NOTICE

THIS DOCUMENT HAS BEEN REPRODUCED FROM MICROFICHE. ALTHOUGH IT IS RECOGNIZED THAT CERTAIN PORTIONS ARE ILLEGIBLE, IT IS BEING RELEASED IN THE INTEREST OF MAKING AVAILABLE AS MUCH INFORMATION AS POSSIBLE
DEPRIMING OF ARTERIAL HEAT PIPES: AN INVESTIGATION OF CTS THERMAL EXCURSIONS

214 p HC A10/MF A01 CSCL 20D 63/34 28773

FINAL REPORT
AUGUST 20, 1980

D. Antoniuk
D. K. Edwards

TRW SALES NO. 34129.000
CONTRACT NAS 3-21740

Prepared for
NASA-LEWIS RESEARCH CENTER
CLEVELAND, OHIO, 44135
This final report describes the analyses and experimentation performed to determine the cause(s) and/or contributing factor(s) responsible for four (4) thermal excursions of the Transmitter Experiment Package (TEP) on the Communications Technology Satellite (CTS) during its eclipse season in 1977. These so-called "anomalies" were the result of the depriming of the arteries in all three (3) heat pipes in the Variable Conductance Heat Pipe System (VCHPS) which cooled the TEP. The determined cause of the depriming of the heat pipes was the formation of bubbles of the nitrogen/helium control gas mixture in the arteries during the thaw portion of a freeze/thaw cycle of the inactive region of the condenser section of the heat pipe. Conditions such as suction freezeout or heat pipe turn-on, which moved these bubbles into the active region of the heat pipe, contributed to the depriming mechanism. Methods for precluding, or reducing the probability of, this type of failure mechanism in future applications of arterial heat pipes are also included in the report.
## CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0 INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>1.1 CTS Thermal Anomalies</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Review of Previous Work</td>
<td>5</td>
</tr>
<tr>
<td>1.3 Phase I Studies</td>
<td>10</td>
</tr>
<tr>
<td>1.4 Objectives of Phase II Studies</td>
<td>23</td>
</tr>
<tr>
<td>2.0 ANALYSIS OF BUBBLE LIFETIMES IN HEAT PIPE ARTERIES</td>
<td>25</td>
</tr>
<tr>
<td>2.1 The Spherical Bubble Model</td>
<td>25</td>
</tr>
<tr>
<td>2.2 Governing Equations for the Spherical Bubble</td>
<td>27</td>
</tr>
<tr>
<td>2.3 Transformation and Numerical Solution</td>
<td>29</td>
</tr>
<tr>
<td>2.4 The Elongated Bubble Model</td>
<td>32</td>
</tr>
<tr>
<td>2.5 Properties Used in Calculations</td>
<td>33</td>
</tr>
<tr>
<td>2.6 Results for the Spherical Bubble</td>
<td>33</td>
</tr>
<tr>
<td>2.7 Results for the Elongated Bubble</td>
<td>40</td>
</tr>
<tr>
<td>2.8 Significance of Analytical/Numerical Results</td>
<td>46</td>
</tr>
<tr>
<td>3.0 CYCLIC FREEZE-THAW TESTS ON SNO09 HEAT PIPE</td>
<td>48</td>
</tr>
<tr>
<td>3.1 Test Objectives</td>
<td>48</td>
</tr>
<tr>
<td>3.2 Apparatus and Procedure</td>
<td>48</td>
</tr>
<tr>
<td>3.3 Test Results</td>
<td>54</td>
</tr>
<tr>
<td>3.4 Significance of Test Results</td>
<td>72</td>
</tr>
<tr>
<td>4.0 CONCEPTS TO AVERT ARTERY DEPRIMING</td>
<td>75</td>
</tr>
<tr>
<td>4.1 Noncondensible Control Gas Selection</td>
<td>75</td>
</tr>
<tr>
<td>4.2 Operational Procedure for Start-Up of a Frozen Condenser</td>
<td>76</td>
</tr>
<tr>
<td>4.3 Mechanical Modifications</td>
<td>77</td>
</tr>
<tr>
<td>5.0 CONCLUSIONS</td>
<td>79</td>
</tr>
<tr>
<td>6.0 REFERENCES</td>
<td>80</td>
</tr>
</tbody>
</table>

### APPENDICES

<table>
<thead>
<tr>
<th>Appendix</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>A. PHASE I STUDIES</td>
<td></td>
</tr>
<tr>
<td>A.1 Excess Skew in Load Partitioning</td>
<td>A-1</td>
</tr>
<tr>
<td>A.2 Sudden Cooling Mechanisms</td>
<td></td>
</tr>
<tr>
<td>A.2.1 Instantaneous Heat Flow After Heat Pipe Reservoir Eclipse</td>
<td>A-4</td>
</tr>
</tbody>
</table>

---

iii
## CONTENTS (Continued)

<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>A.2.2</td>
<td>Instantaneous Heat Flow After Heat Pipe Radiator Eclipse</td>
<td>A-11</td>
</tr>
<tr>
<td>A.2.3</td>
<td>Condenser and/or Reservoir Shadowing Tests on SNO09 Heat Pipe</td>
<td>A-19</td>
</tr>
<tr>
<td>A.3</td>
<td>Freezing Blowby Tests on SNO09 Heat Pipe</td>
<td>A-21</td>
</tr>
<tr>
<td>A.4</td>
<td>TEP Thermal Analysis</td>
<td></td>
</tr>
<tr>
<td>A.4.1</td>
<td>Brief Review of LeRC Thermal Studies</td>
<td>A-23</td>
</tr>
<tr>
<td>A.4.2</td>
<td>Generalized Variable Conductance Heat Pipe Modeling</td>
<td>A-26</td>
</tr>
<tr>
<td>A.5</td>
<td>Bubble Studies</td>
<td></td>
</tr>
<tr>
<td>A.5.1</td>
<td>Potential for Bubble Formation in CTS Heat Pipes: Fundamentals</td>
<td>A-80</td>
</tr>
<tr>
<td>A.5.2</td>
<td>Potential for Bubble Formation in CTS Heat Pipes: Sample Calculations</td>
<td>A-86</td>
</tr>
<tr>
<td>A.5.3</td>
<td>Bubble Nucleation Experiments</td>
<td>A-95</td>
</tr>
<tr>
<td>A.5.4</td>
<td>Glass Heat Pipe Bubble Nucleation/Migration Experiments</td>
<td>A-99</td>
</tr>
<tr>
<td>B.</td>
<td>PHASE II STUDIES</td>
<td></td>
</tr>
<tr>
<td>B.1</td>
<td>Spherical Bubble Model Computer Program Listing</td>
<td>B-1</td>
</tr>
<tr>
<td>B.2</td>
<td>SNO09 Heat Pipe Cyclic Test Data</td>
<td>B-17</td>
</tr>
</tbody>
</table>
1.0 INTRODUCTIONS

1.1 CTS THERMAL ANOMALIES

The Communication Technology Satellite (CTS) was launched into an equatorial geosynchronous orbit in January, 1976 and is stationed at 116° longitude. After completing nearly four years of operation, the CTS mission has recently been terminated.

The major payload on the CTS is the Transmitter Experiment Package (TEP) which consists of a high power travelling wave tube (TWT), a power processing system (PPS) and a variable conductance heat pipe system (VCHPS) shown schematically in Figure 1-1. Heat produced in the tube collector is directly radiated to space, whereas power dissipated in the PPS is first conducted to the TEP saddle and the spacecraft south panel from where it is radiatively rejected.

The temperature control of the TWT is provided primarily by a variable conductance heat pipe (VCHP)/Radiator system which transports and rejects most of the heat dissipated in the tube body. Mounting details of the tube body to heat pipe evaporator saddle and the spacecraft south panel are sketched in Figure 1-2. As shown there are several interfaces through which the heat must flow to reach the heat pipes. Normally, heat reaching the aluminum baseplate is mostly absorbed by the heat pipes and is transported axially to the VCHP radiator. A fraction of it, however, is conducted to the spacecraft south panel where it is radiated to space.

The VCHP system, shown in Figure 1-3, consists of three stainless steel/methanol dual artery heat pipes whose variable conductance is achieved with the help of external cold wicked reservoirs loaded with a 10 percent helium/90 percent nitrogen gas mixture. Figure 1-3 shows the six positions in the VCHP/radiator system at which temperatures are measured in flight. The Transmitter Experiment Package is instrumented with four additional temperature sensors, two located on the Multistage Depressed Collector (MDC) and two sensors located on and near the tube body.

The CTS is the first spacecraft to rely on heat pipes to control the thermal performance of a major onboard system. The Transmitter Experiment Package has performed satisfactorily except on four occasions in 1977, March 16, March 23, April 11, and September 10 when measured tube body
Figure 1-1. CTS Transmitter Experiment Package
Figure 1-2. Mounted CTS OST
temperatures displayed sudden, rapid increases not normal for the TEP operating conditions and inconsistent with the design of the thermal control systems. The anomalous behavior of the TEP in these four occasions are referred to as the "CTS Thermal Anomalies" of days 75, 82, 101, and 253 respectively.

During all the anomalies, the continuous increase of the tube body temperatures was reversed by reducing the output power without damage to TEP. Operation of the TWT at the reduced power resulted subsequently in the recovery of the thermal control system with no evidence of degradation or changes in its performance.

After the fourth anomaly (day 253), the TEP was successfully operated at somewhat reduced power (below the open artery capacity of the heat pipes), and no further anomalies were detected.

1.2 REVIEW OF PREVIOUS WORK

Following the first occurrence (day 75), the CTS anomalies were the subject of investigations both at NASA Lewis Research Center (LeRC) and at TRW in an effort to understand the cause of the thermal failures, focussing particular attention on the VCHPS since it is in the thermal path from the tube body to space. The work performed at LeRC and preliminary studies conducted at TRW are documented in References 1 and 2, respectively.

LeRC's investigation included the use of flight-type TEP components in ambient and vacuum ground tests, analytical studies, flight data review, and CTS on-orbit tests. The objectives were to determine the most probable cause of the anomalies and to identify procedures or TEP operating modes which would preclude damage to the TWT in the event of recurring anomalies or of continuous degraded capacity of the thermal control system.

From the review of flight data for both normal and anomalous days the following observations about the anomalies were made:

1) All occurred after relatively long periods at constant, although differing, TWT RF power levels or heat rejection rates from the tube baseplate.

2) All the anomalies took place near the Spring and Fall equinox periods during which sun angle with respect to the VCHPS radiator is relatively small and during the orbit quadrant where the CTS is almost directly between Earth and Sun in which case the VCHPS surface becomes increasingly shadowed by the spacecraft itself.
3) All the anomalies, except on day 253, were characterized by being preceded by temperatures measured at the extremity of heat pipe number 1 (HP1 with sensor HPT5. See Figure 1-3) being at or below -98 C, the methanol freezing point.

4) All the thermal anomalies were preceded or accompanied by a sudden abnormal but small increase in the difference in measured temperatures between the adiabatic sections of heat pipe number 3 and 1, \( \Delta T_{1-3} \). This increase in \( \Delta T_{1-3} \) was followed by a second increase, and at the outset of unstable tube body temperature rise, \( \Delta T_{3-1} \) was observed to drop to a lower level.

Tests were performed with a TWT almost identical to the one outboard CTS. The results indicated that the anomalies could not have been caused by thermal interface failure inasmuch that: a) it is improbable that a failed interface could recover its integrity completely after an anomaly, and b) it was demonstrated that an interface failure results in tube body temperature rates higher than those observed during the anomalies.

Vacuum tests that most closely approximated anomalous temperature profiles were those in which the tube base plate cooling was abruptly reduced from that required for thermal equilibrium at normal operating temperatures. These test results suggested that the reduction in heat transfer capability of the variable conductance heat pipe system is, to a high degree of certainty, the cause of the thermal anomalies.

Room-ambient tests at LeRC with a flight-type VCHPS indicated that \( \Delta T_{3-1} \) increases are caused by depriming of heat pipe 1 (HP1) or heat pipe (HP1) and heat pipe 2 (HP2). The temperature difference between the adiabatic sections of heat pipe 3 (HP3) and HP1 was found to depend on the priming states of the heat pipe arteries. The measured temperature differences \( \Delta T_{3-1} \) were:

a) \( \Delta T_{3-1} \approx 1.1 \text{C}, \) all heat pipes are primed

b) \( \Delta T_{3-1} \approx 5.1 \text{C}, \) HP1 is deprimed

c) \( \Delta T_{3-1} \approx 7.8 \text{C}, \) HP1 and HP2 deprimed

d) \( \Delta T_{3-1} \approx 2.3 \text{C}, \) all heat pipes deprimed
These values for $\Delta T_{3-1}$ are consistent with the temperature levels observed
during anomalous days and tend to confirm the postulate that the anomalies
are initiated by depriming of HP1, followed by depriming of HP2, and the
onset of tube body unstable temperature rise caused finally by depriming
of HP3.

On day 253, while LeRC personnel were performing experiments on-board
CTS, the fourth anomaly occurred. This event gave the unique opportunity
to establish the stable VCHPS heat rejection capacity beyond which the
tube body temperature becomes unstable. It was found the open artery
capacity of the VCHPS is approximately 106 watts.

The comprehensive investigative work performed at LeRC made signifi-
cant contributions to the understanding of the CTS thermal anomalies. The
most significant contribution is the fact that convincing evidence was
made available to establish that the thermal anomalies were caused by
unexpected reductions in the VCHPS heat transfer capability resulting from
depriming of the heat pipe arteries.

The first preliminary investigation of the CTS thermal anomalies con-
ducted at TRW is documented in Reference 2. This investigation, performed
in direct support of LeRC efforts, focussed particular attention at the
identification and analysis of potential artery depriming mechanisms.

Selection of depriming mechanisms was based on the premise that they
must be consistent with three key observations:

1) All the anomalies occurred during the eclipse season.

2) These anomalies are sporadic. Similar conditions on successive
days can yield an anomaly one day but not the next.

3) The anomalies are triggered by depriming of arteries in HP1.

In the early study, among a number of postulated mechanisms, only
three were identified which could be associated with eclipse seasons:

I) Condenser freezing

II) Marangoni flow

III) Gas evolution in the arteries due to rapid chilling
It was argued that under certain conditions freezing of CTS arteries could lead to depriming. Arterial failure could result from the transfer of mass from the evaporator to the frozen condenser where it would freeze and not be available for circulation. This transfer of mass could be effected by vapor diffusion, by liquid pumping in fillets due to reduction of specific volume as it cools and/or by Marangoni flow. The fact that the CTS-type heat pipes will not fail due to condenser freezing when operated under no load or at high heat loads, led to the argument that the condenser freezing mechanism will work if the pipe is operated at the right load, not too high, not too low. However, several factors implied that, although condenser freezing can lead to artery depriming, it could not be the primary cause of the anomalies.

These factors are the fact that on day 253 an anomaly occurred before the radiator portion of HP1 approached freezing conditions, and the fact that with all pipes primed HP3 should freeze and deprime first since it carries least load and its radiator portion radiates from both sides, however, HP1 failed first. An additional factor is the fact that after the anomalies the VCHPS recovers before the radiator begins to warm up. It was thus argued that if sufficient excess fluid was frozen to cause artery depriming there would be none available to rewet the evaporator. Therefore, condenser freezing was ruled out.

Marangoni Flow (surface tension flow due to temperature gradients along the a VCHP gas-blocked region) could induce a flow along fillets and excess fluid reservoirs toward the condenser end along the vapor-liquid interfaces. No flow, however could be induced in the wick and arteries which have structure to support the surface tension gradient. It was argued that Marangoni Flow could influence the VCHP in two ways:

1) If the condenser end is frozen, Marangoni Flow toward that end would increase freezing under load. However, condenser freezing was argued not to be the primary mechanism for depriming.

2) If the heat pipe is not frozen, Marangoni Flow will cause a pressure drop along fillets and natural reservoirs reducing their pumping capacity. However, the reduction must be small, otherwise anomalies due to Marangoni Flow effects would occur on consecutive days under similar conditions, not sporadically. It was argued, however, that once deprimed,
the open artery capacity could be measurably influenced by Marangoni Flow.

In view of the fact that the control gases helium and nitrogen, particularly helium, have solubilities in methanol decreasing with decreasing temperature and pressure, it was argued that this behavior could give rise to a potential depriming mechanism which is consistent with sporadic occurrences during the eclipse seasons. This mechanism is gas evolution within the arteries during heat pipe chilldown.

It was postulated that when TEP ceases to operate several hours before an eclipse, the gas blocks the entire length of the heat pipes. During this off period, liquid in the arteries becomes saturated with gas surrounding the arteries at the gas-blocked condenser temperature and evaporator pressure. When the CTS enters eclipse, the heat pipes experienced rapid chilldown which causes the temperature and pressure to drop rendering the liquid in the arteries sufficiently supersaturated in dissolved gas to cause gas bubble nucleation within the arteries.

In the early study, it was postulated that gas would evolve at unspecified nucleating sites leading to numerous small bubbles, some of which would coalesce to form fewer but larger bubbles with intrinsic longer lifetimes. When the TEP is reactivated, the heat pipes would turn on leading to the establishment of a gas/vapor front and a build-up of stress in the active portion. It was postulated then that if the local liquid stress exceeds a critical level for a bubble before this bubble is redissolved or vented, the arteries would deprime. Of the three potential mechanisms for artery depriming considered in this early study, gas evolution was argued to be the most plausible inasmuch that it ties the anomalies to eclipse seasons when the VCHP system experiences rapid chilldowns. Furthermore, it also provided an explanation for the sporadic occurrences within eclipse seasons since, it depends on the operating conditions before and after the eclipse and on bubble nucleation and aglomeration which are statistical phenomena.

