet sections (sections 1 and 5 in Fig. 2). This can be explained by a flow-choking phenomenon at the throat. The changes in p_4 only resulted in a change in the location of the shock wave in the diffuser. The back pressure altered the conditions at the throat when the shock wave reached the mixing chamber. We considered that the jet pump worked when the conditions: $p_4 > p_5$ and $p_4 > p_1$, were satisfied. The maximum pressure rise produced by the jet pump $\Delta p_{\max}$ was defined as the difference between the back pressure, corresponding to the location of the shock wave at the throat inlet $p_{4\max}$ and pressure at the nozzle inlet p_1. Figure 6 shows the measured and calculated $\Delta p_{\max}$ as a function of flow ratio $\varphi = m_5/m_1$ (see also Fig. 2). The experimental jet pump performance was obtained by the variations of p_5 maintaining p_1, x_1, and T_5 at a constant level. For each value of p_5, the back pressure $p_{4\max}$ was determined by the variation of the valve flow area (valve no. 1 in Fig. 3). The results presented in Fig. 6 show that predicted $\Delta p_{\max}$ is very sensitive to k_{mc}. It was also found that the jet pump had three different causes that limited its operating range:

1) Too high back pressure: any point above the performance curve in Fig. 6. In this case, the shock wave entered the mixing chamber. The flow became subsonic at the mixing chamber outlet and p_4 altered the conditions of the throat and of the mixing chamber. This caused an increase in pressure in the mixing chamber. The two-phase flow velocity at the nozzle outlet decreased. As a result, the kinetic energy of the power flow was not sufficient to reach the required total pressure at the diffuser outlet.

2) Insufficient mass flow rate or too high a temperature of the subcooled liquid: the left limit of the performance curve in Fig. 6. In this case p_4 was lower than p_1. The condensation of vapor did not occur at the jet pump and the two-phase flow was at the diffuser outlet.

3) Excessive mass flow rate of the subcooled liquid: the right limit of the performance curve in Fig. 6. In this case, p_4 was lower than p_5. High value of m_5 caused the complete

condensation of the vapor at the mixing chamber, i.e., the liquid entered the throat. The back pressure p_4 affected the mixing chamber conditions and resulted in an increase in p_2. The two-phase mixture did not achieve the required velocity at the nozzle outlet.

The highest values of Δp_{max} achieved in the experiments with the IC-EU-2, IC-76-60, and IC-57-60 jet pumps were 65, 80, and 100 kPa, respectively. The last two jet pumps initially had the 40 subcooled liquid sprayers of 1 mm in diameter. An increase in the number of the sprayer up to 60 permitted the startup without the bypass line and led to about a 10% increase in Δp_{max}.

Several series of tests were performed to obtain the operating characteristics for the two-phase cooling system with the jet pump. Figure 7 summarizes the results of these tests for the IC-EU-2 jet pump. The cooling system was operated successfully at the heat loads of 3.3–4.2 kW. An increase in Q resulted in an increase in x_1 and in a decrease in the critical flow rate through the nozzle, i.e., m_1. The position of valve no. 1 was fixed. The system failed because of an insufficient m_5 at low Q and of an excessive m_5 at high Q. The objective of another series of tests was to establish the minimal possible temperature difference between the heat source and heat sink $\Delta T = T_1 - T_5$ for a given heat load. To determine minimal ΔT, the heat sink temperature was increased, maintaining the system pressure and Q at a constant level. The results of these experiments are shown in Fig. 8. The system failure at ΔT = 59 K occurred because of a low m_5 and high T_5 to condense the vapor in the jet pump. The minimum value of ΔT of 45 K was obtained in the experiments with the IC-57-60 jet pump.

A theoretical study was performed to predict the performance characteristics for a spacecraft two-phase cooling system with a jet pump with a design heat load of 7 kW. The relevant dimensions for the components and basic parameters of the system are listed in Table 2. The predicted performance characteristics of the fluid loop are shown in Fig. 9. An increase in Q results in an increase of radiator temperature, and therefore in an increase in the temperatures of all parts of the system. The cold plate outlet quality also increases. Working fluid moves from the system to the accumulator. The system pressure and boiling temperature of the coolant at the cold plates T_1 increase. The predicted operating range of the loop is near 2 kW.

