Communications Technology Satellite: A Variable Conductance Heat Pipe Application

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A variable conductance heat pipe system has been designed to provide thermal control for a transmitter experiment package (TEP) to be flown on the Communications Technology Satellite. The variable conductance heat pipe system (VCHPS) provides for heat rejection during TEP operation and minimizes the heat leak during power-down operations. The VCHPS described features a unique method of aiding priming of arterial heat pipes and a novel approach to balancing heat pipe loads by staggering their control ranges. This paper describes the CTS variable conductance heat pipe system; discusses the system design parameters; and presents the results of the heat pipe subsystem and system level test programs.

Introduction

A VARIABLE conductance heat pipe system has been designed to provide thermal control of a transmitter experiment package (TEP) to be flown on the Communications Technology Satellite (CTS). The CTS is a joint U.S.-Canadian Government project (Fig. 1) launched from the Eastern Test Range into a near equatorial, synchronous orbit.

The variable conductance heat pipe system (VCHPS) provides for heat rejection from the TEP during operation and minimizes the heat leak during power-down conditions. The VCHPS was selected because additional radiator area was required to handle system load and the radiator had to be thermally decoupled during the power-down condition to prevent equipment temperatures from falling below acceptable levels. The VCHPS is designed to reject 196 w at maximum temperature (full-on condition) and less than 3 w during power-down condition. A system heat rejection of 196 w is achieved for a total system weight of 16.2 lb.

The use of variable conductance heat pipes on CTS represents one of the first times that gas loaded heat pipes will have been used in the primary thermal control system of a spacecraft. Considerable work has been reported on VCHPS[1-5] but most of it has been developmental, involving prototype hardware. A previous spacecraft which carried a variable conductance heat pipe system was the Orbiting Astronomical Observatory (OAO-C). However, the heat pipe system was flown as a functional experiment and, as such, was not totally relied on as the primary thermal control mechanism. Similarly, the forthcoming flight of the Advanced Thermal Control Flight Experiment (ATFE) on the ATS-F satellite [3] includes a VCHPS, but again the VCHPS is an experiment and its success is not critical to the spacecraft's primary mission. Although the CTS heat pipes are on the spacecraft in a functional capacity, they will be instrumented to provide data on their performance in orbit.

The CTS VCHPS features the use of a unique method of ensuring priming of arterial heat pipes. This method is described here to the extent applicable to the VCHPS. The fundamentals of the priming mechanism are described more fully in Ref. 6.

Finally, the CTS VCHPS introduces the concept of load balancing, a design technique whereby the degree to which the total load is shared by each heat pipe is established by proper selection of the heat pipe gas inventories.

Heat Pipe Description

The heat pipe selected for use on the CTS is an arterial, gas loaded heat pipe using methanol as the working fluid. Methanol was selected as a result of the need for a low

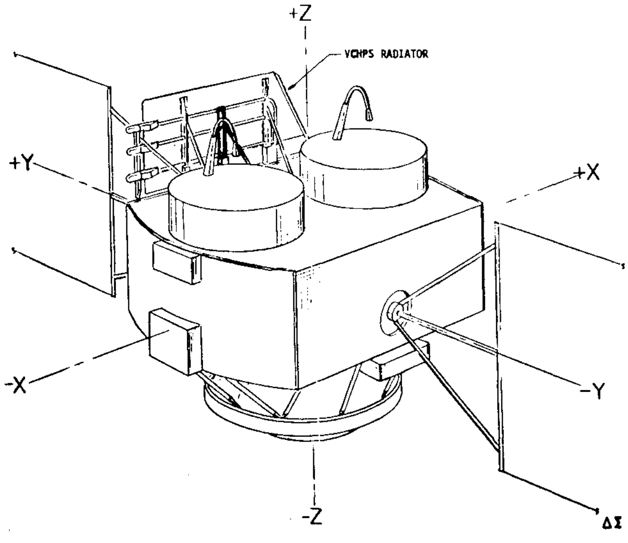

Fig. 1 CTS spacecraft.

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freezing point (-144°F) and a low operating pressure. A low operating pressure minimizes the possibility of the arteries depriming as a consequence of pressure fluctuations within the pipe. This phenomenon has been experienced in higher pressure systems such as ammonia.[5]

The control gas is 90% nitrogen and 10% helium. Nitrogen was selected as the control gas to provide a close molecular weight match with the methanol vapor to minimize stratification effects during one-g operation. The helium is introduced to allow the heat pipes to be leak checked during fill and closure operations.