Although no attempts were made at that time to analytically treat the gas evolution scenario, a simple experiment was performed to determine whether gas bubbles would be nucleated in the arteries under CTS conditions.
The experimental setup consisted of a short section of CTS-type arterial wick inserted into a glass tube sealed at one end and valved at the other. The tube was filled with appropriate amounts of 10 percent He/90 percent N₂ gas and methanol simulating CTS design parameters. The tube was then subjected to temperature reduction under hydrostatic stress in excess of open artery capacity. The imposed stress was equivalent to increased supersaturation which would enhance the probability of bubble nucleation. Experiments at temperature reduction rates of -1.71 C/Min and -0.9 C/Min yielded negative results even though the cooling rates were about twice those observed in orbit, and the imposed hydrostatic stress enhanced the probability of bubble nucleation.

The results of this experiment lead to the conclusion that the gas evolution mechanism could not have caused artery depriming under conditions experienced by the CTS.

Thus, this preliminary investigation at TRW was unsuccessful in its quest for a single or sequential series of mechanisms for artery depriming consistent with all flight data. However, depriming due to gas evolution within the arteries was regarded as qualitatively consistent with all the observed anomalies and warranted further investigation.

1.3 PHASE I STUDIES

In view of the fact that previous investigations of the CTS thermal anomalies convincingly established depriming of the heat pipes in the VCHP system as the cause of the anomalies but failed to identify the mechanism or sequential series of mechanisms that led to depriming of the arteries, TRW (under contract to NASA LeRC - Contract NAS3-21740) undertook a two-step investigation of the hydrodynamic characteristics of CTS-type heat pipes.

This section of the report summarizes the work performed during Phase I and refers to attached appendices (documents generated during Phase I) for further details.

The first task performed was to identify mechanisms which under a wide range of feasible conditions could lead to depriming of the heat pipe arteries. Ten mechanisms were postulated which are briefly described and classified in four basic groups in Figure 1-4. Also shown are comments on
Figure 1-4. Observations on Postulated Artery Depriming Mechanisms
the postulated mechanisms in light of observed characteristics of the anomalies and flight data. The approach in Phase I to investigate the potential of these mechanisms is shown in Figure 1-5.

The basic approach has been, for a given depriming mechanism, to conduct analyses in the following sequential order with the implementation of each succeeding task being contingent upon the results of the previous one:

a) Analytical "back-of-the-envelope" calculations and/or flight data review.

b) Calculations with more realistic analytical models.

c) Bench-scale experiments such as with glass heat pipe.

d) Experiments with CTS SN009 heat pipe which is similar to heat pipe 1 on the CTS.

e) Correlations of experimental data with thermal (SINDA) and heat pipe (UCHPDA) analytical models results.

Tasks a, b, c, and d were performed by TRW, whereas task e was the result of a joint effort with LeRC in which the experimental data generated at TRW facilities were correlated with the results of steady state and transient thermal analyses of TEP performed at LeRC. The LeRC study is briefly reviewed in Appendix A.4.1.

Mechanisms Exceeding Heat Pipe Capacity

The arteries in a heat pipe can certainly be caused to deprime if the imposed heat load exceeds the designed heat transfer capacity of the heat pipe. Exceeding the heat pipe capacity can be the result of increased dissipation by TEP, excess skew in load partitioning on the VCHPS, and high transient heat pipe load due to condenser and/or reservoir shadowing.

Increased Dissipation (Column D of Figure 1-4). This is clearly not the cause of artery depriming since:

1) TEP power dissipation exceeding designed levels is not supported by flight power data

2) The heat loads are well below the system capacity when all three heat pipes are primed. In fact a single primed heat pipe can carry the maximum designed dissipation heat load.
<table>
<thead>
<tr>
<th>INVESTIGATION AREA</th>
<th>EXPERIMENT</th>
<th>ANALYSIS</th>
<th>COMPUTER MODELLING</th>
</tr>
</thead>
<tbody>
<tr>
<td>SUCTION AND BLOW-BY FREEZING B AND C</td>
<td>SN 006: FREEZE CONDENSER AS FUNCTION OF S S. LOAD AND TILT. ALSO AS PART OF TRANSIENT POWER INCREASE</td>
<td>IF DE-PRIMING OBSERVED</td>
<td>• MODIFY SINDA MODEL FOR CORRECT GAS AND LIQUID INVENT. ALSO, RESISTANCES, DIMENSIONS, ETC. 0 DET LHC MODEL FROM LOU GEDIONI, ADD SUBROUTINE TO DECOUPLE RES FROM COND WITH ICE PLUG</td>
</tr>
<tr>
<td></td>
<td>(1-0 EFFECTS ON BLOW-BY?)</td>
<td></td>
<td>• RUN SINDA TO DETERMINE GAS FRONT POSITIONS AND TEMP DISTRIBUTION ON ANOMALY DAYS</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• MODIFY MULTIWICK TO INCLUDE MARANGONI FLOW AND FREEZING</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• RUN MULTIWICK TO ESTABLISH INVENTORY DISTRIBUTION AND STRESS AS FUNCTION OF FREEZING PROCESS</td>
</tr>
<tr>
<td>BUBBLE PRODUCTION G, H AND I</td>
<td>GLASS TEST SECTION AND/OR GLASS HEAT PIPE: TRY TO PRODUCE BUBBLES IN N₂-SATURATED METHANOL BY CHILLING, BY FREEZING, BY SUDDEN DROP IN PRESSURE</td>
<td>CALCULATE NUMBER OF CRITICAL SIZE BUBBLES WHICH CAN BE GENERATED DUE TO SUPERSATURATION IN RAPID CHILDOWN. CONSIDER BOTH TEMPERATURE AND PRESSURE EFFECTS</td>
<td></td>
</tr>
<tr>
<td>SUDEN COOLING F</td>
<td>SN 009: RAPIDLY CHILL/FREEZE CONDENSE? AND/OR RESERVOIR TO SEE IF PIPE CAN BE DEPRIMED</td>
<td>FIRST ORDER CALCULATION OF INCREASE IN LOAD DUE TO COOLING-INDUCED EXPANSION OF FRONT</td>
<td>SINDA CALCULATIONS: ARE MINIMUM NECESSARY CONDITIONS FEASIBLE. DOES THERE EXIST A WINDOW IN POWER PROFILE (LOW LOAD - EXCESS RESERVOIRS FULL OF LIQUID TO HANDLE TRANSIENT, HIGH LOAD - RADIATOR NEARLY FULL ON WITH LITTLE POTENTIAL FOR LOAD INCREASE, OR - RES ON NO. 1 MUST BE SHADOWED WITH HIGH LOAD ON 1 BUT 2 AND 3 NOT SIGNIFICANTLY)</td>
</tr>
<tr>
<td></td>
<td>YES</td>
<td>IF SIGNIFICANT</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• SINDA MODELLING: CALCULATE LOAD ON H, P, NO. 1 THROUGHOUT ORBITAL TRANSIENT</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• MULTIWICK: CALCULATE CAPACITY OF H, P, NO. 1 THROUGHOUT ORBITAL TRANSIENT USING SINDA OUTPUT FOR GAS FRONT AND TEMP DISTRIBUTIONS</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• DO PARAMETRIC LAB TESTS TO DETERMINE LEAST SEVERE CONDITIONS NECESSARY TO DEPRIME PIPES</td>
</tr>
<tr>
<td>EXCESS SKEW IN LOAD PARTITIONING E</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>INERTIAL ACCELERATION LOAD J</td>
<td>EXAMINE S/C ACCELEROMETER DATA, IF AVAILABLE</td>
<td></td>
<td></td>
</tr>
<tr>
<td>DEPRIMING SEQUENCE</td>
<td>EXAMINE FLIGHT DATA IN DETAIL. DAY 25: ΔT₂, 2 AND ΔT₂, 1 ON ANOMALY AND NORMAL DAYS: CORRELATION OF T₁, T₂, T₃ WITH BODY TEMPERATURE INDICATING PRIMED STATE</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 1-5. Approach to Investigate Potential Depriming Mechanisms
3) On many days preceding and/or following the anomalies, higher heat loads were accommodated without difficulty.

**Excess Skew in Load Partitioning** (Column E of Figure 1-4). The three heat pipes of the VCHPS on CTS are designed to turn on sequentially so as to balance the load between them at full-on design conditions. However, very low sink temperatures during equinox conditions could result in the load of heat pipe HP1 exceeding transiently its capacity before HP2 and HP3 take up a significant share of the load. Failure of HP1 by the above mechanism or by any other could then result in a rapid transfer of load to HP2 which, due to inertial effects associated with rapid load transfer, could fail leading to a rapid transfer of load to HP3, yielding, perhaps, a "domino effect" failure of the whole system.

Failure of HP1 due to excess skew in load partitioning is discounted since LeRC transient thermal results for periods preceding the four anomalies indicate the maximum load on HP1 never exceeds 80 watts, a load that is well below its primed heat transfer capacity.

In Appendix A.1 flight data are analyzed to determine whether there exists a regularity in the depriming sequence associated with the anomalies, and the possibility of a "domino effect" failure of the system is investigated experimentally.

The analysis of flight data showed that there exists some regularity in the depriming sequence as follows: HP1 deprimes first, HP2 deprimes next followed by depriming of HP3. (On days 89 and 253 the temperature data suggest that heat pipe 2 or 3 was deprimed, while HP1 carried the load.) Experimentally, attempts were made to deprime the SN009 heat pipe with rapid increases of load, simulating, under more stringent conditions, rapid sequential load transfer as heat pipes deprime.

Tests performed for both high (-18C) and low (-96C) sink temperatures and at 2.5 cm evaporator tilt, indicated no measurable rate effect attributable to fluid inertia and thus argued against a "domino effect".

**High Transient Heat Pipe Load Due to Condenser and/or Reservoir Shadowing** (Column F of Figure 1-4). Sudden shadowing cools the heat pipe reservoir which causes the vapor/gas front to move toward the reservoir end of the radiator increasing the active portion of the condenser which
causes the heat load on the pipe to increase. In addition, shadowing of
the VCHPS radiator increases the load on the heat pipes due to a drop in
the effective environment temperature.

Although both effects will eventually lead to cooling of the heat
source, movement of the gas/vapor front reducing the active condenser sec-
tion, and reequilibration at the dissipation load, the transient peak may
be sufficient to exceed the heat pipe capacity and deprime the arteries.

An examination of flight data corresponding to periods preceding and
including the onset of the four anomalies shows a scenario of decreasing
temperatures of the gas reservoirs and on sections of the VCHP system
radiator. Telemetry data for temperature sensors HPT5 and HPT6 (see Fig-
ure 1-3) show that at these locations the temperature drop monotonically
prior to the anomalies.

In order to explore the potential of this postulated mechanism for
artery depriming, the transient loads possible under typical orbital con-
ditions were estimated using a simplified analytical model of the VCHP
system. The description of the model and the basis for the transient loads
calculations are presented in Appendices A.2.1 and A.2.2, which address the
effects of shadowing of the reservoir and condenser, respectively. The
analytical estimates show, that under some assumed cooling rates, the
instantaneous heat loads may be as much as 45 percent larger than steady
state conditions would indicate. These analytical results were found sig-
ificant enough to warrant further tests on the SNO09 heat pipe.

Appendix A.2.3 describes a series of tests performed on the SNO09
heat pipe during which the condenser or the gas reservoir was cooled at
various rates while the evaporator was maintained at a constant heat load.
In tests loads of 100 and 150 watts and cooling of the condenser at
approximately 2.8 C/Min produced no depriming. No depriming was observed
either in tests with 125 watts and at a condenser cooling rate of approxi-
mately 3.9 C/Min. Thus, depriming due to rapid chilldown of the condenser
was not demonstrated.

During tests in which the gas reservoir was cooled repeatedly at suc-
cessively higher rates depriming was observed. With 100 watts at the
evaporator and the inactive condenser at -18 C, depriming occurred when the
reservoir was cooled at the threshold rate of approximately -3.2 C/Min.
Although depriming due to rapid chilldown of the reservoir was demonstrated, the required rates were substantially higher than those observed during the anomalies. This fact eliminates the postulated sudden cooling mechanism as a potential cause of the CTS anomalies.

Mechanisms for Depletion of Liquid Inventory

Another possible cause of artery depriming in a heat pipe is depletion of liquid inventory, either depletion of total heat pipe inventory such as that resulting from a leak, or depletion of liquid from the active portion of the pipe such as could result from diffusion freezeout, suction freezeout or freezing blow by. Suction freezeout is the loss of liquid from freezing. Freezing blowby is the loss of liquid from the thawing of an ice plug allowing a high pressure evaporator to blow liquid from the active pipe into a low pressure gas reservoir.

Loss of inventory due to leaks will cause permanent changes in heat pipe performance, an event that is not supported by the observed complete recovery of the system following an anomaly. Diffusion freezeout is the transfer of inventory in the vapor phase from the active pipe into the frozen section where the vapor condenses and freezes. It is too slow a transfer process to have been effective during the time available prior to the anomalies.

**Suction Freezeout** (Column B of Figure 1-4). Suction freezeout has long been recognized as an artery depriming mechanism. Depriming occurs due to depletion of evaporator liquid because of the local density increase as a freezing front moves toward the evaporator end of the pipe. It is necessary, however, that the condenser freeze while the heat pipe is under load so that the natural liquid reservoirs in the evaporator do not contain excess liquid. Such excess liquid would satisfy the demand of the freezing process without stressing the arteries to failure. Suction freezeout is a viable explanation for depriming on anomaly days 75, 82, and 101. But not on day 253, since the results of steady state and transient calculations on the CTS model performed at LeRC indicate no freezing on this day at the time of the anomaly.

This depriming mechanism was investigated analytically in the previous TRW's study\(^{(2)}\) where the mechanism is referred to as "Condenser Freezing".
The results of this investigation made in light of flight data and observation about the anomalies were summarized in Section 1.2 of this report.

In Phase I of this program, suction freezeout was subjected to experimental investigation with tests on the SNO09 heat pipe. Several attempts to deprime the arteries by freezing the condenser while the SNO09 heat pipe was under hydrostatic and thermal load were unsuccessful.

The experimental results support the conclusion of the previous investigation(2) which states that although suction freezeout or condenser freezing can and may lead to artery depriming, it is not the primary cause of the anomalies.

Freezing Blowby (Column C of Figure 1-4). The CTS heat pipes contain substantial amounts of excess liquid. Such excess inventory is required for successful priming of the arteries in earth gravity testing. In zero gravity a slug of excess liquid generally bridges the condenser vapor spaces, acting as a moving membrane between portions of the non-condensible control gas. Subfreezing radiator temperatures can cause the slug of liquid to become an immobilized plug of ice separating the active section of the pipe from its gas reservoir. If during a transient a pressure differential develops across the ice plug, with the pressure higher on the evaporator side, and then the plug partially thaws, this pressure difference can blow liquid from the evaporator to the reservoir side and deplete the active pipe inventory causing the arteries to deprime.

Because freezing blowby requires freezing, this mechanism can not be the cause of the fourth anomaly, since freezing had not occurred immediately preceding the anomaly on day 253. However, freezing blowby could account for some of the anomalies during which freezing occurred.

To explore experimentally the depriming potential of this mechanism, it was first necessary to develop means of forming an ice plug in the laboratory in the absence of the natural slugging of excess liquid expected in zero gravity. Appendix A.3 describes a successful technique that was developed to form an ice plug in the SNO09 heat pipe, and the procedure followed during the blowby tests. Three tests were performed during which depriming occurred each time. The results of these tests clearly established
that freezing blowby is a bonafide depriming mechanism and a candidate to explain at least some of the anomalies.

The latter statement is supported, to a certain extent, by some results from the CTS model analyses which showed that ice plugs formed in all the heat pipes on day 82 several hours prior to the onset of the anomaly. In addition, the results showed that the pressure on the evaporator side would have been higher than on the reservoir side when the plugs thawed forty five minutes before the last heat pipe deprimed. The significance of these calculated results is that they show conditions for freezing blowby may have existed on at least one day of the anomalies.

Mechanisms for Bubble Nucleation

The presence of gas inside the arteries of the heat pipe has long been recognized as detrimental to the stable operation of the pipe. Because the solubilities of nitrogen and, particularly, helium in methanol, decrease with decreasing temperature and pressure, there exists the potential of bubble nucleation within the arteries in CTS-type heat pipes as they undergo rapid temperature reduction. This potential is compounded by the decrease in pressure that results from temperature reduction in the closed heat pipe environment. Liquid methanol saturated with gas can become supersaturated with gas when undergoing rapid temperature and pressure reduction, an event that enhances the potential of bubble nucleation.

Bubbles that might nucleate inside the arteries during the transient cooldown of the heat pipes after the power is turned off and the spacecraft enters into eclipse, can coalesce to form fewer but larger bubbles. If, as the result of their size and the rather slow process of gas diffusion back into surrounding liquid, the bubbles survive until the heat pipes are once again activated under load, some bubbles might be convected into high-stress liquid regions where they can grow in size and consequently deprime an artery.

The process of bubble coalescence and migration is recognized as statistical in nature.

The above postulated mechanism was examined during a previous TRW investigation; however, experiments under simulated anomaly conditions failed to induce bubble nucleation.
Despite these results, this mechanism was reexamined during the current program due to the fact that the postulated conditions for the mechanism to be activated can be supported by flight data corresponding to all the anomalies, and the probabilistic nature of the mechanism bears positive correlation to the sporadic occurrences of the anomalies. In addition, a thorough examination required the performance of experiments simulating more realistically the operating characteristics and anomaly conditions of the CTS heat pipes and investigation of the potential of other mechanisms, particularly freezing-thawing, to induce bubble nucleation.

**Bubble Nucleation Due to Temperature and Pressure Reduction** (Columns G and H on Figure 1-4). An analysis of the potential for bubble formation was performed based on fundamentals described in Appendix A.5.1. Sample calculations were made for two cases (1) a condenser depressurization case where the evaporator temperature is dropped at constant gas-blocked condenser temperature and (2) a condenser chilldown case where the gas-blocked condenser is cooled at constant total pressure.