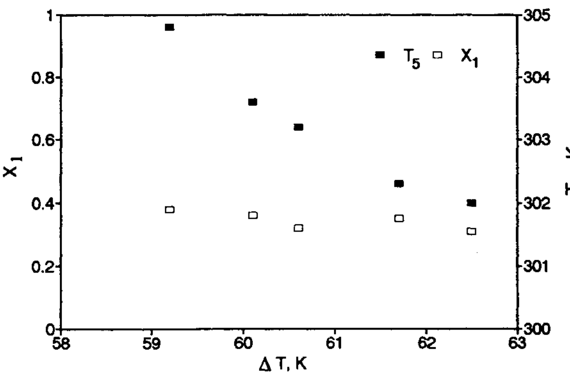

Fig. 8 Measured subcooled liquid temperature and quality at the evaporator outlet for different heat sink conditions (Q = 3.3 kW, T_1 = 323.2 K).

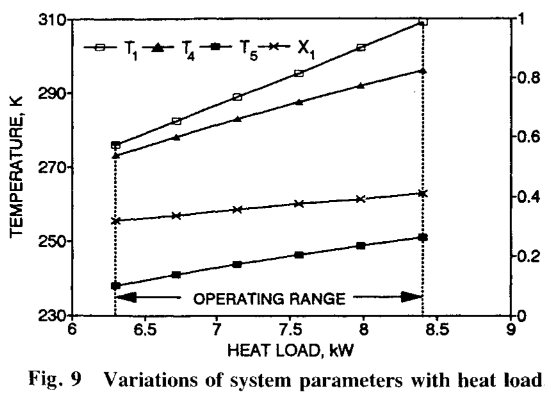

Fig. 9 Variations of system parameters with heat load.

Table 2 System sizes and system parameters

Working fluid	Freon 113
Design heat load, kW	7
Radiator area, m^2	28
Total length of pipes, m	100
Pipe diameter, m	
Liquid	0.008
Two-phase	0.012
Volume of the accumulator, m^3	24×10^{-3}
Gas at the accumulator	Nitrogen

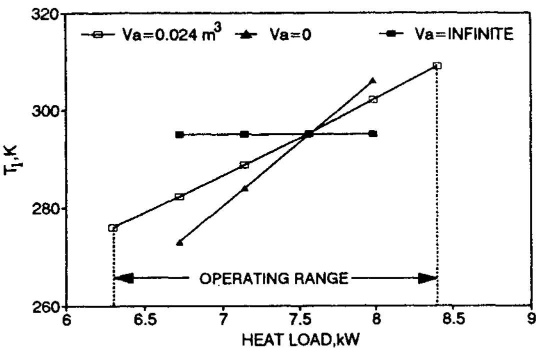

Fig. 10 Influence of the accumulator volume on the system performance characteristics.

Figure 10 illustrates the influence of the accumulator volume on the system performance characteristics and operating range. The results are shown in the $Q - T_1$ coordinate plane. In the case of infinite volume accumulator, the system pressure is constant. The constant saturation pressure provides a constant T_1 that is independent of Q. The predicted operating range is about 1.3 kW. In the case of a zero volume accumulator, the system volume is constant and T_1 strongly depends on Q. The calculations also show that x_1 decreases with Q in this case. This can be explained by the fact that the increase in the temperature of a two-phase mixture in a closed system of a constant volume results in a decrease in the quality when the specific volume of the two-phase mixture is less than the critical specific volume. The operating range for a system with an accumulator of final volume is larger than for a fluid loop having an infinite volume accumulator. The cold plate temperature is variable; nevertheless for most of the spacecraft applications these temperature variations are acceptable.

Conclusions

A two-phase cooling system with a jet pump is proposed for the thermal control of spacecraft. The loop does not require an external source of energy: the pumping of fluid is performed by utilization of the waste heat dissipated. The lack of moving parts makes it more attractive for multiyear missions compared with the mechanically pumped loop. Substantially reduced noise and vibration levels make this type of heat transfer system particularly suitable for applications where undisturbed microgravity conditions are required. The results of the theoretical and experimental studies presented in this article show the feasibility of the proposed thermal control technique. We also believe that the cooling technique may find an application for the cooling of nuclear power plants during an accident connected with a failure of the basic pumps.

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