A cross section of the heat pipe is shown in Fig. 2 and the component materials are shown in Table 1. The tube and wick are 304 stainless steel, while the arteries are 316 stainless steel. Extensive life testing shows both materials to be compatible with the methanol working fluid.

The arteries are attached to the homogeneous wick, as shown in Fig. 2. The unique feature of the artery is the priming foil shown in Fig. 3. This priming foil, attached to the artery at the evaporator end of the heat pipe, contains a pattern of 0.010-in. holes. These holes allow gases trapped in the artery to be vented as the fluid in the artery flows toward the evaporator during initial heat pipe start-up. When the gas bubble reaches the priming foil, menisci coalescence causes the fluid bridging the 0.010-in. holes to be removed, allowing the entrapped gas to vent.[6]

The CTS heat pipe utilizes a variable temperature, cold wicked reservoir for heat pipe control. The cold wicked reservoir was selected over a heated, nonwicked reservoir because the control range requirement $T_{max} \leq 122°F$; $T_{min} \geq 50°F$ allows the use of the simpler passive control system. The reservoir was sized to maintain the evaporator temperature nominally between 85°F and 120°F using the flat front theory.[2] The tradeoff between reservoir to condenser volume ratio and minimum evaporator temperature is shown in Fig. 4. Since the preliminary system analyses indicated the minimum sink temperature to be -150°F, a reservoir to condenser volume ratio of 1.5 was selected for reservoir sizing. The volume ratio on the final design varies from 1.5 to 2.0, since it is desireable to fabricate only one size of reservoir and the condenser size varies on each of the three heat pipes. The actual control ranges and required gas inventories were calculated with a more comprehensive computer program, as discussed later.

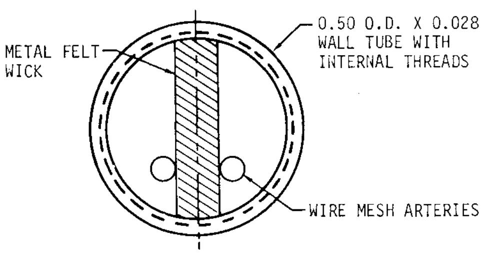


Fig. 2 CTS heat pipe.

Table 1 Heat pipe design details.

Tubes	304 stainless steel, 0.500-in. o.d. × 0.028-in. wall, internally threaded with 100 tpi, 0.005 in. deep, 40° included angle grooves
Reservoirs	304 stainless steel, spun hemispherical cap with 1.75-in. o.d. cylindrical center section. Reservoir to condenser volume ratio varies from 1.5 to 2.0
Wicks	Reservoir: 304 stainless steel metal felt, 0.020 in. thick, spot welded to interior walls Tube: 304 stainless steel metal felt, 0.050 in. thick, interference fit across diameter of tube Arteries: 150 mesh 316 stainless steel screen formed and welded to 0.063-in. i.d. tubes and spot welded to diametral wick Priming foils: 0.0005-in. thick 304 stainless steel foil with 0.010-in. holes, formed and welded to 0.063-in. i.d. tubes and spot welded to ends of arteries and diametral wick
Saddles	6061 aluminum alloy extrusion soldered to tubes
Working fluid	Methanol, spectrophotometric grade
Control gas	90% nitrogen, 10% helium, research grade

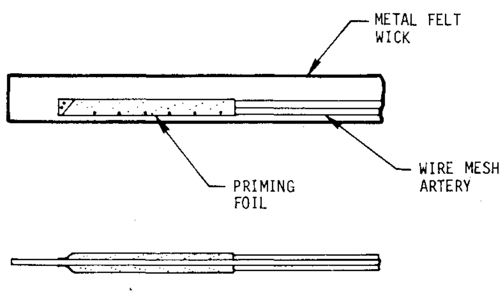


Fig. 3 CTS heat pipe wick assembly.