The sample calculations presented in Appendix A.5.2 showed the possibility for bubble formation upon (1) depressurization by reduction in heat pipe loading and reservoir chilling and (2) condenser chilldown. Depressurization showed greater potential, for approximately 500,000 bubbles/cm³ were found possible, compared to only one of equal size resulting from condenser chilldown. The above calculations, based on two hypothetical cases, were followed by a number of calculations for various anomaly conditions. The predicted number of bubbles of varying critical size was found substantial for all anomaly conditions considered in the analysis.

In order to verify qualitatively the above analytical results, a series of simple experiments were formulated which are described in Appendix A.5.3. Tests were performed with a glass vessel half filled with liquid methanol containing a short section of mesh screen artery laying on its bottom. The vessel was instrumented to permit continuous monitoring of temperature and pressure. The liquid methanol was saturated with either helium or nitrogen at room temperature subsequent to which three series of tests were performed. In the first two series of tests the saturated methanol was subjected to various rates of temperature and/or pressure reduction. During the tests bubble nucleation in the bulk of the liquid
or on the surface of the artery mesh screen was never observed. These results suggested that temperature and pressure reduction are not potential mechanisms for bubble generations in CTS-type heat pipes. In the last series of tests with the glass vessel, the saturated methanol was subjected to several freeze/thaw cycles. Large numbers of bubbles were observed streaming from the ice surface.

**Bubble Generation Due to Freezing (Column I of Figure 1-4).** The results of the glass vessel tests were of paramount importance, for they identified freezing and thawing of gas-saturated methanol in the arteries as a potential mechanism for bubble formation in the arteries. The fact that freezing occurred preceding three of the four anomalies and sometime during the 24 hours before all of them, the involvement of ice-generated bubbles in the four thermal anomalies was shown to be a distinct possibility.

To explore this bubble nucleation mechanism in a realistic heat pipe environment, a series of tests were performed with an existing glass heat pipe. The pipe contains a slab wick with a CTS-type artery attached to one side and permits observations on the behavior of the artery in an operational heat pipe. For the tests the heat pipe was gas loaded with a 90 percent N₂/10 percent He gas mixture at a pressure equivalent to that in the CTS heat pipes. The experiments performed on the glass heat pipe are documented in Appendix A.5.4.

The results of bubble nucleation experiments showed that each time the artery in the condenser section underwent freezing and thawing bubbles were observed inside the arteries although the number of bubbles, their size and location along the artery varied from test to test. The time required for the bubbles to disappear by diffusion was found to vary enormously depending on their size. Small spherical bubbles dissolved within hours, while elongated bubbles required up to several weeks.

Additional experiments were conducted with this heat pipe during which the gas/vapor front was forced to move into the previously frozen condenser section in attempts to incorporate ice-generated bubbles into the active condenser and cause depriming. A number of deprimings were observed, but they were sporadic, probabilistic in nature. These results
conclusively demonstrated that: 1) Control gas is liberated from saturated methanol every time the heat pipe goes through a freeze/thaw cycle, 2) the number, size, and durability of gas bubbles generated within the arteries have a statistical variation and are influenced by bubble coalescence, and 3) arterial bubbles can lead to depriming if they migrate or are convected into the active region of the pipe under normal load conditions. Thus, it is clear that freezing and thawing of the condenser can, but does not necessarily, lead to artery depriming, depending on subsequent history.

Attempts to induce arterial failure in the SNO09 heat pipe due to the ice-generated-bubble mechanism were partially successful. Depriming was not observed in several tests when the heat pipe was brought under a heat load immediately after the condenser had gone through a freeze/thaw cycle. However, depriming was discovered on two occasions after the SNO09 heat pipe had been left unattended under hydrostatic load for several hours during which time the condenser froze following the termination of a normal freeze/thaw cycle.

The results of tests with the SNO09 and glass heat pipes clearly indicated that in order to establish the statistics of this artery failure mode, repeated experiments would be required.

SUMMARY

In the course of Phase I of this program, numerous depriming mechanisms were postulated and subjected to analytical and experimental investigation. Several mechanisms were convincingly rendered highly improbable causes of artery failure on CTS. Four mechanisms, however, were identified as potential candidates to explain the anomalies and are listed in decreasing likelihood in Table 1.

It was argued during this Phase of the study that potential depriming mechanisms must be consistent with three key observations about the anomalies:

1) The anomalies occur only during the radiator eclipse season when condenser freezing may be realized.

2) The anomalies are sporadic. Similar conditions on successive days can yield an anomaly on one day and not on any of several similar days.
Table 1. Summary of Potential Depriming Mechanisms from Phase I.

<table>
<thead>
<tr>
<th>ARTERY DEPRIMING RESULTS FROM:</th>
<th>ANALYTICALLY</th>
<th>EXPERIMENTALLY</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FLASK</td>
<td>GLASS H.P</td>
</tr>
<tr>
<td>I. BUBBLE NUCLEATION DUE TO:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>a) FREEZE/THAW</td>
<td>---</td>
<td>YES</td>
</tr>
<tr>
<td>b) PRESSURE REDUCTION</td>
<td>YES</td>
<td>NO</td>
</tr>
<tr>
<td>c) TEMP REDUCTION</td>
<td>YES</td>
<td>NO</td>
</tr>
<tr>
<td>II. FREEZING BLOW-BY</td>
<td>YES</td>
<td>---</td>
</tr>
<tr>
<td>III. RADIATOR/RESERVOIR</td>
<td>YES</td>
<td>---</td>
</tr>
<tr>
<td>RAPID COOLING</td>
<td></td>
<td></td>
</tr>
<tr>
<td>IV. SUCTION FREEZE-OUT</td>
<td>YES</td>
<td>---</td>
</tr>
</tbody>
</table>

3) On day 253 condenser freezing did not occur immediately prior to the onset of the anomaly.

In light of the above observation, the following comments can be made on the mechanisms listed on Table 1.

Suction freezeout is not a statistical depriming mechanism and cannot account for the anomaly on day 253. Artery depriming due to suction freezeout was not demonstrated with the SN009 heat pipe which seems to indicate that none of the other anomalies could have been caused by this mechanism alone. This mechanism can only be considered a potential contributing factor to some of the anomalies.

The sudden cooling mechanisms could account in principle for all the anomalies, however, they are not statistical in nature. Furthermore, tests with the SN009 heat pipe showed that the cooling rates required to fail the arteries far exceed those indicated by flight data. Thus, this artery failure mode can be assigned a low probability of success and only be considered a contributing factor to some of the anomalies.

Artery depriming due to freezing blowby was verified in repeated experiments with SN009 heat pipe, establishing freezing blowby as a bonafide potential depriming mechanism. Analyses of the CTS thermal model
for day 82 predicted the formation of ice plugs in the CTS heat pipes and calculated pressure differential across the barrier (pressure higher on evaporator side) which were sustained until the ice plugs thawed. These predicted results tend to indicate that conditions for blowby may be present on some of the anomaly days. Although freezing blowby is not a statistical mechanism and can not be the cause of the anomaly on day 253, it can not be totally discounted.

In view of the results of glass flask and glass heat pipe experiments, the ice-generated bubble mechanism appears to be a prime candidate for explaining all four CTS anomalies. It is consistent with (1) their season occurrence (eclipse conditions are necessary to cause condenser freezing), (2) their sporadic occurrence (due to the statistical nature of bubble population and behavior), and (3) the lack of freezing on day 253 (bubbles were generated by a freeze/thaw process during day 252).

1.4 OBJECTIVES OF PHASE II STUDIES

The Phase I studies demonstrated that bubbles are generated each time liquid methanol containing dissolved gas undergoes freezing and thawing, and that these bubbles may result in artery depriming if inducted into the active portion of the heat pipe. This depriming mechanism was demonstrated on the glass heat pipe. However, depriming due to ice-generated bubbles was not conclusively demonstrated on the SNO09 heat pipe. Because the SNO09 heat pipe is similar in overall dimensions and performance characteristics to heat pipe number 1 onboard CTS, the need to demonstrate depriming on the SNO09 due to bubbles became apparent in order to demonstrate beyond a reasonable doubt that the postulated bubble mechanism is the cause of the thermal anomalies.

The Phase I studies also warranted further study of the behavior of bubbles inside arteries, particularly their lifetimes, owing to the fact that the probability of arterial failure due to bubbles induction into high stress regions is positively correlated to longer bubbles lifetimes.

As a result a second phase of this program was continued. The second phase was necessary to bring the CTS-anomaly studies to the satisfactory conclusion envisioned at the outset.
The objectives of Phase II were:

1. To analyze the diffusion process for non-condensible control gases in arteries and to determine lifetimes of bubbles of various sizes at various temperature levels and rates of change of temperature level and for various selected control gases.

2. To run continuous cyclic freeze/thaw tests on the SN009 heat pipe for five consecutive days in an effort to produce artery depriming by freeze-thaw bubble formation. The successful triggering of a failure would establish that freeze-thaw bubble formation is the logical cause of the CTS thermal anomalies.

3. To propose concepts to avert artery depriming.
2.0 ANALYSIS OF BUBBLE LIFETIMES IN HEAT PIPE ARTERIES

2.1 THE SPHERICAL BUBBLE MODEL

Consistent with Phase II objectives 1 and 3, it is desired to calculate how fast vapor bubbles in heat pipe arteries might grow or collapse and how the kind of gas used affects the growth/collapse rates. The first model selected for such calculations is the one-dimensional spherical annulus, the domain $R_1 < r < R_2$ as shown in Figure 2-1.

The vapor-and-gas-filled bubble ($r < R_1$) has uniform composition, because gas-phase diffusivities exceed liquid phase diffusivities by several orders of magnitude. The external gas $r > R_2$ has known pressure and composition. The interface at $R_2$ is considered to be a screen like the wall of an artery so that the pressure difference across the interface is uncoupled from that otherwise dictated by surface tension and $R_2$. At $r=R_1$ no screen exists, so the vapor-to-liquid pressure difference is $2\sigma/R_1$ where $\sigma$ is surface tension.

Figure 2-1. Spherical Bubble Model

The vapor-and-gas-filled bubble ($r < R_1$) has uniform composition, because gas-phase diffusivities exceed liquid phase diffusivities by several orders of magnitude. The external gas $r > R_2$ has known pressure and composition. The interface at $R_2$ is considered to be a screen like the wall of an artery so that the pressure difference across the interface is uncoupled from that otherwise dictated by surface tension and $R_2$. At $r=R_1$ no screen exists, so the vapor-to-liquid pressure difference is $2\sigma/R_1$ where $\sigma$ is surface tension.
The following heat pipe scenario is postulated: The heat pipe pictured in Figure 2-2 operates with an active section at 300K and a gas-blocked condenser at 250K long enough for the artery liquid to saturate with the gas. Then as pictured in Figure 2-3 the condenser is briefly frozen and then thawed to 180K. Upon thawing of the gas-blocked condenser, there are, as has been observed experimentally, large numbers of minute bubbles released from the ice, but most diffuse back into solution. A few are postulated to condense to form the bubble of prescribed size $R_{1,0}$. The noncondensible gas compositions in the bubble and in the liquid were taken to be the same, the same as that in the saturated liquid at 250 K. The condenser then warms up at a prescribed rate as pictured in Figure 2-3, affecting the vapor composition in the external gas. Because thermal diffusion proceeds at a much higher rate than mass diffusion (Prandtl number is much smaller than Schmidt number), the artery liquid is assumed to be in thermal equilibrium with the condenser temperature, and the vapor composition of the internal gas is also affected.
2.2 GOVERNING EQUATIONS FOR THE SPHERICAL BUBBLE

Within the liquid, conservation of mass species $i$ takes the form.

$$\frac{\partial x_i}{\partial t} + v \frac{\partial x_i}{\partial r} = D_i \left( \frac{\partial^2 x_i}{\partial r^2} + \frac{2}{r} \frac{\partial x_i}{\partial r} \right)$$  \hspace{1cm} (2-1)

where $i = 2$ to $n$ ($i=1$ is reserved for the liquid). The quantity $x_i$ is the mole fraction of species $i$, and $r$ is the radius. The assumption has been made that the solution is dilute and isothermal, so the molar concentration $c$ is constant at that of the liquid, and the diffusivity $D_i$ for species $i$ in the liquid is unaffected by the other dilute species present. The radial velocity $v$ is caused entirely by growth or collapse of the central bubble, because the diffusion mass transfer rates are so low, conservation of mass dictates

$$v = \left( \frac{R_1}{r} \right)^2 \frac{dR_1}{dt}$$  \hspace{1cm} (2-2)
Within the central bubble the gas is assumed to be uniform in composition and pressure, because the gas-phase diffusivities are high compared to the liquid-phase values, and inertia forces are negligible for the small changes in bubble growth or collapse rate. Accordingly the gas-phase species equations equivalent to Eq. (2-1) are not used explicitly. Rather, they are replaced by the simple expression

$$\frac{dw_i}{dt} = -4\pi R_1^2 N_{i,1}$$  \hspace{1cm} (2-3)

where \(w_i\) is the molar content of species \(i\) within the bubble and \(N_{i,1}\) is the molar flux for species \(i\) at \(r=R_1\), given by Fick's Law on the liquid side of the bubble meniscus as

$$N_{i,1} = c \frac{\partial x_i}{\partial r} \bigg|_{r=R_1}$$ \hspace{1cm} (2-4)

where \(c\) is liquid molar concentration (g-moles/cm\(^3\)).

The molar content of the bubble is given by

$$w_i = \frac{4}{3} \pi R_1^3 \frac{P_i}{R^T_c}$$ \hspace{1cm} i=2,n \hspace{1cm} (2-5)

where \(T_c\) is the condenser temperature, and \(R\) is the universal gas constant. Similarly the conservation of momentum equation is replaced by a quasi-hydrostatic balance equation

$$\sum_{i=1}^{n} P_i = P_{liq} + 2\sigma/R_1$$ \hspace{1cm} (2-6)

The partial pressures within the bubble are given by Raoult's and Henry's Laws. Since \(x_1\) is nearly unity, Raoult's Law gives simply

$$P_1 = P_v(T_c)$$ \hspace{1cm} (2-7)

while Henry's Law is

$$P_i = C_i(T_c) x_i$$ \hspace{1cm} i=2,n \hspace{1cm} (2-8)

Outside the liquid at \(r=R_2\) the total pressure of the vapor and gas is \(P_v(T_{ev})\) where \(T_{ev}\) is the evaporator temperature. The gas pressure in the
gas-blocked condenser is

\[ P_g = P_v (T_{ev}) - P_v (T_c) \]  

(2-9)

The gas composition is specified by \( Y_i \), i=2 to n, where \( Y_i \) is the mole fraction of species i in the noncondensible. For example, if 2 is helium and 3 is nitrogen, \( Y_2 \) might be 0.10 and \( Y_3 \) 0.90 for a 10 percent helium, 90 percent nitrogen gas mixture. The saturation values of \( x_i \) at \( r = R_2 \) are

\[ x_i = Y_i \frac{P_g}{G_i} (T_c) \]  

(2-10)

The initial conditions at \( t=0 \) are specified by a uniform set of \( x_i \), i=2 to n, for the initial liquid composition, a set of \( Y_i \) in the gas bubble and the initial radius \( R_1 \) of the bubble. From these there can be derived the set of \( w_i \), i=2 to n. First, from Eqs. (2-6) and (2-7)

\[ P_g = P_{liq} + 2\sigma (T_c)/R_1 - P_v (T_c) \]  

(2-11)

Then from the gas law

\[ W_g = \frac{4}{3} \pi R_1^3 \frac{P_g}{RT_c} \]  

(2-12a)

where

\[ w_g = \sum_{i=2}^{n} w_i \]  

(2-12b)

Finally

\[ w_i = Y_i w_g \]  

(2-13)

2.3 TRANSFORMATION AND NUMERICAL SOLUTION

It is convenient to anchor the spatial coordinate to the bubble interface and make the coordinate dimensionless with respect to \( R_2 - R_1 \).

\[ n = \frac{r - R_1(t)}{R_2(t) - R_1(t)} \]  

(2-14)
where \( R_2(t) \) is given by
\[
\frac{4}{3} \pi R_2^3 - \frac{4}{3} \pi R_1^3 = \frac{4}{3} \pi R_{2,0}^3 - \frac{4}{3} \pi R_{1,0}^3 \tag{2-15}
\]

Velocity \( v \) given by Equation (2-2) becomes
\[
v = \left( \frac{R_1}{R_1 + n\Delta R} \right)^2 \cdot R_1 \tag{2-16}
\]

Partial derivatives with respect to \( r \) at constant \( t \) (and thus at constant \( R_1 \) and \( \Delta R = R_2 - R_1 \)) are given by
\[
\frac{\partial x_i}{\partial r} = \frac{\partial x_i}{\partial n} \frac{1}{\Delta R} \tag{2-17}
\]

The partial derivative with respect to \( t \) at constant \( r \) must be transformed to one with respect to \( t \) at constant \( n \)
\[
\frac{\partial x_i}{\partial t} \bigg|_r = \frac{\partial x_i}{\partial t} \bigg|_n + \frac{\partial x_i}{\partial n} \frac{\partial n}{\partial t} \bigg|_r \tag{2-18}
\]
\[
\frac{\partial n}{\partial t} \bigg|_r = - \frac{1}{\Delta R} \cdot R_1 - \left( \frac{r-R_1}{\Delta R^2} \right) \Delta R \tag{2-19}
\]

Hence Equation (2-1) becomes
\[
\frac{\partial x_i}{\partial t} \bigg|_n + \frac{\partial x_i}{\partial n} \bigg|_t \left( - \frac{1}{\Delta R} \cdot R_1 - \frac{n}{\Delta R} \cdot \Delta R \right) + \left( \frac{R_1}{R_1 + n\Delta R} \right)^2 \cdot R_1 \frac{\partial x_i}{\partial n} \frac{\partial n}{\partial t} \bigg|_r = D_i \left( \frac{1}{\Delta R^2} \frac{\partial^2 x_i}{\partial n^2} + \frac{2}{R_1 + n\Delta R} \frac{1}{\Delta R} \frac{\partial x_i}{\partial n} \right) \tag{2-20}
\]

Collecting terms gives
\[
\frac{\partial x_i}{\partial t} + \left( \frac{R_1^2 \cdot R_1}{(R_1 + n\Delta R)^2} - \frac{(1+n)R_1}{\Delta R} - \frac{2D_i}{(R_1 + n\Delta R)\Delta R} \right) \frac{\partial x_i}{\partial n} - \frac{D_i}{\Delta R^2} \frac{\partial^2 x_i}{\partial n^2} = 0 \tag{2-20}
\]

Since large values of \( \frac{\partial x_i}{\partial n} \) are expected near \( n = 0 \), computational efficiency is improved by transforming the \( n \) coordinate to \( z \) so as to expand the region near \( n = 0 \) relative to the region near \( n=1 \). A negative
value of the parameter $\gamma$ in the following coordinate transformation achieves the desired result.

$$z = \frac{1 - e^{\gamma n}}{1 - e^{\gamma}}$$  \hspace{1cm} (2-21)

After transformation Equation (2-20) is in the form of

$$\frac{\partial x_i}{\partial t} + F \frac{\partial x_i}{\partial z} + G \frac{\partial^2 x_i}{\partial z^2} = 0$$  \hspace{1cm} (2-22)

where $F$ and $G$ are functions of $z$, $R_1$, $R_1^\prime$, $D_i$, and $\gamma$ (recall that $R_2$ and hence $\Delta R$ is given by Equation (2-15)).