VCHPS Description

The VCHPS consists of three heat pipes connecting an equipment mounting plate to a space radiator (Fig. 5). The heat pipes are attached to the equipment plate by a common saddle and to the radiator by individual saddles. The location of the heat pipes on the radiator was optimized with respect to

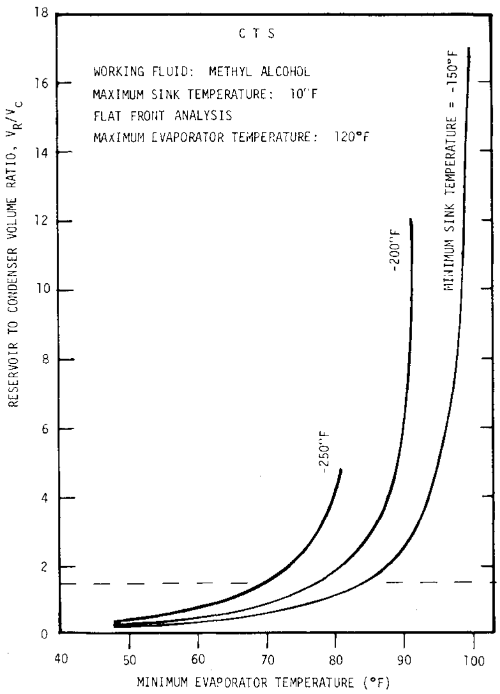


Fig. 4 CTS reservoir sizing tradeoffs.

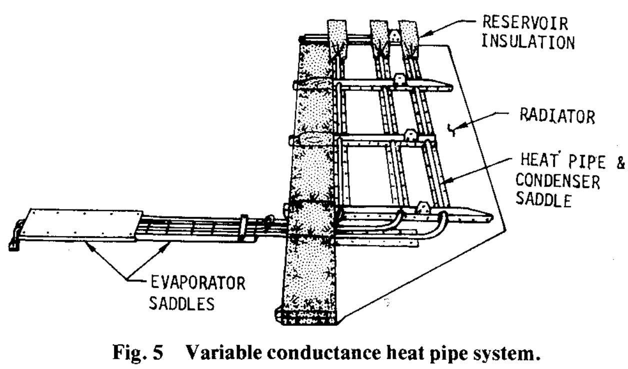


Fig. 5 Variable conductance heat pipe system.

radiator weight and heat rejection capacity with any one of the three pipes nonoperational. The heat pipes and saddles are soldered together prior to filling the heat pipes and all mechanically attached interfaces use RTV560 as a thermal interface material. The VCHPS mounts to the south panel of the spacecraft with the radiator surfaces facing north and south (Fig. 1).

The radiator is an 0.040-in. sheet of 6061-T6 aluminum coated with second-surface silvered Teflon thermal control tape. The initial solar absorptance to emissivity ratio is 0.1 ($\alpha_s/\epsilon = 0.08/0.8$) and is expected to degrade to approximately 0.25 after two years in the synchronous orbit environment. The silvered Teflon is applied to the radiator using a hot-vacuum bag application to assure adequate out-gassing of the adhesive.

The radiator is sized to provide heat rejection of 196 w under worst-case design conditions (one heat pipe failed and maximum insolation with end-of-mission optical properties) at an evaporator saddle temperature of 122°F. The mechanical attachment of the radiator to the spacecraft is accomplished with the use of fiberglass structural members to minimize the conductive heat leak when the heat pipes are shut off. The total conductive heat leak from the spacecraft to the radiator (heat pipes and structural attachments) is less than 3 w.

Design Considerations

The design of the heat pipe system is based on satisfying a number of criteria. The design criterion most influential on the individual heat pipe control range was a desire to balance the thermal load on each pipe. Because of the physical geometry imposed on the system design, the outboard pipe will tend to carry a disproportionate share of the total load since its radiator effectiveness is the highest. This is an undesirable situation, as the outboard pipe's load carrying capability is the least due to its longer length. To prevent this situation from occurring, the control gas inventory in each pipe is adjusted to stagger the vapor fronts so as to have each pipe carry a load more proportionate to its capacity. A SINDA thermal model[2] was used to determine the gas inventories required to balance the load. The predicted vapor front locations for each of the heat pipes resulting from the design gas inventories at design load are shown in Fig. 6.

The SINDA thermal model was used to predict the thermal load in each heat pipe for various pipe failure modes at the design gas inventories. The predicted heat pipe loads for each failure mode are compared with predicted pipe capacities in Table 2.