Equation (2-22) is solved numerically in finite difference form. Let subscript $j$ denote $z$ location and superscript $o$ and $oo$ denote values at $t-\Delta t$ and $t-2\Delta t$ respectively. The species subscript $i$ is dropped to avoid confusion. The temporal derivative is approximated as

$$\frac{\partial x_i}{\partial t} = \frac{1}{\Delta t} \left[ a_1 x_j + a_2 x_j^o + a_3 x_j^{oo} \right]$$  \hspace{1cm} (2-23)

where for constant time increment $\Delta t$, $a_1 = 3/2$, $a_2 = -2$, and $a_3 = 1/2$.

The spatial derivatives are

$$\frac{\partial x_i}{\partial z} = \frac{x_{j+1} - x_{j-1}}{2\Delta z}$$  \hspace{1cm} (2-24)

$$\frac{\partial^2 x_i}{\partial z^2} = \frac{x_{j+1} + x_{j-1} - 2x_j}{(\Delta z)^2}$$  \hspace{1cm} (2-25)

Equation (2-22) takes the form

$$x_j = A_j x_{j+1} + B_j x_{j-1} + C_j, \quad j = 2, N$$  \hspace{1cm} (2-26)

where $z_2 = 0$ and $z_N = 1$. The coefficients $A_j$, $B_j$, and $C_j$ may be calculated at any time step in terms of the previous known sets $x_j^o$ and $x_j^{oo}$ and the known values of $R_1$, $R_1^\prime$, $D_i$, $\gamma$, and $z_j$. The new set of $x_j$ values is computed by means of Gaussian elimination in which
Equations (2-26) and (2-27) combine to permit the forward (j=2, 3, ...) calculation of $A_j^*$ and $B_j^*$.

$$A_2^* = 0, \quad B_2^* = B_2 \ x_1 + C_2$$

(2-28a,b)

$$A_j^* = A_j / (1 - B_j \ A_{j-1}^*)$$

(2-28c)

$$B_j^* = (B_j \ B_{j-1}^* + C_j) / (1 - B_j \ A_{j-1}^*)$$

(2-28d)

Then Equation (2-27) allows the backward (j=N-1, N-2, ...) calculation of $x_j$ starting with the prescribed boundary value $x_n$.

Appendix B.1 contains the listing of the program used to calculate $x(z_j,t)$ and $R_1(t)$.

2.4 THE ELONGATED BUBBLE MODEL

When a bubble grows so that its radius ($R_1$) reaches the artery radius, further growth occurs through elongation of the bubble within the artery. In the absence of a priming foil, the artery wall prevents meniscus coalescence, and a sheath of liquid is retained about the elongated bubble. Mass transfer of species $i$ ($i=2$ to $n$) can occur through the sheath and into or out of the end-cap liquid. Equations (2-5) and (2-12) become

$$w_i = \left( \frac{4}{3} \pi R_a^3 + \pi R_a^2 L \right) P_i / RT_c$$

(2-29)

$$w_g = \left( \frac{4}{3} \pi R_a^3 + \pi R_a^2 L \right) P_g / RT_c$$

(2-30)

while Equations (2-6) through (2-11) and (2-13) remain unchanged.

Equation (2-3) and (2-4) are well approximated by

$$\frac{dw_i}{dt} = cD_i \left[ (2\pi R_a L \ v/\tau \Delta R) + (F/R_a) \right] \Delta x_i$$

(2-31)

where $\Delta x_i$ is the difference in mole fraction between that of the inside liquid at the bubble-liquid interface and that of the outside liquid at the condenser-gas-liquid interface. The quantity $v$ is the artery screen...
void fraction, \( \tau \) is its tortuosity, and \( F \) is the conduction shape factor for a hemispherical bubble of radius \( R_a \) in a semi-infinite cylinder of radius \( R_a + \Delta R \). For the CTS heat pipe arteries \( R_a = 0.0127 \text{ cm}, \ c = 0.37, \ R_a/\Delta R = 8 \), and \( F = 8 \).

Equation (2-31) was integrated numerically with Equation (2-30) used to find \( L \) in a Runge-kutta type algorithm.

2.5 PROPERTIES USED IN CALCULATIONS

Figures 2-4 (Ref. 3) shows the solubility data used for the calculations. Shown versus temperature is the mole fraction in the liquid for a partial pressure of 1 atm, that is the reciprocal of Henry's constant in \( \text{atm}^{-1} \). The variation with temperature will be shown to be quite significant. For now, merely note that methane and argon become less soluble in methanol with increasing temperature (as does air in water) while helium becomes less soluble with decreasing temperature. The solubility of nitrogen in methanol is nearly independent of temperature. Helium is seen to be only sparingly soluble while methane is quite soluble in methanol.

Figure 2-5 (Ref. 4) shows the diffusivity of argon and helium in liquid methanol versus reciprocal temperature. The theory shown in the figure agreed poorly with experimental data, and it was decided to fit the experimental data with an Arrhenius relation. Since data were lacking and theory seemed dubious, it was decided to use the experimental helium fit for helium diffusivity and the experimental argon fit for the diffusivity of argon, nitrogen, and methane. Thus the calculated results for the latter two gases should be regarded as only qualitatively correct. When reliable data for these latter two gases become available, a minor time-scale expansion or contraction will be necessary to adjust for the approximate values of diffusivity used.

2.6 RESULTS FOR THE SPHERICAL BUBBLE

Figures 2-6 through 2-9 show the calculated results for the spherical bubbles. Each of the four figures is for a different gas: helium, argon, methane, and 10 percent He 90 percent N\(_2\), respectively. For a given condenser warmup rate and initial bubble size, helium bubbles are seen to persist for very much longer times than argon bubbles. Methane bubbles disappear much faster than even argon bubbles. The explanation is simply
Figure 2-4. Solubility of Various Gases in Methanol as Function of Temperature (from Reference 3)
Figure 2-5. Diffusivity of Helium and Argon in Methanol as Function of Temperature (from Reference 4)
Figure 2-6. Spherical Bubble Radius History, Helium Gas
Figure 2-8. Spherical Bubble Radius History, Methane Gas
Figure 2-9. Spherical Bubble Radius History, 10% Helium/90% Nitrogen Gas
that, despite its higher diffusivity, the solubility of helium in methanol is low; hence the values of \( c_{D_i} \Delta x_i \) are low for helium. Because of the helium component, the 90\%N\(_2\) - 10\%He noncondensible gas mixture displays long bubble lifetimes.

For a given gas and condenser warmup rate, small bubbles disappear much faster than large bubbles. In the small bubble the gas pressure is larger due to surface tension, so the values of \( x_i \) are larger. The mass transfer coefficient area product decreases with size as \( R_i^{-1} \), but the volume of gas remaining decreases as \( R_i^{-3} \).

With helium gas in a bubble of a certain initial size, increasing the condenser warmup rate increases the solubility and diffusivity of the helium and hastens the reabsorption of the bubble, despite a decreased surface tension, and an increased pressure of methanol vapor which tends to swell the bubble.

With argon the increase in diffusivity with increasing temperature causes the same type of behavior as in the case of helium but to a lesser degree. With methane, however, the increase in diffusivity offsets the decrease in solubility and lessened surface tension, so that the time to reabsorb small bubbles is nearly independent of condenser warmup rate. A complex behavior is seen for the large bubble and high condenser warmup rate. In that case, the fall in solubility of methane and the increase in methanol vapor pressure combine to cause the methane gas and methanol vapor bubble to swell before ultimately collapsing.

Table 2-1 summarizes the bubble lifetime results. Note the much longer times necessary for helium-containing bubbles to be reabsorbed.

2.7 RESULTS FOR THE ELONGATED BUBBLE

Figures 2-10 through 2-13 show calculated results for the elongated bubbles. Again each figure is for a different gas or gas mixture. Again helium-containing bubbles are seen to persist to a much longer time, while methane bubbles are most readily reabsorbed. Again an increase in condenser warmup rate strongly hastens the reabsorption of helium bubbles and weakly hastens the reabsorption of argon and methane bubbles. Bubble swelling, elongation in this case, is seen in all cases, because reabsorption is initially slower than for small spherical bubbles. A final
Table 2-1. Calculated Spherical Bubbles Lifetimes

<table>
<thead>
<tr>
<th>Gas/Gas Mixture</th>
<th>Initial Radius (cm)</th>
<th>Condenser Warm-up Rate (K/sec)</th>
<th>Time, in seconds, at which bubble radius reaches 1/10 of initial radius.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane</td>
<td></td>
<td>0.0</td>
<td>0.01</td>
</tr>
<tr>
<td>CH₄</td>
<td>0.01</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td></td>
<td>0.02</td>
<td>68</td>
<td>68</td>
</tr>
<tr>
<td></td>
<td>0.04</td>
<td>3775</td>
<td>4362</td>
</tr>
<tr>
<td>Helium</td>
<td></td>
<td>0.01</td>
<td></td>
</tr>
<tr>
<td>He</td>
<td>0.01</td>
<td>10717</td>
<td>4055</td>
</tr>
<tr>
<td></td>
<td>0.02</td>
<td>64411</td>
<td>9998</td>
</tr>
<tr>
<td></td>
<td>0.04</td>
<td>371408</td>
<td>38651</td>
</tr>
<tr>
<td>Argon</td>
<td></td>
<td>0.01</td>
<td></td>
</tr>
<tr>
<td>Ar</td>
<td>0.01</td>
<td>208</td>
<td>204</td>
</tr>
<tr>
<td></td>
<td>0.02</td>
<td>3614</td>
<td>2790</td>
</tr>
<tr>
<td></td>
<td>0.04</td>
<td>27890</td>
<td>10740</td>
</tr>
<tr>
<td>CTS Mixture</td>
<td></td>
<td>0.01</td>
<td></td>
</tr>
<tr>
<td>90% N₂/10% He</td>
<td>0.01</td>
<td>1514</td>
<td>1216</td>
</tr>
<tr>
<td></td>
<td>0.02</td>
<td>10777</td>
<td>4950</td>
</tr>
<tr>
<td></td>
<td>0.04</td>
<td>67520</td>
<td>16038</td>
</tr>
</tbody>
</table>
Figure 2-10. Elongated Bubble Length History, Helium Gas

PARAMETERS:
L0 = 2.5 CM
Conductor Warmup Rate (CMR) k/SEC

Dimensionless Length, L/L0
Figure 2-13. Elongated Bubble Length History, 10% Helium/90% Nitrogen Gas

PARAMETERS:
$L_0 = 2.5$ cm
CONDENSER WARMUP RATE (CWR) K/sec

DIMENSIONLESS LENGTH, $L/L_0$
observation is that the helium in the helium-nitrogen gas mixture becomes the residual gas in the bubble at long times and low condenser warmup rates and requires a long time to diffuse out of the bubble.

Table 2-2 summarizes the calculations of the elongated bubble lifetimes.

Table 2-2. Calculated Elongated Bubbles Lifetimes

<table>
<thead>
<tr>
<th>Gas/Gas Mixture</th>
<th>Initial Length CM</th>
<th>Condenser Warm-up Rate (K/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>0.0</td>
</tr>
<tr>
<td>Methane CH₄</td>
<td>2.5</td>
<td>110,070</td>
</tr>
<tr>
<td>Helium He</td>
<td>2.5</td>
<td>5,069,500</td>
</tr>
<tr>
<td>Argon Ar</td>
<td>2.5</td>
<td>406,600</td>
</tr>
<tr>
<td>CTS Mixture 90% N₂/10% He</td>
<td>2.5</td>
<td>998,800</td>
</tr>
</tbody>
</table>

2.8 SIGNIFICANCE OF THE ANALYTICAL/NUMERICAL RESULTS

The CTS heat pipes contained 90%N₂ - 10%He control gas. The helium was added to facilitate leak detection. In retrospect, that decision appears to have been unfortunate. For should gas bubbles be created by freezing in the gas-blocked condenser and should they coalesce into bubbles approaching 0.8 mm in diameter, their lifetimes are seen to be on the order of 20 hours when the condenser warmup rate is slow. Larger sausage bubbles are seen to persist hundreds of hours, that is, some ten days or more. Such bubbles, if present through cyclic freezing and thawing may be enlarged through a fractionation process reminiscent of the making of New England applejack. With each freezing episode dissolved gas is forced out
of the ice into the bubble. A large bubble can persist many days even after freezing no longer occurs provided it is in the gas-blocked region. When the heat pipe is powered up, and the gas front approaches the bubble, Marangoni Flow brings the bubble into the active condenser, and condensate flow sweeps it up the pipe toward the evaporator to a location where the local vapor-liquid pressure difference exceeds the capillary pressure of the artery, and the artery deprimes.

The results suggest that methane would make a good control gas, because methane bubbles up to 0.8 mm in diameter are reabsorbed in approximately one hour. Thus there is little likelihood that a 24 hour periodic freezing and thawing could create a large bubble, and the small bubbles released from thawing ice would be reabsorbed before the approach of the gas front.
3.0 CYCLIC FREEZE/THAW TESTS ON SNO09 HEAT PIPE

3.1 TEST OBJECTIVES

Phase II objective 2 was to simulate an anomaly with the SNO09 heat pipe by cyclic freezing and thawing of the condenser during a 5-day extended continuous test period. The goals of the tests were (a) to demonstrate the generation and collection of bubbles in the arteries and their migration or induction into the active section of the heat pipe where their presence could result in depriming of at least a single artery, and (b) to establish by test data analysis that freezing blowby or suction freezeout rather than bubble formation was not the cause of the observed heat pipe failure.

3.2 APPARATUS AND TEST PROCEDURE

The test assembly described in Figure 3-1 is similar to the one used for tests on the SNO09 heat pipe during Phase I of this program. As shown, the heat load is applied on the heat pipe over a 0.305 meter long section by means of an attached 4.0 kg heated aluminum block.

The heat pipe condenser, 0.91 meter long, is mounted on an aluminum plate whose area and thermal mass corresponds approximately to the effective portion of the CTS radiator. This plate is coupled to the sink through 0.32 centimeters of cork over a 0.1045 square meter area. Tape heaters attached along the plate allow changing the condenser temperatures even with a constant temperature sink.

A cooling coil with tape heaters is attached to the gas reservoir for purposes of controlling its temperature independently from the rest of the heat pipe assembly.

The heat pipe test assembly is instrumented with seventeen temperature sensors, the location of which are shown in Figure 3-2. The sensors are connected to a 24-channel strip chart recorder which allows monitoring these temperatures at 1.5 minute intervals.

The test procedure in Table 3-1 outlines the basic steps for conducting the cyclic freeze/thaw tests on the SNO09 heat pipe. Steps 1 through 5 represent the original procedure for repriming the heat pipe and adjusting the operating parameters in preparation for the commencement of a new cycle. As indicated in the procedure, the selected evaporator elevation and heat
Figure 3-1. Engineering Sketch of SNO09 Heat Pipe Test Assembly
Figure 3-2. Locations of Thermocouples on SNO09 Test Assembly
Table 3-1. SNO09 Heat Pipe Freeze/Thaw Cyclic Test Procedure Outline

Using the test assembly described in SK80010 and SK80011, perform the cyclic tests for a period of 120 continuous hours in accordance with the following procedure:

1. Level heat pipe to within 1.0 mm.
2. Isothermalize heat pipe to within 1°C and wait 30 minutes.
3. Elevate evaporator end 1.2 cm above condenser.
4. Apply 130 watts to evaporator heater and verify artery priming. If arteries are primed, proceed with Step 5, otherwise, repeat Steps 1 through 4.
5. Reduce sink plate, inactive condenser and gas reservoir temperatures to the following levels:
   a) Sink plate: lower than -145°C (-229°F)
   b) Inactive condenser: -40°C ±5°C
   c) Gas reservoir: -95°C ±5°C (-139°F)
      Reservoir temperature reduction rates should not exceed -2°C/min (-3.6°F/min).
6. Simultaneously, reduce temperatures in inactive condenser section below -102°C (-152°F) and increase gas reservoir temperature to -40°C ±5°C over a period of approximately 25 minutes. Maintain subfreezing condenser temperatures for at least 12 minutes. If heat pipe fails, repeat Steps 1 through 6.
7. Increase inactive condenser temperatures to -40°C ±5°C over a period of approximately 20 minutes and simultaneously start reducing gas reservoir temperature to -95°C over a period of 180 to 200 minutes. At no time should the reservoir cooling rate exceed -2°C/min. If heat pipe fails, repeat Steps 1 through 7.
8. Repeat Steps 6 through 8.
The load on the heat pipe for the tests are 1.25 centimeters and 130 watts, respectively.