As the temperature of the evaporator section decreases, the total pressure in the heat pipe decreases. This drop in total pressure causes the control gas in the reservoir to expand and fill an increasing length of the condenser until the condenser is completely filled with the control gas (and some vapor at a partial pressure dictated by the condenser temperature). At this point the heat pipes are shut-off and cannot transport thermal energy except by conduction along the heat pipe. Since the gas inventory is different in each pipe, the heat pipes shut-off (or turn-on) at different temperatures. The widest control range is from an evaporator vapor temperature of 70°F for minimum sink conditions to 118°F full-on at maximum sink conditions. The difference between actual control range and that exhibited in Fig. 4 is a result of several factors: excess fluid residing in the reservoir, control gas inventory variance, actual reservoir to condenser volume ratio and the actual diffuse vapor front vs the flat front theory aspects of Fig. 4.

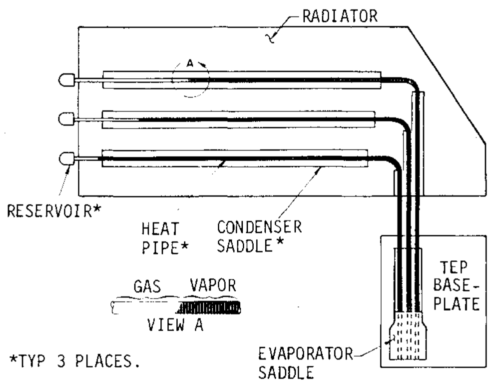


Fig. 6 Vapor front positions.

Table 2 CTS variable conductance heat pipe system performance

Conditions	System performance[a] load (w)	Pipe no.	Load (w)	Capacity (w)	Remarks
1) Outboard pipe (No. 3) failed	196	1	80	166	Full on
		2	116	156	Full on
2) Middle pipe (No. 2) failed	208	1	87	166	Full on
		3	121	146	
3) Inboard pipe (No. 1) failed	198	2	100	156	Full on
		3	98	150	
4) All pipes operating	240	1	73	166	Full on
		2	75	160	
		3	92	170	

[a] At a 122°F saddle under zero-g conditions.

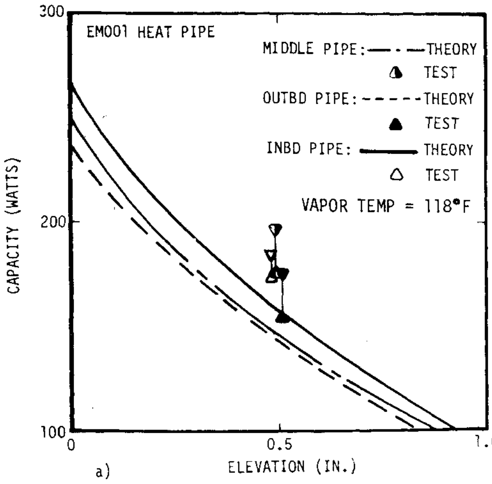
Test Program

The VCHPS thermal test program consists of individual heat pipe performance tests to verify heat pipe wick assembly integrity and a system verification test to verify thermal radiator design adequacy.

Since the heat pipes are assembled into a common evaporator saddle, the heater power applied to the saddle during testing will be distributed among the three heat pipes according to the control gas inventory (assuming equal heat sink effectiveness of the three heat pipe condensers). Thus, to test an individual heat pipe, it is necessary to heat the reservoirs of the remaining two heat pipes to expand the control gas such that it fills the condensers and maintains the vapor front within the evaporator region. This blockage of the evaporator with control gas prevents the remaining two pipes from transporting thermal energy to the cold plate and ensures that the pipe being tested transports all the applied energy.

The results of the individual heat pipe tests are compared with theoretical one-g predictions for each of the pipes in Fig. 7. The tests are conducted by applying a power level to the heater which is less than the predicted capability of the heat pipe. The power is then increased in small increments allowing the system to equilibrate after each power increment until heat pipe evaporator dry-out begins. The data shown in Fig. 7 a (engineering model 001) and Fig. 7b (engineering model 002) are the last power transported and the subsequent power increment that caused the arteries to deprime (evaporator dryout). As shown, the measured performance agrees well with theoretical one-g predictions, lending confidence to the calculated zero-g capacities shown in Table 2.