The tilt on the pipe is meant to minimize the contribution of excess fluid inventory to the pumping capacity of the pipe and to compensate to a certain extent for that component of the buoyancy force vector tending to keep the bubbles against the upper walls of the arteries.

Although heat loads over 90 watts are known to be in excess of the heat pipe one-artery capacity, the higher heat load of 130 watts is chosen in order to enhance the migration of bubbles toward the evaporator by virtue of higher fluid flow rates inside the arteries.

Steps 6 through 8 of an idealized freezing/thawing test cycle are shown graphically in Figure 3-3 where segments (B-C), (B-D), and (B-E) represent different condenser warmup rates. These steps are described in conjunction with the figure in what follows.

Figure 3-3. Idealized Freezing/Thawing Cycle "A"
By turning off the heaters on the condenser plate, the inactive condenser section is allowed to cool from -40°C (Point A) to below the freezing point of methanol (B) at a rate of approximately -4°C/min. Simultaneously, heat is applied to the gas reservoir to raise its temperature from about -95°C to -40°C in approximately 25 minutes. During this period of the test cycle, the advancement of the gas front toward the evaporator resulting from reduced condenser temperatures is further enhanced by the increasing temperature of the gas reservoir, the net effect of the above being to inactivate temporarily a portion of the condenser to allow the liquid in the arteries to freeze. Subfreezing temperatures in the inactive condenser are maintained for ten minutes to ensure complete freezing inside the arteries.

Failure of the heat pipe during condenser freezing could be attributed to be the result of either suction freezeout or the bubbles mechanism. If this event is repeatable in every cycle, then artery depriming will be, with a high degree of certainty, due to suction freezeout. On the other hand, sporadic failure of the heat pipe during condenser freezing will be consistent with the recognized statistical nature of the bubbles mechanism. If no evidence of heat pipe burnout is observed through the end of the freezing process, the test cycle continues with Step 7, otherwise, Steps 1 through 6 are repeated.

During Step 7, (e.g., B-C) power is applied to heaters on the condenser plate to thaw the inactive condenser and raise its temperature in approximately 25 minutes to -40°C where it is held within ±5°C for the remainder of the 240-minute test cycle. About the time the inactive condenser starts to thaw (B+), cooling of the gas reservoir is initiated and allowed to continue through the end of the cycle (F) at an average rate of 0.3°C/min. Reservoir cooling rates are not permitted to exceed 2°C/min to preclude the sudden cooling mechanism from taking place.

During this period of the test cycle, (e.g., B-C) rising condenser temperatures will cause the gas/vapor front to advance toward the reservoir end of the condenser, thawing the previously inactive section of the pipe and presumably engulfing into the active section bubbles that were generated during the freeze/thaw process. The simultaneous and continuous reduction of the gas reservoir temperature will have two significant effects
on the bubbles mechanism. First, it will cause a continuous movement of the gas front toward the reservoir during which additional bubbles will be induced into the active portion of the heat pipe where their migration toward the evaporator is enhanced. Second, reservoir cooling will tend to diminish the pressurization of the heat pipe resulting from increasing condenser temperatures and will cause subsequently, after the condenser reaches a steady state, (e.g., C) a continuous decrease of the total pressure in the heat pipe.

Exploratory analyses using the spherical bubble model indicate that pressure reduction in the heat pipe environment can significantly increase bubbles lifetimes. Longer bubble lifetimes are certainly a factor enhancing the potential of the bubble mechanism to deprime the arteries.

A heat pipe failure observed about the time the inactive condenser undergoes thawing (B+) could be attributed to either the postulated freezing blowby or the bubbles mechanism. The fact that artery depriming due to freezing blowby is not statistical in nature requires in order to consider the freezing blowby mechanism the culprit, that failures during the thawing process (B+) occur at every cycle. Sporadic heat pipe failures during the thawing process or at any other time during the test cycles will be considered to be the result of bubble migration and growth in the arteries. If no evidence of heat pipe burnout is observed by the end of the 240-minute test cycle, (F) Steps 6 and 7 are repeated for the remainder of the 120-hour continuous test period.

3.3 TEST RESULTS

The cyclic tests on the SNO09 heat pipe were performed from April 14 to April 19, 1980. A total of twenty seven partial and complete cycles were achieved during which seventeen burnouts or heat pipe failures were observed. A summary of the test results is presented in Figure 3-4. It shows the temperatures at nine different locations along the heat pipe at the end of the cooling period (Condition 'A') and those observed near the time of burnout or at the end of a cycle (Condition 'B').

The tests commenced at 0600 hours when the heat pipe was tilted 1.25 cm and 130 watts were applied to the evaporator block. After having established that both arteries were primed, cooling of the condenser was
## Table: Summary of SNO09 Heat Pipe Freeze/Thaw Cyclic Tests

<table>
<thead>
<tr>
<th>RUN S/N</th>
<th>CONDITION 'A'</th>
<th>CONDITION 'B'</th>
<th>TIME FROM 'A' to 'B'</th>
<th>MAXIMUM C.W.R.*</th>
<th>COMMENT/OBSERVATION</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>TEMPERATURES AT END OF COOLING PERIOD</td>
<td>TEMPERATURES AT BURNOUT OR AT END OF CYCLE</td>
<td>(MIN)</td>
<td>(C/Min)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>SENSOR TEMPERATURE (°C)</td>
<td>SENSOR TEMPERATURE (°C)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>-40 - 4 -132 -121 -55 6 31 32 34</td>
<td>-46 - -79 -43 - -32 -38</td>
<td>11</td>
<td>5.0</td>
<td>Burnout</td>
</tr>
<tr>
<td>2</td>
<td>-46 -125 -146 -140 -96 19 29 31 33</td>
<td>-96 -109 -103 -94 4 27 29 31 34</td>
<td>190</td>
<td>5.0</td>
<td>Normal Cycle</td>
</tr>
<tr>
<td>3</td>
<td>-48 -103 -127 -121 -73 10 29 31 33</td>
<td>-44 -101 -71 -101 -44 21 31 34 35</td>
<td>38</td>
<td>12</td>
<td>Burnout</td>
</tr>
<tr>
<td>4</td>
<td>-40 -101 -119 -111 -59 21 31 32 34</td>
<td>-36 -109 -51 -13 31 36 37 39</td>
<td>12.5</td>
<td>5.3</td>
<td>Burnout</td>
</tr>
<tr>
<td>5</td>
<td>-33 -136 -131 -86 21 31 32 34</td>
<td>-39 - -87 -72 8 31 34 34 36</td>
<td>10</td>
<td>5.3</td>
<td>Burnout</td>
</tr>
<tr>
<td>6</td>
<td>-37 -123 -124 -122 -75 9 29 32 33</td>
<td>-91 -102 -40 -41 2 27 29 31 34</td>
<td>180</td>
<td>2.2</td>
<td>Normal Cycle</td>
</tr>
<tr>
<td>9</td>
<td>-51 -126 -128 -122 -75 11 28 31 32</td>
<td>-57 -68 -69 -41 23 30 33 34 36</td>
<td>45</td>
<td>2.2</td>
<td>Burnout</td>
</tr>
<tr>
<td>12</td>
<td>-49 -132 -133 -126 -82 -6 28 31 32</td>
<td>-51 -103 -106 -91 -29 23 31 32 35</td>
<td>22</td>
<td>1.4</td>
<td>Burnout</td>
</tr>
<tr>
<td>14</td>
<td>-48 -126 -127 -121 -74 11 28 31 32</td>
<td>-41 -117 -118 -86 18 29 31 32 36</td>
<td>12</td>
<td>0.8</td>
<td>Burnout</td>
</tr>
<tr>
<td>15</td>
<td>-54 -107 -121 -100 -60 23 29 31 33</td>
<td>-53 -72 -69 -38 23 31 33 34 36</td>
<td>70</td>
<td>0.8</td>
<td>Burnout</td>
</tr>
<tr>
<td>18</td>
<td>-48 -105 -124 -113 -71 13 28 31 32</td>
<td>-94 -57 -39 16 27 29 31 33 33</td>
<td>275</td>
<td>0.4</td>
<td>Normal Cycle</td>
</tr>
<tr>
<td>23</td>
<td>-79 -131 -134 -118 -73 21 26 28 30</td>
<td>-71 -47 -34 3 29 32 33 34 36</td>
<td>15</td>
<td>15</td>
<td>Power off after Freezing/Power on after Thawing</td>
</tr>
<tr>
<td>24</td>
<td>-68 -147 -154 -147 -102 -26 21 28 30</td>
<td>-64 -51 -39 8 27 31 33 34 36</td>
<td>183</td>
<td>0.4</td>
<td>Normal Cycle</td>
</tr>
<tr>
<td>25</td>
<td>-46 -99 -118 -105 -54 18 29 31 33</td>
<td>-60 -73 -65 -34 23 29 30 31 33</td>
<td>180</td>
<td>0.4</td>
<td>Normal Cycle</td>
</tr>
<tr>
<td>26</td>
<td>-40 -100 -120 -113 -69 12 29 32 34</td>
<td>-71 -124 -126 -113 -66 21 28 29 31</td>
<td>30</td>
<td>0.2</td>
<td>Burnout</td>
</tr>
<tr>
<td>27</td>
<td>-69 -121 -124 -111 -63 3 27 29 31</td>
<td>-84 -73 -63 -31 24 29 31 32 34</td>
<td>165</td>
<td>0.2</td>
<td>Burnout when C.W.R increased from 0.19 to 5 C/Min.</td>
</tr>
</tbody>
</table>

*C.W.R.: Condenser Warm-up Rate (dT/dt)*

**Time from end of warm-up period.

Figure 3-4. Summary of SNO09 Heat Pipe Freeze/Thaw Cyclic Tests
initiated by turning the radiator heaters off. This process continued until subfreezing temperatures in the inactive condenser section, as indicated by temperature sensors 10, 11 and 12, were realized. These sensors showed minimum temperatures of -120°C, -132°C and -114°C respectively prior to the initiation of the condenser warm-up period. Approximately eleven minutes after the radiator heaters were turned on, the heat pipe failed. About the time of the failure, the condenser warm-up rate (CWR) was about 9°F/min and the minimum temperature in the condenser section, shown by sensor 11, was -79°C which is 19°C above the freezing point of methanol, thus well past point B in Figure 3-3.

Subsequent to burnout, the heat pipe was leveled and allowed to reach close to isothermal conditions over a period of two hours. The pipe was then tilted 1.25 cm and 130 watts were applied to the evaporator, but the heat pipe failed to sustain the load. The above priming procedure was repeated with similar results.

The inability of the heat pipe to hold the load after two priming attempts was postulated to be the result of the presence of residual bubbles in the arteries. It was decided then to modify the procedure to prime the pipe, by requiring that the evaporator end be raised 61 cm for one minute before leveling the pipe. The high tilt would presumably eject all the liquid and residual bubbles from the arteries. After the pipe was leveled for thirty minutes, 130 watts were applied to the evaporator raised 1.25 cm. This time priming of the arteries was verified and the second cycle was started. The latter procedure for priming the heat pipe was followed through the rest of the test period with successful results.

Cycle number 2 was the first complete normal cycle and was followed by three burnouts observed during the condenser warm-up periods of cycles 3, 4 and 5, at which times the CWR was, approximately 5°C/min (9°F/min).

The partial history of selected temperatures along the heat pipe during normal cycle 2 is shown in Figure 3-5. It can be seen that prior to condenser warmup the ice front almost reaches the locations of sensor 9, i.e., approximately 18 inches of the condenser end were frozen. During the warmup period of the cycle the figure shows the gas front moved close to sensor 10 which indicates that a portion of the previously frozen condenser became part of the active condenser section.
Figure 3-5: Temperature Histories During SN009 Heat Pipe Cyclic Test No. 2
The large proportion of failures, four out of five, was attributed to rapid thawing and warming of the inactive condenser, events which were originally postulated to result in the formation of bubbles inside the arteries and in their rapid induction into the active portion of the condenser before the bubbles could vent by diffusion. Once in the active portion of the pipe, these bubbles would grow and deprime the artery.

A typical temperature history of the four cycles during which depriming was observed is that of cycle 4 shown in Figure 3-6. As shown, the ice front was located somewhere between sensors 9 and 10 at the end of the cooling period of the test cycle. During the warmup period depriming is observed apparently at the time the gas front reached the previous location of the ice front. The fact that depriming occurred approximately 9 minutes after the inactive condenser completely thawed, the possible involvement of freezing blowby in these heat pipe failures can be discounted, therefore it can be argued that depriming can only be the result of bubbles.

In order to determine the effect of condenser warm-up rates on the frequency of heat pipe failures, the CWR during the next six cycles was reduced an average of fifty percent. The results were that the heat pipe performed normally during two consecutive complete cycles, cycles 6 and 7, at the reduced CWR of about 2.2°C/min. The following four cycles, however, resulted in burnouts during condenser warmup at rates varying from 2.2°C/min to 3.6°C/min. The variation in these CWRs was not intentional but rather the result of the limited heater control capability of the test assembly.

The typical partial temperature history of normal cycles 6 and 7 is that of cycle 6 shown in Figure 3-7. The temperature history of cycle 11 shown in Figure 3-8 is similar to that of cycles 8 and 10. As in most previous cycles, the ice front can be seen located between sensor 9 and 10 at the end of the cooling period. Figure 3-8 shows however, that depriming occurred under significantly different condenser conditions in most cycles at the lower condenser warmup rate: 1) the inactive condenser was still partially frozen (last 12 inches) and 2) temperatures of sensors 8 and 9 indicate the gas/vapor front never reached the previously frozen region.

Because the condenser was not completing the thawing process at the time of burnout, freezing blowby can be discounted as the cause of pipe
Figure 3.6: Temperature Histories During SN009 Heat Pipe Cyclic Test No. 4
Figure 3-8. Temperature Histories During SN009 Heat Pipe Cyclic Test No. 11
failure which, then can be attributed only to be the result of bubbles. However, the fact that portions of the previously frozen condenser section were never incorporated into the active heat pipe section, a condition that was postulated to be necessary to induce bubbles into the evaporator, seems to indicate that other mechanisms such as Marangoni Flow were involved to move bubbles into the active portion of the condenser where further growth or movement would be expected.

It was postulated that depriming due to bubbles released in the inactive section occurred as follows. During the cooling period the freezing front moves toward the evaporator expelling bubbles which are then locked inside the ice and building stress on the arteries due to suction freezeout. This process continues until the end of the cooling period when the imposed stress on the arteries due to freezeout reaches a maximum level. This increase in stress is accompanied by a reduction of the hydrodynamic stress which results from shrinkage of the effective length due to advancement of the gas/vapor front toward the evaporator.

This imposed stress on the arteries due to suction freezeout is either overcompensated by the reduced hydrodynamic stress or the unused pumping capacity of the artery, for no heat pipe failures were observed during the cooling periods of the test cycles. During the warmup period, the advancement of the gas/vapor front toward the reservoir end results in increased hydrodynamic stress as the effective length expands, and in the propagation of a conduction heat wave which causes the ice front to recede and release liquid and bubbles upon thawing. Although the release of the liquid reduces the stress that came from suction freezeout, there remains sufficient back stress on the arteries in the gas-blocked region to induce growth of the released bubbles. These bubbles agglomerate and then reach the active section due to continuing growth and/or Marangoni Flow.

Whereas depriming during cycles 8, 10, and 11 occurred 10 to 12 minutes into the warmup period before the gas/vapor front reached previously frozen sections, depriming in cycle 9, as shown in Figure 3-9, occurred approximately 45 minutes into the warmup period, 25 minutes after the condenser completely thawed. An interpretation of the data seems to indicate that bubbles first released about the previous location of the ice front somewhere between sensors 9 and 10 were subsequently incorporated into the
Figure 3-9: Temperature Histories During SN009 Heat Pipe Cyclic Test No. 9.
active portion and caused depriming when the gas/vapor front reached that location 45 minutes later.

During test cycles 12 through 18, the heatpipe inactive condenser was warmed up at even lower rates ranging from 0.4°C/min to 1.7°C/min. Burnouts were observed in all these cycles, except in cycle 18 which had the lowest CWR of 0.4°C/min.

The temperature history of cycle 13 is shown in Figure 3-10. It can be seen that depriming occurred before the gas/vapor reached previously frozen sections and while the last 12 inches of the condenser were below the freezing point. Similar observations can be made of deprimings observed during cycles 12, 14, 16 and 17.

Depriming during cycle 15, as shown in Figure 3-11, occurred under different conditions. It can be seen that when the gas/front apparently reached previously frozen areas a burnout was triggered. At this time the condenser had been completely thawed for approximately 40 minutes.

Normal freezing cycle 18, was followed by three cycles during which the inactive condenser was rapidly cooled to temperatures just above the freezing point and then warmed at rates as high as 6.7°C/min. The idealized nonfreezing cycle is described graphically in Figure 3-12. These nonfreezing test cycles were performed to support the hypothesis that the frequency of failures due to bubbles could be greatly diminished if freezing and thawing did not occur in every cycle. The fact that the heatpipe evidenced normal operation during these cycles tends to support the above. The typical temperature history of a nonfreezing test cycle is that for cycle 19 shown in Figure 3-13.

During test cycles 22 through 24, the inactive condenser was rapidly cooled below the freezing point with the heatpipe operating under 130 watts. Subfreezing condenser temperatures were maintained for at least ten minutes subsequent to which the evaporator heaters were turned off and the inactive condenser was rapidly warmed. When the inactive condenser temperatures reached about -40°C the evaporator heaters were turned on again. The idealized cycle is described graphically in Figure 3-14. A burnout was observed during the second cycle, cycle 23, 15 minutes after the heat load
Figure 3-12. Idealized Nonfreezing Cycle "B"
Figure 3-14. Idealized Freezing/Thawing Cycle "C"

was reestablished. This can be seen in Figure 3-15 which shows that depriming occurred apparently when the gas/vapor front reached the previously frozen condenser section.

In the last three cycles of the test program, the heat pipe, under a continuous load of 130 watts, was subjected to condenser warmup rates ranging from 0.2C/min to 0.4C/min. Cycle 25 with the higher CWR of 0.4C/min was normal. Burnouts were observed in the next two cycles. The last heat pipe failure during cycle 27, however, is worth noting since it occurred 165 minutes into the warmup period when the CWR was increased from 0.2C/min to 5C/min. The temperature history of this cycle is shown in Figure 3-16. Although the cooling period of the cycle is not shown in this figure, Figure 3-2 indicates the ice front prior to warmup was located somewhere between sensor 9 and 10. Figure 3-16 shows the gas/vapor front at the end of the slow warmup period was located between sensors 8 and 9. As the warmup rate was increased, it can be seen that the gas front rapidly moved between sensors 9 and 10 seemingly reaching the previously frozen section and triggering a burnout.
Figure 3-15. Temperature Histories During SN009 Heat Pipe Cyclic Test No. 23.
Following this final burnout, the heat pipe was leveled and left undisturbed over the weekend. Fifty-five hours later, when 130 watts were applied to the evaporator, the heat pipe failed to hold the load, an event seemingly pointing to the survival of bubbles from the last freeze/thaw cycle as the probable cause of artery depriming. The temperature histories of cycles not shown in this section can be found in Appendix B.2.

3.4 SIGNIFICANCE OF TESTS RESULTS

The results of the cyclic tests on the SNO09 heat pipe clearly established that thawing the frozen condenser of an arterial heat pipe under high load is an operating mode that results in arterial failure with significant frequency. Out of twenty one freezing and thawing cycles under high load, sixteen cycles resulted in heat pipe failures. Although thawing the condenser under high load enhances the probability of artery depriming, it is not a necessary condition. This proposition is supported by the fact that out of three cycles in which the condenser was thawed under no heat load one cycle (number 23) resulted in artery failure subsequent to thawing when the heat load was reestablished on the pipe.

In twenty four freezing and thawing cycles no heat pipe failures occurred during condenser freezing. The lack of failures during freezing when the suction freezeout mechanism has its greatest potential to deprime the arteries seems to indicate that none of the observed failures during the test cycles can be solely attributed to this mechanism. Even though bubbles are known to be released during freezing they are immobilized inside the ice structure rendering a failure during freezing due to bubbles an unlikely event.

All the observed anomalies occurred during or following condenser warmup periods. The fact that the inactive condenser was either completely thawed or still frozen along a considerable length when deprimings occurred, it can be argued that the freezing blowby mechanism could not account for any of the heat pipe failures observed during the cyclic tests.

Since neither suction freezeout nor freezing blowby were the sole cause of any of the observed heat pipe failures, it can be concluded only that all the anomalies during the cyclic tests on the SNO09 heat pipe were the result of the freezing/thawing bubble mechanism.
The freezing and thawing process is required for bubbles to form inside the arteries. Periodic freezing and thawing replenishes the bubble population which enhances the probability of artery depriming due to bubbles. The results of three normal successive cycles without freezing supports the above.

The presence of bubbles in the arteries is necessary for depriming to occur. However, depriming does not necessarily occur when bubbles are present. This statement is supported by the fact that out of 24 freeze/thaw test cycles seven were normal (i.e., no failure occurred). These test results indicate that the occurrence of the observed heat pipe failures is random, which is consistent with the demonstrated statistical nature of the bubble mechanism.

During the freeze/thaw process bubbles are generated in the gas-blocked condenser section. It was postulated at the outset of the cyclic tests that the active condenser length had to expand into the previously frozen section. However, in roughly half the failures the active condenser length had not yet reached the previously frozen section.

The results of the tests seem to indicate that there exists some correlation between the condenser warm-up rate and the incidence of heat pipe failures. The reduction of the condenser warm-up rate used in the first five cycles by approximately 50 percent was followed by the only two consecutive normal cycles observed during the tests. In addition, during cycle 27 depriming occurred almost two hours into the warm-up period following a rapid and large increase in the condenser warm-up rate.

Deprimings observed during cycles with high warm up rates occurred after the condenser had completely thawed and soon after the active condenser length reached the previously frozen section. On the other hand, most heat pipe failures during cycles with low warm up rates occurred while portions of the condenser were still frozen and before the active condenser length ever reached the previously frozen section. The latter failures are attributed to movement of bubbles into the active condenser helped by suction freezeout and/or Marangoni flow.

The results of several unsuccessful attempts to reprime the heat pipe after the first freeze/thaw cycle indicate that a depriming does not clear
out all the bubbles. Consequently bubbles form during a single freeze/thaw process can result in repeated successive failures due to residual bubbles.

Numerous deprimings observed during the tests occurred 10 to 30 minutes after freezing conditions in the condenser ceased to exist. Following the last freezing and thawing test cycle, the SNO09 heat pipe deprimed 52 hours later when the normal heat load was reestablished. This heat pipe failure can only be attributed to be the result of bubbles generated during the last freeze/thaw cycle of the test program.

These results indicate that ice-generated bubbles can last a long time, and therefore freezing at the very time of the CTS anomalies was not necessary. Thus the anomaly on day 253 can be attributed to ice-generated bubbles from day 252 or earlier.
4.0 CONCEPTS TO AVERT ARTERY DEPRIMING

4.1 NONCONDENSIBLE CONTROL GAS SELECTION

Unquestionably the simplest and most effective concept to avert arterial depriming is the selection of a control gas that would preclude the formation of bubbles when the liquid methanol undergoes freezing and thawing. This concept is unfortunately unrealizable, for the freeze/thaw process, being a degasifying process, will result always in the formation of bubbles regardless of the noncondensible control gas selected.

Nucleation experiments performed during Phase I, showed that a large number of minute bubbles are initially released during freezing and thawing. Most of them rapidly disappear owing to their size, but others coalesce forming fewer but larger bubbles whose lifetimes strongly depend on their sizes. Larger bubbles have longer lifetimes which enhance the probability of further coalescence which results in bubbles of even longer lifetimes which can then survive until the gas/vapor front moves into previously frozen sections or until suction freezeout and/or Marangoni flow succeed in making them grow and transporting them into the active section of the heat pipe.

The above observations indicate that reducing bubble lifetimes by proper selection of a control gas can decrease to a certain extent the probability of bubble coalescence and of their induction into the active section of the heat pipe. The results of the bubble lifetime analysis in Section 2.0, show, for example, that a spherical CTS control gas bubble (90% N₂/10% He) of 0.8 mm in diameter (i.e., half the CTS artery diameter) under some assumed conditions has a lifetime of about 19 hours compared to a lifetime of less than one hour for a methane bubble of the same size. The rapid dissolution of methane bubbles suggests the selection of methane for the control gas.

Although the selection of a new control gas, such as methane, will result in bubble lifetimes significantly shorter than in the CTS heat pipes which will diminish to a certain extent the probability of depriming, this concept cannot guarantee that depriming will be averted. However, the use of methane in conjunction with other steps to avoid depriming should enhance the effectiveness of those steps.
4.2 OPERATIONAL PROCEDURE FOR START-UP OF A FROZEN CONDENSER

Since the bubbles leading to arterial depriming are formed by a freeze/thaw process, the avoidance of freezing will avert artery depriming. Selection of sun angle and radiator radiation properties or the use of heaters could prevent freezing in many missions. However, in missions to outer planets, avoidance of freezing may be costly or even impossible. Even when the condenser has been frozen, it is possible to thaw the heat pipe and restore it to a primed and serviceable state, provided proper procedure is followed.

The key to successful restoration of a previously frozen heat pipe is dissolution or venting of the freeze/thaw generated bubbles. Venting is discussed in Section 4.3. Dissolution requires leaving the condenser in a quiescent state somewhat above the freezing point for a time long enough to allow the bubbles to collapse. This time depends upon the control gas, the size of the largest bubble, and the pressure and temperature in the gas-blocked condenser. The advantage of methane as a control gas has been discussed. It has a high solubility-diffusivity product leading to more rapid bubble dissolution. Since the bubble agglomeration mechanism appears to be statistical, and it is not clear that a laboratory test at 1-g is a good indicator of agglomeration at zero-g, it is presently difficult to predict with certainty the size of the largest bubble and thus the time required for it to collapse. However, start-up can be attempted earlier than necessary merely to test whether a sufficiently long time has been allowed to elapse. If depriming occurs, the heat pipe is returned to a quiescent state for more time to pass. In the attempt at operation, at least one bubble was vented. Thus successive attempts at start up contribute to the bubble clearing process.

High pressure during the bubble collapse period hastens it. With an actively controlled heat pipe, the gas reservoir and evaporator could both be heated somewhat to raise the gas-blocked condenser pressure. It should be noted that the restart procedure would have to be used each time a segment of the gas-blocked condenser freezes and is thawed.
4.3 MECHANICAL MODIFICATIONS

In a gas-controlled arterial heat pipe it has been found that a priming foil at the end of the evaporator is necessary for successful priming [5]. Without it, gas in the pipe can be trapped inside the artery during priming. With the priming foil and with a gentle heating of the evaporator bubbles trapped in the evaporator are transported to the foil and vented.

The priming foil shown in Figure 4-1 consists of a very thin metallic membrane with one or more holes no larger than the effective pumping pore diameter needed for the capillary flow in the active pipe. The size of the vent hole and the thickness of the foil must be such that the meniscus of a bubble of arterial size contacts the meniscus attached to the outer corners of the vent hole. Meniscus coalescence then ruptures the liquid.

Figure 4-1. Segment of an Artery with a Priming or Venting Foil
film bridging the vent hole and allows the gas to be vented. It should be noted that the heat load during venting must be less than the open-artery capacity of the heat pipe.

The same principle can be applied to the condenser region or even to the entire pipe [5] as long as the local stress does not exceed the capillary pumping capability of the artery at that point and time. Thus a location in the condenser (or perhaps even in the adiabatic section) can be identified where under restart heat load the local stress is below this limit. At that location a screen across the artery can be placed to filter out and collect small bubbles tending to move toward the evaporator. On the condenser side of the screen a priming foil would be placed (venting holes located one artery radius from the screen) to allow venting of the collected gas should the gas volume reach arterial size.

The bubble-filter/venting-foil concept should allow a normal start up from a frozen condenser state and permit continuous normal operation of the heat pipe even though the gas-blocked condenser is undergoing periodic freezing and thawing.
5.0 CONCLUSIONS

Evidence strongly suggests that bubbles generated through freezing and thawing of methanol in the gas-blocked condenser arteries were responsible for the CTS thermal anomalies. Out of the 24 freeze/thaw cyclic tests with the SNO09 heat pipe, 17 deprimings were observed. While some of the tests were conducted under conditions where probability of depriming was high, others were conducted where the probability was low, thus emphasizing a statistical variation in the freeze-thaw bubble mechanism.

Most often depriming occurred while the gas-blocked condenser was still frozen along a portion of its length but some occurred 10 to 30 minutes after thawing was completed. In one test depriming occurred 52 hours after thawing.

Analytical-numerical studies of bubble collapse show bubble lifetimes to be strongly dependent upon bubble size and control gas composition. The helium in the CTS control gas mixture causes long bubble lifetimes when the bubbles are of arterial size. Selection of methane as a control gas would significantly shorten bubble lifetime for equal sized bubbles.

At the present level of understanding, selection of methane for control gas cannot ensure avoidance of bubble depriming. Prevention of freezing in the gas-blocked condenser by active or passive control should avert depriming. Should avoidance of freezing be inconvenient or not feasible, either installation of a bubble-filter/priming foil element at the condenser-adiabatic section interface or adoption of an appropriate thawing and bubble clearing procedure would probably allow successful heat pipe operation.
6.0 REFERENCES


APPENDIX A.1
EXCESS SKEW IN LOAD PARTITIONING

The three heat pipes of the CTS VCHP system are designed to turn on sequentially so as to balance the load between them at full-on design conditions. However, a build up of tolerances on gas inventory during manufacture, and very low sink temperature during equinox conditions may result in the load on heat pipe No. 1 exceeding its capacity before Nos. 2 and 3 take up a significant share. Failure of No. 1 by the above mechanism or by any other could then result in a rapid transfer of load to No. 2, etc., perhaps yielding a "domino effect" failure of the whole system.

The first step in investigating this was to further review the flight data to search for regularity in the depriming sequence associated with the anomalies. Toward this end, NASA LeRC personnel provided computer plots of pertinent flight data for subsequent review.

Experimentally, attempts were made to deprime the SN009 heat pipe with rapid load increases, simulating rapid sequential load transfer as pipes deprime.

The tasks performed in this investigation are summarized below.

Flight Data Analysis to Determine Depriming Sequence

Computer plots and tabulations of CTS telemetered data were provided for days 75, 82, 89, 101 and 253 of 1977. Day 89 was a so-called "normal" day whereas the others were the four anomaly days.

Flight data had already been examined in a previous study effort. However, the new data included overlay plots of all pertinent data on single pages, and values of $\Delta T_{3-2}$ and $\Delta T_{2-1}$ in addition to $\Delta T_{3-1}$ provided earlier. It was hoped that this data format might shed additional light on the system behavior, particularly the sequence in which the heat pipes deprimed on the anomaly days.

Unfortunately, the telemetry increment in the temperatures was 1.8C to 1.9C, which is of the same magnitude as the normal values of $\Delta T$ between pipes as well as the changes in temperature which accompany depriming. Consequently, it is very difficult to distinguish between
actual physical phenomena and telemetry noise. In view of this limitation, the data tentatively suggests the following:

Day 75: It appears that heat pipe No. 1 deprimed first, yielding increased values for $\Delta T_{2-1}$ and $\Delta T_{3-1}$. Several hours later heat pipe No. 2 deprimed, followed almost immediately by heat pipe No. 3 and the sudden rise in body temperature.

Day 82: The data is particularly unclear this day, but as with day 75, it appears that heat pipe No. 1 failed first, followed by Nos. 2 and 3.

Day 89: No anomaly occurred on this day. However, the data suggests that heat pipe No. 2 was deprimed, causing No. 3 to carry a greater load.

Day 101: The data indicates the failure sequence was 1:2:3, as with day 75.

Day 253: In this case, heat pipe No. 3 did not even "turn-on" until the anomaly began, and apparently could not prevent it. Thus, it had to be deprimed from the start. In fact, the data suggests that heat pipes 2 and 3 deprimed first, with the anomalous increase in body temperature corresponding to the depriming of No. 1.

Specific interpretation of the depriming sequences is covered in the SNO09 heat pipe test discussion that follows. However, one item of note bears discussion. If bubble nucleation within the arteries is a cause of the anomalies, then the statistical nature of the mechanism should yield one heat pipe failed occurrences far more often than two or three heat pipe failed occurrences. Day 89, in which No. 2 appears deprimed, while Nos. 1 and 3 are primed, may be such a day.

Rapid Increases in Load Tests on SNO09 Heat Pipe

Extensive testing was performed with the SNO09 heat pipe. The first series of tests were to determine whether there exists significant inertial effects on heat pipe capacity which could contribute to a "domino effect" failure of all three CTS pipes. That is, if heat pipe No. 1
deprimed, most of the load it had carried would be suddenly transferred to No. 2, etc. If the capacity of a heat pipe were substantially lower for the sudden application of load than for gradual application (due to fluid inertia), the probability of a sequential failure of all three pipes would be enhanced.

In order to maximize the rate of load transfer into the heat pipe evaporator, attached heater blocks normally used to simulate the thermal mass of a heat source were removed and tape heaters were placed on the heat pipe saddle over a 30-cm long section of the pipe which was then properly insulated. Testing was done for both high (-18C) and low (-96C) sink temperatures with the heat pipe evaporator elevated 2.5 cm. Heat loads ranging from 100 to 150 watts were rapidly applied resulting in no heat pipe failures. The results indicated essentially no rate effect attributable to fluid inertia. This argued against the postulated "domino effect" failure of the whole VCHP system.
APPENDIX A.2.1

INSTANTANEOUS HEAT FLOW AFTER HEAT PIPE RESERVOIR ECLIPSE
INTRODUCTION

When a gas-controlled heat pipe's reservoir experiences an eclipse, the gas partial density within the reservoir rises, and the gas front retreats until the evaporator temperature drops sufficiently to establish a new equilibrium. During the transient, the evaporation from the warm end may exceed the capacity of the axial wicking, and arterial depriming may occur. It is desired to estimate the instantaneous heat flow during such a transient for comparison to the burnout heat flow capacity.