The performance of the priming foils was verified by priming tests conducted on the engineering model 001 heat pipes. Each of the heat pipes was repeatedly primed from a temperature of 20°F down to -130°F. The results of the priming test demonstrated that the priming foils greatly



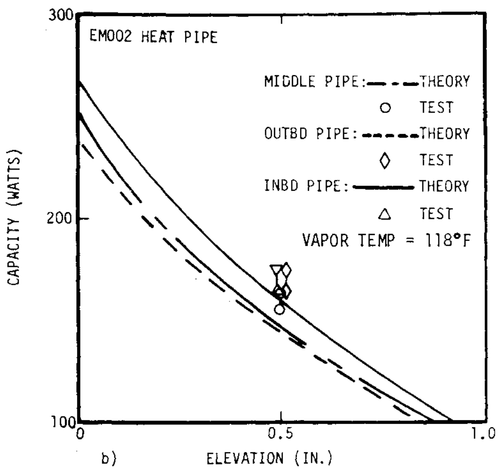


Fig. 7 Performance test results

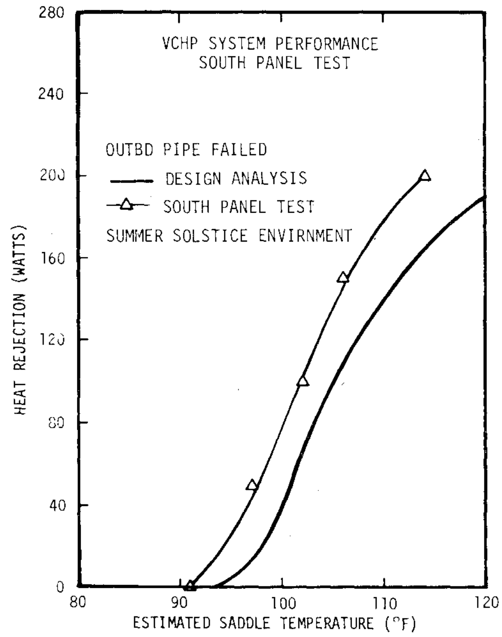


Fig. 8 VCHP system performance test.

enhance the priming capability of gas-loaded arterial heat pipes using methanol as a working fluid.

The system level VCHPS was tested at the NASA Lewis Research Center as part of the spacecraft south panel thermal vacuum test. The VCHPS system was elevated such that the evaporator was 0.375 in. above the reservoir to simulate zero-g operation. This test was conducted using a heat pipe system radiator coared with Kapton encapsulated strip heaters to simulate the insolation and absorbed infrared energy from areas of the spacecraft not present in the test. In addition, the area not covered by the heaters was coated with Kapton tape to provide a surface emittance close to that of the flight radiator.

The test results for the one heat pipe failed case are shown in Fig. 8, along with the results of the system characteristic from the design analyses. The differences between the two characteristic curves are due to the differences between the configuration of the system tested and the flight model. The difference in evaporator saddle temperature for a given heat rejection is primarily due to the difference in radiator surface emittance between the test ($\epsilon = 0.86$) and the design analysis ($\epsilon = 0.80$). Analyses are being conducted by the Canadian Communications Research Center (CRC) comparing the south panel thermal vacuum test results with the total spacecraft thermal control system design analyses. The results of the comparison will identify the degree to which the performance differences shown in Fig. 8 are the result of the test article configuration. However, at this point it appears that the VCHPS meets the heat rejection requirements of ≥ 196 w at an evaporator saddle temperature of 122°F or greater and a heat leak no greater than 3 w under heat pipe shut-off conditions.

Conclusions

A variable conductance heat pipe system was designed as a thermal control element for a transmitter experiment package on the CTS spacecraft. The heat pipe/radiator system provides a minimum of 196 w heat rejection in the "on" mode at an evaporator saddle temperature of ≥ 122°F and a maximum of 3 w leakage in the "off" mode for an evaporator saddle temperature of ≤ 70°F. This performance is achieved with any one of the three heat pipes nonoperational. Subsystem and system level tests verified that the individual heat pipes (VCHP's) and the heat pipe/radiator system (VCHPS) meet or exceed the design requirement.

The stainless steel-methanol heat pipes utilize a two-artery/slab wick system with menisci-coalescence priming foils to aid in venting the arteries of entrapped gas bubbles. Measured capacities of 160 $\pm$ 5 to 195 $\pm$ 10 w (7900-9800 w-in.) at 118°F and ½ in. evaporator elevation agreed closely with theory.

The heat pipe gas inventories were established to stagger the heat pipe turn-on temperatures and more nearly balance the total heat rejection load between the operational pipes (any two or all three).

Worst case system level tests indicated a 196-w rejection capacity at a saddle temperature of only 114°F, as opposed to the 122°F design point. However, it is believed that much of this excess rejection capacity was due to deviations in test conditions from those assumed for flight and used in the design analyses.

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