RESERVOIR COOLING RATE

The reservoir temperature versus time may be known from measurement. It is set by a heat balance between the surrounds and the thermal capacity. Let the mass of the reservoir be M and its specific heat c. Conductive coupling to the radiator is KA/L, and insulation is modelled with massless radiation shielding.

\[-m c \frac{dT}{dt} = A_r F \left( \sigma T^4 - \frac{q}{\epsilon} \right) - \frac{KA}{L} [T - T_c] \] (1)

Before eclipse q is high, and the quasi-equilibrium temperature is given by setting the left-hand side to zero. After q is reduced, T approaches Tc at the rate given by Eq. (1).
FLOW VELOCITY INTO RESERVOIR

When a wicked reservoir of volume V cools at a rate of \(-dT/dt\), the vapor partial pressure inside the reservoir falls. Since the total pressure is fixed at the evaporator end (neglecting vapor flow pressure drop), the partial pressure of the gas rises. Furthermore, the gas temperature falls (assuming thermal equilibrium). These two effects increase the gas partial density. The rate at which the partial density times the volume rises must come from the flow of gas from the heat pipe radiator into the reservoir.

\[
\frac{d}{dt} (\rho g V) = \rho g c V A_c
\]  

(2)

\[
\rho g = \frac{P - P_v(T)}{R g T}
\]

(3)

Differentiating Eq.(3) gives

\[
\frac{dp_g}{dt} = \frac{P - P_v(T)}{R g T} \frac{1}{T} \left( -\frac{dT}{dt} \right) + \frac{1}{R g T} \frac{dP_v}{dt} \left( -\frac{dT}{dt} \right) + \frac{1}{R g T} \frac{dP}{dt}
\]

Substituting into Eq.(2) then gives

\[
v = \frac{V}{\rho g c A_c} \left( \frac{1}{R g T} \right) \left\{ \frac{P - P_v(T)}{T} + \frac{dP_v}{dT} \left( -\frac{dT}{dt} \right) + \frac{dP}{dT} \right\}
\]

(4)

Let \(dP_v/dT\) be approximated from the Clausius - Clapeyron equation

\[
P_v = P_0 e^{-(\Delta h/RT_0)} \left( T_0/T - 1 \right)
\]

\[
\frac{dP_v}{dT} = P_v \left( \frac{\Delta h}{R T^2} \right)
\]
Then Eq. (4) becomes

\[
\dot{v} = \frac{v}{A_c} \left( \frac{T_c/T}{P-P_{vc}} \right) \left\{ \frac{[P + P_v (\Delta h/R_T - 1)] (-dT)}{T} + \frac{dP}{dT} \right\}
\]

(5)

The total pressure \( P \) is the vapor pressure at the evaporator temperature \( T_v \).

\[
\frac{dP}{dt} = \frac{dP_v}{dT_v} \left( \frac{dT_v}{dt} \right) = - \frac{dP_v}{dT_v} \left( \frac{-dT_v}{dt} \right)
\]

(6)

The evaporator temperature in turn is set by a heat balance upon it. The worst case occurs when the evaporator is well-coupled thermally to a large thermal mass. In this worst case \( dP/dt \) may fall only very slowly compared to the other terms. Measured data may be used to evaluate \( dT_v/dt \). A crude model is a one-capacity model.

\[
M_{ev} c_{ev} \frac{dT_v}{dt} = \dot{Q}_{elec} - \dot{Q}_{elec} - \dot{Q}_{hp} - A \int (oT_v^4 - oT_s^4)
\]

(7)

The last term in the equation models heat leak to surrounds at temperature \( T_s \).

HEAT FLOW

The instantaneous heat flow through the pipe \( \dot{Q}_{hp} \) is fixed by the length of active condenser and the rate of advancement of the gas front.

\[
\dot{Q}_{hp} = \dot{Q}_{static} + \dot{Q}_{dynamic}
\]

(8)

The static load is that caused by condensation on the gas-free condenser wall

\[
\dot{Q}_{static} = \epsilon P L_c \eta (oT_v^4 - \frac{a}{c} q_c^-)
\]

(9)
where the length $L_c$ is fixed by a gas inventory $m_g$

$$m_g = \rho_g V + r_{gc}A_c (L - L_c) \quad (10)$$

The dynamic load is proportional to the velocity $v$ at which the front advances

$$\dot{Q}_{\text{dynamic}} = \rho_f c_f A_f^\eta \left( T_V - T_c \right) \quad (11)$$

where $\rho_f$ is the condenser fin density, $c_f$ its specific heat, $A_f$ its area (volume per unit length), and $\eta$ is not the radiating fin effectiveness, because of the fourth power dependence of the black body radiosity. The appropriate value of $\eta$ is between the radiating fin effectiveness and the fourth root of that value, depending upon the ratio of $(T_c/T_V)^4$.

**EXAMPLE**

For example consider the CTS heat pipe 1 on Day 75. The following data are available

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$t$ min</td>
<td>$T_V$ °C</td>
<td>$T_c$ °C</td>
<td>$T$ °C</td>
</tr>
<tr>
<td>460</td>
<td>27.5</td>
<td>-93</td>
<td>-68</td>
</tr>
<tr>
<td>470</td>
<td>28</td>
<td>-94</td>
<td>-71</td>
</tr>
<tr>
<td>480</td>
<td>29</td>
<td>-95.5</td>
<td>-74</td>
</tr>
</tbody>
</table>
From these data one can estimate $-dT/dt = 18$ K/hr and $dT_v/dt = 4.5$ K/hr.

The following vapor pressure data pertain:

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Pressure</th>
<th>Latent Heat</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_v$</td>
<td>300</td>
<td>$1.15 \times 10^6$</td>
</tr>
<tr>
<td>$T$</td>
<td>205</td>
<td>$1.2 \times 10^6$</td>
</tr>
<tr>
<td>$T_c$</td>
<td>180</td>
<td>$1.2 \times 10^6$</td>
</tr>
</tbody>
</table>

For a 90% $N_2$ 10% He mixture the mean molecular weight is 25.6 and $R_g = 8314.3/25.6 = 325$ J/kg K. The density of the gas in the gas-blocked condenser is

$$\rho_{gc} = \frac{p_{gc}}{RT} = \frac{19500}{(325)(180)} = 0.333 \text{ kg/m}^3$$

Other parameters needed are

$$\frac{\Delta h}{RT} \bigg|_{T=205} = \frac{1.2 \times 10^6}{(205)(205)} = 22.51$$

$$\frac{\Delta h}{RT} \bigg|_{T=300} = \frac{1.15 \times 10^6}{(260)(300)} = 14.74$$

The pertinent parameters of the pipe itself are understood to be as follows:

Reservoir Volume $V = 8.70$ inch$^3 = 1.426 \times 10^{-4}$ m$^3$

Pipe Vapor Area $A_c = 0.1245$ in$^2 = 8.03 \times 10^{-5}$ m$^2$
Equation (5) then gives

\[
v = \frac{1.426 \times 10^{-4}}{8.03 \times 10^{-5}} \frac{(180/205)}{(19500 - 1.5) \left[ \frac{19500}{205} + \frac{13}{205} (22.5 - 1) \right] + \frac{19500}{300} (14.74) (4.5)}
\]

\[
v = (1.78) \left(4.50 \times 10^{-5}\right) \left[95.1 + 1.36\right] (18) + (958) (4.5)
\]

\[
v = 8 \times 10^{-5} \left(1736 + 4311\right) = 0.484 \text{ m/hr} = 1.59 \text{ feet/hr}
\]

Note that, in this example, the fall of the vapor pressure in the reservoir was a negligible consideration but the rise of vapor pressure in the evaporator was quite significant.

For purposes of an estimate \(nA_F\) is taken to be 0.04 inches by 12 inches. Thus Equation (11) gives

\[
\dot{Q}_{\text{dynamic}} = (174) (0.21) (0.04/12) (12/12) (1.59) (300 - 180) (1.8)
\]

\[
\dot{Q}_{\text{dynamic}} = 41.8 \frac{\text{BTU}}{\text{hr}} = 12.25 \text{ Watts}
\]

DISCUSSION

While a rapid decrease in reservoir temperature has the potential of adding a significant dynamic heat flow to the static load, the example worked here indicates that the decrease must be rapid, much greater than the 18K/hr value used in the example.
APPENDIX A.2.2

INSTANTANEOUS HEAT FLOW AFTER HEAT PIPE RADIATOR ECLIPSE
Introduction

When a gas-controlled heat pipe's radiator experiences an eclipse, the heat flow in the pipe increases until the evaporator cools to a new equilibrium operating condition. During the transient, the evaporation from the warm end may exceed the capacity of the axial wicking, and arterial depriming may occur. It is desired to estimate the instantaneous heat flow during this transient for comparison to the burnout heat flow capacity. This memo is a companion to one that explored the instantaneous heat flow after heat pipe reservoir eclipse.

A Simplified Thermal Model.

To explore the magnitude of the expected effect and to show the effect of the major parameters, a simplified model is proposed. In this model the heat loss out of the radiator is taken to be

$$Q_r = F_{scr}WL\left(\sigma T_v^4 - \sigma T_e^4\right) e_f \quad (1)$$

where $F_{scr}$ is the so-called script - F radiant transfer factor (equal to emissivity for an isolated radiator) $W$ is the width or perimeter of the panel per pipe (including both sides if both sides radiate), $L$ is the active length of the vapor in the radiator, $T_v$ is the vapor temperature, $T_e$ is the environmental temperature

$$T_e = \left[\frac{d}{F_{scr}} \right]^\frac{1}{4} \quad (2)$$

and $e_f$ is the fin efficiency. The vapor filled length is assumed to be given by

$$L = L_{tot} \frac{T_v - T_o}{\Delta T} \quad (3)$$

where $L_{tot}$ is the total length, $T_o$ is the turn-on point, and $\Delta T$ is the control
band width. Equation (1) may be factored into the form

\[ Q = \frac{e_F}{\kappa_s} \frac{W_0 \sigma}{L_0} (T_v^2 + T_e^2) (T_v + T_e) (T_v - T_e) \]

and Eq. (3) may be introduced to write

\[ \dot{Q} = \frac{1}{R_T} \left( \frac{T_v}{\Delta T} \right) (T_v - T_e) \] (4)

where

\[ R_T = \frac{1}{\frac{e_F}{\kappa_s} \frac{W_0 \sigma}{L_0} (T_v^2 + T_e^2) (T_v + T_e)} \] (5)

This latter parameter is insensitive to mild excursions in \( T_v \) and \( T_e \)

since the absolute temperatures appear. In contrast Eq. (4) varies rapidly

with \( T_v \), because \((T_v - T_o)\) and \((T_v - T_e)\) change significantly when \( T_v \) changes,

particularly the former quantity, and \((T_v - T_e)\) changes when \( T_e \) changes.

Note that this simplified model neglects the thermal capacity of the

radiator on the grounds that it is small compared to that of the heat source.

This neglect leads to an overprediction of thermal shock on the heat pipe from

sudden changes in environmental temperature \( T_e \).

The heat source is modelled as a single lumped capacity of mass \( m \) and

specific heat \( c \).

\[ mc \frac{dT_h}{dt} = Q_e - \frac{1}{R_e} (T_h - T_v) \] (6)

where \( T_h \) is the heat source temperature, \( Q_e \) is the net electrical power

dissipation, and \( R_e \) is the thermal resistance into the heat pipe evaporator.

It is the heat flow into the evaporator that concerns us

\[ \dot{Q} = \frac{1}{R_e} (T_h - T_v) \] (7)

The heat source thermal capacity \( mc \) is assumed to be sufficiently large.
that $T_h$ remains constant during the time that the radiator and gas front readjust. In this respect the heat transfer through the heat pipe is assumed quasi-steady.

**Analysis Based Upon the Simplified Model**

The model in its bare essentials consists only of Eqs. (4) and (7). The heat pipe behavior is determined by the evaporator resistance $R_e$, the radiator resistance $R_r$, the turn-on point $T_0$, and the control band width $\Delta T$. The parameters $R_e$ and $R_r$ may be inferred from observed temperatures at, say, the full-on operating point. Equation (4) shows that

$$R_r = \frac{T_{v,full} - T_e}{Q_{full}} = \frac{T_0 - T_e + \Delta T}{Q_{full}}$$  \hspace{1cm} (8)

and from Eq. (7)

$$R_e = \frac{T_{h,full} - T_{v,full}}{Q_{full}}$$  \hspace{1cm} (9)

Hence

$$\frac{R_e}{R_r} = \frac{T_{h,full} - T_{v,full}}{T_{v,full} - T_e} = r$$  \hspace{1cm} (10)

Equating Eqs. (4) and (7) and rearranging gives

$$\Delta T(T_h - T_v) = r(T_v - T_0)(T_v - T_e)$$

This quadratic equation may be solved for $T_v$, and the result put into Eq. (7). Equations (8) and (9) are also introduced for convenience

$$Q = Q_{full} \frac{\Delta T_o + \Delta T/r - \sqrt{(T_o - T_e)^2 + 2(\Delta T/r)\Delta T_o \cdot (\Delta T/r)^2}}{2r(T_o - T_e + \Delta T)}$$  \hspace{1cm} (11)
where, to shorten notation,

$$\Delta T_0 = 2(T_h - T_e) - (T_o - T_e)$$  \hspace{1cm} (12)

Note that in the form of Eq.(11) neither $R_e$ nor $R_r$ appear explicitly, but only their ratio $r$.

Sample Parametric Calculations

To show the importance of the parameter $r$ we show three sample calculations, one with $r=1$, one with $r=\frac{1}{4}$, and another $r=1/8$. We choose the following nominal values:

$$T_o = 20^\circ C \quad T_e, max = -20^\circ C$$
$$\Delta T = 5^\circ C \quad T_e, min = -110^\circ C$$
$$T_h = 30^\circ C \quad r = 1, \frac{1}{4}, 1/8$$

In order to keep the same nominal value of $R_r$ as $T_e$ is changed, one multiplies by the ratio of $(T_o - T_e + \Delta T)/(T_o - T_e + \Delta T)_{nom}$. The following results are obtained:

| $r$  | $T_e$ ($^\circ C$) | $(Q/Q_{Full})[(T_0 - T_e + \Delta T)/(T_o - T_e + \Delta T)_{nom}]$  \\
|------|-------------------|-------------------------------------------------------------------|
| 1    | -20               | $[0.1981][45/45] = 0.1981 \quad 8\%$  \\
|      | -110              | $[0.0713][135/45] = 0.0714 \quad 8\%$  \\
| 0.25 | -20               | $[0.6074][45/45] = 0.6074 \quad 28\%$  \\
|      | -110              | $[0.2571][135/45] = 0.7714 \quad 28\%$  \\
| 0.125| -20               | $[0.9384][45/45] = 0.9384 \quad 45\%$  \\
|      | -110              | $[0.4550][135/45] = 1.3651 \quad 45\%$  \\

A-15
Nonlinear Effect

In the preceding the effect of radiator eclipse was explored in the linearized limit. When the sink temperature changes greatly, the heat flow is affected by the nonlinear variation of $T^4$ in the radiation terms. A numerical calculation is easily made despite the nonlinearity.

We had

$$Q = F_{\text{SCR}} \cdot e_f \cdot W_{\text{tot}} \cdot \frac{T_v - T_o}{\Delta T} \cdot (\sigma T_v^4 - \sigma T_e^3) \quad (13)$$

and

$$Q = \frac{1}{R_e} (T_h - T_v) \quad (14)$$

with

$$r = R_e \cdot F_{\text{SCR}} \cdot e_f \cdot W_{\text{tot}} \cdot \sigma (T_v^2 + T_e^2) (T_v + T_e) \quad (15)$$

Equating (13) and (14) and introducing (15) gives

$$(T_h - T_v) = r \cdot \frac{T_v - T_o}{\Delta T} \cdot \frac{T_v^4 - T_e^4}{(T_{v,1}^2 + T_{e,1}^2)(T_{v,1} + T_{e,1})} \quad (16)$$

Equation (15) is understood to apply to both the original condition before eclipse when $T_v = T_{v,1}$ and $T_e = T_{e,1}$ and to the conditions after eclipse when $T_v$ and $T_e$ are colder.

Given the value of $T_h$, the original sink condition $T_{e,1}$, the turn-on point $T_o$, the band width $\Delta T$, and the value of $r$ based upon the original conditions, Eq. (16) is used to find $T_{v,1}$ and $Q_1$ from Eq. (14). Then when $T_{e,1}$ goes to $T_{e,2}$, the equation is used again to find $T_{v,2}$ and $Q_2$ is found from Eq. (14).
Consider the example used before:

\[ T_o = 20^\circ C \quad T_{e,1} = -20^\circ C \quad \Delta T = 5^\circ C \quad T_h = 30^\circ C \]

\[ r = 1/4, \quad T_{e,2} = -110^\circ C \]

In the linearized limit we obtained \( \frac{Q_2}{Q_1} = 1.28 \). Allowing for the non-linearity gives the following values:

**Condition 1, By Binary Search**

<table>
<thead>
<tr>
<th>( T_v ) (°C)</th>
<th>LHS (°C)</th>
<th>RHS (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>0</td>
<td>25</td>
</tr>
<tr>
<td>20</td>
<td>10</td>
<td>0</td>
</tr>
<tr>
<td>25</td>
<td>5</td>
<td>11.25</td>
</tr>
<tr>
<td>22.5</td>
<td>7.5</td>
<td>5.31</td>
</tr>
<tr>
<td>23.75</td>
<td>6.25</td>
<td>8.20</td>
</tr>
<tr>
<td>23.125</td>
<td>6.875</td>
<td>6.738</td>
</tr>
<tr>
<td>23.4375</td>
<td>6.5625</td>
<td>7.4658</td>
</tr>
<tr>
<td>23.2813</td>
<td>6.7188</td>
<td>7.1008</td>
</tr>
<tr>
<td>23.2031</td>
<td>6.7969</td>
<td>6.9193</td>
</tr>
<tr>
<td>23.1641</td>
<td>6.8359</td>
<td>6.8287</td>
</tr>
<tr>
<td>23.1836</td>
<td>6.8164</td>
<td>6.8739</td>
</tr>
<tr>
<td>23.1738</td>
<td>6.8262</td>
<td>6.8513</td>
</tr>
<tr>
<td>23.1690</td>
<td>5.8310</td>
<td>6.8401</td>
</tr>
<tr>
<td>23.1665</td>
<td>6.8335</td>
<td>6.8344</td>
</tr>
</tbody>
</table>

**Condition 2, By Binary Search**

<table>
<thead>
<tr>
<th>( T_v ) (°C)</th>
<th>LHS (°C)</th>
<th>RHS (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>0</td>
<td>46.35</td>
</tr>
<tr>
<td>20</td>
<td>10</td>
<td>0.00</td>
</tr>
<tr>
<td>25</td>
<td>5</td>
<td>21.55</td>
</tr>
<tr>
<td>22.5</td>
<td>7.5</td>
<td>10.38</td>
</tr>
<tr>
<td>21.25</td>
<td>8.75</td>
<td>5.095</td>
</tr>
<tr>
<td>21.875</td>
<td>8.125</td>
<td>7.714</td>
</tr>
<tr>
<td>22.1875</td>
<td>7.8125</td>
<td>9.0422</td>
</tr>
<tr>
<td>22.0313</td>
<td>7.9688</td>
<td>8.3767</td>
</tr>
<tr>
<td>21.9532</td>
<td>8.0469</td>
<td>8.0455</td>
</tr>
</tbody>
</table>

Ratio of \( \frac{Q_2}{Q_1} = 8.046/6.834 = 1.18 \)
This ratio differs appreciably from unity, but somewhat less so than the value obtained with the linearized equation.

Conclusion.

The importance of the ratio of the thermal resistance between the equipment and the evaporator to the thermal resistance between the radiator and the environment is illustrated. When the ratio is large (near one), the transient load is only a few percent of the base load. When the ratio is small, the transient load is a significant fraction of the base load.
APPENDIX A.2.3
CONDENSER AND/OR RESERVOIR SHADOWING TESTS ON SNO09 HEAT PIPE

SNO09 heat pipe tests were directed toward examining two potential artery depriming modes: 1) rapid chilldown of the condenser, and 2) rapid chilldown of the reservoir. A large evaporator mass was a requisite contributor during these hypothesized depriming modes and, accordingly, four kilograms of aluminum were attached to the SNO09 evaporator.

Two series of rapid condenser chilldown tests were run. In the first, the heat in the condenser sink was cooled to -73°C (-100°F), while the condenser was heated to maintain -18°C (0°F). The sink and condenser were coupled through 0.30 centimeters of cork over a 162 in$^2$ (0.1045 m$^2$) area. With the heat pipe operating at 100 watts (well above open artery capacity), the condenser heater was turned off allowing the condenser temperature to fall approximately 2.3°C/min (5°F/min). When no depriming occurred, the tests were repeated with 100 and 150 watts. Depriming at 160 watts under steady state was verified. Thus, cooling the condenser at approximately 2.8°C/min (5°F/min) produced no depriming.

In the second series, the sink was -129°C (-200°F). When the condenser heater was turned off, a cooling rate of approximately 3.9°C/min (7°F/min) was achieved. When no depriming occurred, the test was repeated with 125 watts, and again no depriming was observed. However, when an attempt was made to raise the power level to 150 watts, depriming occurred (at the steady condenser temperature of -18°C (0°F). The pipe was reprimed, held 100 watts, but failed to retain prime at 125 watts. A third attempt resulted in failure to reprim. It is thought that ice-generated bubbles may have been formed in the earlier transient cooling tests, and these bubbles caused the repeated deprimings. Depriming due to rapid chilldown of the condenser was not demonstrated.

Rapid chilldown of the reservoir was achieved by blowing vapors from boiling liquid nitrogen through a cooling coil block attached to the reservoir. With 100 watts power at the evaporator and the condenser at -13°C (0°F), the reservoir was cooled repeatedly at successively higher
rates. Depriming did occur at a rate of -3.2C/min (-5.6F/min). Depriming did not occur at rates of -0.8, -1.2, -1.6 and -2.5C/min (-1.5, -2.1, -2.8 and -4.5F/min, respectively).

Although depriming due to reservoir chilldown was demonstrated, the rates required were substantially higher than those indicated by flight data. This argues against reservoir and/or condenser shadowing as the cause of the CTS anomalies.
APPENDIX A.3
FREEZING BLOWBY TESTS ON SNO09 HEAT PIPE

Freezing blow-by was hypothesized as a possible depriming mechanism. This requires the formation of an incompletely frozen slug bridging the vapor core during a transient in which the pressure on the evaporator side of the slug is increasing with respect to the reservoir side. The resulting pressure difference across the slug can blow liquid from the evaporator side and deplete the evaporator inventory.

To explore this mechanism, it was first necessary to determine how to perform a meaningful 1-g test in the absence of the natural slugging of excess liquid which occurs in 0-g. A successful technique to form an ice plug in heat pipe SNO09 was developed. The procedure is as follows:

a) Instrument the heat pipe with temperature sensors along its length to enable monitoring the location of the gas front.

b) Apply sufficient power to turn the heat pipe on i.e., to move gas front into the condenser section. Maintain the sink temperature a few degrees above the methanol freezing point.

c) Midway along the inactive portion of the condenser, locally code a short (∼2.5 cm) section of the pipe to subfreezing temperatures. Periodically raise the reservoir-end of pipe to force excess liquid to pass over the frozen pipe section in order to enhance ice build-up.

d) Apply periodically, a power pulse to gas reservoir to induce a transient temperature/pressure increase in the reservoir while monitoring the temperature profile about the location of the gas front. Formations of all ice plug will decouple the reservoir from the evaporator side of the heat pipe. The presence of an ice barrier blocking the arteries and vapor spaces can then be determined from the temperature profile along the evaporator side of the plug which should not respond to an induced temperature change in the gas reservoir.

Experiments were then directed toward investigating the hypothesized freezing blow-by depriming mechanism. The SNO09 heat pipe test was modified such that a small, independently chilled, ice plug heat sink was attached.
to the condenser midway along its length. The basic test approach was as follows:

a) The ice plug heat sink was employed to generate a solid plug midway along the condenser.

b) The evaporator power was raised to a value in excess of the open artery heat pipe capacity, raising the temperature and pressure on the upstream side of the ice plug.

c) The ice plug was thawed, allowing the pressure differential to relieve itself by blowing liquid and gas through the thawed region.

If there is liquid communication between the region to thaw first and the arteries, the above sequence of events was hypothesized to lead to depriming of the arteries by pumping them dry. The experiment was run so that this liquid communication condition existed and, in fact, the arteries did deprime. The experiment was run three times, always with the same result. Thus, it has been clearly established that freezing blow-by is a bonafide depriming mechanism and a candidate to explain at least some of the observed anomalies.

To enable modelling the blow-by mechanism, the variable conductance heat pipe subroutine was expanded to include the effects of an ice plug forming in the condenser section of the heat pipe and open or deprimed artery performance. Such a plug decouples the reservoir from the active portion of the heat pipe and alters its control characteristics. It also provides the basis for the freezing blow-by depriming hypothesis; i.e., a pressure difference is established across the ice plug which blows the liquid through as the plug begins to thaw resulting in artery depriming which causes the heat pipe to continue operating with reduced or open artery capacity.

In Appendix A.4.2 a transient thermal analysis of a simple VCHP/radiator system model is described which shows the effects of an ice plug formed in the heat pipe condenser and subsequent blow-by depriming of the arteries on the performance of the sample VCHP system.
APPENDIX A.4.1
BRIEF REVIEW OF LeRC THERMAL STUDIES

In support of Contract NAS3-21740 (Study of Liquid Dynamics in CTS-Type Heat Pipes), NASA LeRC performed thermal analyses on a 'lump' parameter thermal model of the Transmitter Experiment Package. These efforts were conducted by Louis Gedeon of LeRC and are described in two NASA documents entitled "Comparison of Predicted Versus Measured Temperatures for CTS Thermal Anomalies," dated October 1, 1979, and "Modifications to CTS Thermal Model for TRW Contract NAS3-21740," dated January 9, 1980. A review of this analytical task is presented in the sequel.

The CTS model includes the VCHP system, the traveling wave tube, a power processing system baseplate, the spacecraft forward and part of the south panels, the antenna covers and skirts, and the solar array pallet. TRW's Systems Improved Numerical Differential Analyzer (SINDA) computer program and a Variable Conductance Heat Pipe model subroutine (VCHP2) were utilized to solve the CTS model.

During the first transient calculations with a modified VCHP system model, it was found that the heat pipe model subroutine (VCHP2) was not suitable for transient calculations. An analysis at TRW of the solution algorithm in VCHP2 revealed that the solution scheme, in view of the non-linear nature of the model, was inadequate since it was susceptible to becoming numerically unstable. As a result of this analysis, a new solution method was formulated which proved to be successful in circumventing numerical instabilities.

This solution approach was incorporated into a new version of the variable conductance heat pipe model subroutine renamed VCHPDA. In addition, two new features were introduced into this version which permit simulating the effects of an ice plug formation in the heat pipes and open or deprimed heat pipe capacity on the performance of the whole VCHP system. The description of the ice plug and open artery analytical models and the solution scheme, as well as, a listing of VCHPDA written in FORTRAN IV computer language are presented in Appendix A.4.2.
Subsequent transient runs of the CTS model using the new subroutine VCHPDA were normal, except that the predicted temperatures were lower than flight temperature data. In order to improve the temperature predictions, additional modifications of input parameters were introduced into the CTS model. Among the changes made were:

1) The solar absorptance and total hemispherical emissivity on the VCHP system radiator were increased and decreased respectively.

2) The reflected sun heat load on the VCHPs radiator was assumed diffuse and was multiplied by a factor varying with time.

3) The south panel emissivity was increased.

4) Numerous view factors and shadowing parameters were modified.

Although some of the above changes are deemed somewhat physically unrealistic, they resulted in general, in temperature predictions remarkably close to flight data. Transient runs were made for normal day 89 and for periods preceding and including the anomaly of days 75, 82, 101, and 253. The initial conditions were those obtained from steady state calculations using the conditions prevailing prior to the start of a spacecraft eclipse.

For each anomaly day a minimum of two transient runs were made, one assuming normal heat pipe operation prior and during the period of the anomaly, with considerations for ice plug formation, the other run assuming open artery capacity conditions prior and during the anomaly. Additional calculations were made for day 75 and day 253. For day 75, lower open artery capacity values were assumed, and for day 253 a third run was performed in which normal heat pipe operation was assumed from eclipse until the observed time of the anomaly at which time the heat pipes were set for open artery or deprimed capacity operation.

Some significant results from these transient analyses were:

1) On day 253 freezing was not predicted at the time of the anomaly.

2) The maximum calculated heat pipe loads never exceeded 80 watts.
3) The coldest calculated temperature for heat pipe number 1 always occurred near the reservoir end of the condenser where temperature sensor HPT5 is located (see Figure 1-3 in text). Temperatures for heat pipes HP2 and HP3 at similar condenser locations were predicted to be a few degrees colder.

4) Formation of ice plugs in the three heat pipes were calculated on day 82 prior to the anomaly.

5) The profile of the tube body temperature excursion is closely matched on day 253 when the heat pipes were set from normal to open artery capacity operation at the observed time of the anomaly.
APPENDIX A.4.2

GENERALIZED VARIABLE CONDUCTANCE HEAT PIPE MODELING

This appendix presents analytical models, sample problem solution, and listing of subroutine VCHPDA.
As the result of efforts carried out under the 1973 IR&D Advanced Thermal Control program, a method to analytically simulate a gas loaded, variable conductance heat pipe (VCHP) was developed and is documented in Reference 1.

A FORTRAN IV computer code was implemented to numerically solve the analytical model. This code which is suitable to interface with general thermal models involving heat pipes, has been used successfully in steady state thermal simulation of systems, e.g., the transmitter experiment package (TEP) variable conductance heat pipe system (VCHP) in the Canadian Technology Satellite (CTS). Subsequent attempts, however, to utilize the VCHP2 subroutine for transient thermal analyses yielded unsuccessful results due to numerical instabilities arising during the gas-vapor front location/vapor temperature calculations.

Recently, as part of the efforts to investigate the thermal anomalies in the CTS spacecraft, the solution scheme in VCHP2 was analyzed in order to uncover the source of the numerical problems. The analysis revealed that the iterative approach of the subroutine to obtain a converged solution starting from an initial guessed gas length/vapor temperature, was inadequate in view of the non-linear nature of the model.

As the result of this analysis, a new analytical method was formulated which under most physically realizable operating modes, unconditionally circumvents numerical instabilities.
In addition, two new features were introduced into the VCHP model which permits simulating the effects of an ice plug formation and open artery capacity on the performance of a VCHP system. The appropriate FORTRAN IV logic to solve the analytical model was incorporated into a revised version of the VCHP2 subroutine, now identified as VCHPDA. A logic flow chart and listing of the computer program are attached.

Basic Analytical Model:

Referring to the configuration of a typical, wicked-reservoir, VCHP in Figure 1, and invoking the ideal gas law and the "flat front" gas theory, the cumulative distribution of the number of moles of non-condensible gas along the length of a VCHP can be written as:

$$Ng(z) = (P_T(z) - P_{v,R}) \frac{V_R}{R \cdot T_R} + \int_0^z \frac{P_T(z) - P_v(z)}{R \cdot T(z)} A(z) \, dz$$  \hspace{1cm} (1)

for \( z \leq L_g \)

where

- \( L_g \) - location of gas-vapor front or "gas length"
- \( V_R \) - reservoir volume
- \( T_R \) and \( T(z) \) - temperatures at vapor-liquid interface in reservoir and along pipe, respectively.
- \( R \) - universal gas constant
- \( A(z) \) - gas/vapor space cross sectional area
- \( P_{v,R} \) and \( P_v(z) \) - partial vapor pressure of working fluid at \( T_R \) and \( T \) respectively.
- \( P_T(z) \) - total pressure in the VCHP defined in Eq. (2).

$$P_T(z) = P_v(T_v(z))$$ \hspace{1cm} (2)

where \( T_v(z) \) is the vapor temperature in the active section of the pipe.
Figure 1. Configuration of Typical VCHP
where $h(z)$ is the local heat transfer coefficient and $P(z)$, the wetted perimeter. Equations 1, 2 and 3 form a complete non-linear coupled set of equations which enable the calculation of the length of the gas-blocked region and the vapor temperature in the active portion given the temperature distribution along the wall of a VCHP whose configuration conforms to the one sketched in Figure 1.

**Solution Scheme:**

In order to illustrate the solution procedure incorporated into the current version of subroutine VCHPDA, it is convenient to rearrange Equation (1) as follows:

$$Ng(z) = P_T(z) F(z) - N_v(z)$$

where

$$F(z) = \frac{V_R}{R \cdot T_R} + \int_{0}^{z} \frac{A(z')}{R \cdot T(z')} \cdot dz'$$

$$N_v(z) = \frac{P_v, R \cdot V_R}{R \cdot T_R} + \int_{0}^{z} \frac{P_v(z') \cdot A(z')}{R \cdot T(z')} \cdot dz'$$

Resorting to a numerical method to evaluate the integrals $F(z)$, $N_v(z)$ and $P_T(z)$ are calculated stepping out from the reservoir in $\Delta z$ increments. At each step, $Ng(z)$ is evaluated and compared to the total gas inventory, $N_T$. The above procedure is continued until $Ng(z) \geq N_T$ at which point a quadratic binary search scheme is applied to find the root of the transcendental equation.
\[ Ng(z) - N_T = 0 \] \hspace{1cm} (7)

which is satisfied when \( z = L_g \).

To initiate this iterative process, a gas length is approximated by linear interpolation, i.e.,

\[ L_g = z + \frac{Ng(z+\Delta z) - N_T}{Ng(z+\Delta z) - Ng(z)} \cdot \Delta z \]

The search is terminated when a prescribed convergence criterion is met, which in the current VCHPDPA version is defaulted to

\[ \left| \frac{Ng(z) - N_T}{N_T} \right| \leq 10^{-4} \]

Solution to Equation (7) allows the performance of the partially gas-blocked VCHP to be quantified.

The thermal conductance between the vapor and the pipe walls, \( G(z) \), can be calculated,

\[ G(z) = U(z-L_g) \cdot h(z) \cdot \mathcal{P}(z) \] \hspace{1cm} (8)

where \( U \) is a Heaviside-type function defined as

\[ U = \begin{cases} 0.0 & \text{for } (z-L_g) \leq 0.0 \\ 1.0 & \text{for } (z-L_g) > 0.0 \end{cases} \]

and the heat load on the VCHP

\[ Q = \int_{L_g}^{L_T} G(z') \cdot (T_v(L_g) - T(z')) \cdot H(T_v(L_g) - T(z')) \cdot dz' \] \hspace{1cm} (9)

where \( H \) is similarly defined as

\[ H = \begin{cases} 0.0 & \text{for } \{ T_v(L_g) - T(z) \} < 0.0 \\ 1.0 & \text{for } \{ T_v(L_g) - T(z) \} \geq 0.0 \end{cases} \]
The heat load on the heat pipe in terms of power-length can readily be calculated

\[ Q_L = Q \cdot L_{\text{eff}} \tag{10} \]

where

\[ L_{\text{eff}} = \frac{(L_c - L_g)}{2} + L_a + \frac{L_e}{2} \tag{11} \]

The above solution scheme holds provided the VCHP is under load. In the event that \( N_g(z) < N_T \) for \( z \geq L_T \), the gas length \( L_g = L_T \) and the total pressure in the VCHP is calculated by either Equation (1) or (4). Using the latter Equation, the pressure \( P_T \) is given by

\[ P_T = \left\{ \frac{N_T' - N_{v}(L_T)}{F(L_T)} \right\} \tag{12} \]

Ice Plug Modelling

Owing to excess fluid inventory, usually present in a VCHP, operation of the heat pipe under subfreezing sink conditions can result in the formation of an ice barrier in the condenser section effectively decoupling the active portion of the pipe from the gas reservoir. Formation of such an "ice plug" can have a significant effect on the performance characteristics of a VCHP. Furthermore, it can, under certain operating conditions, lead to depriming of the arteries by what is referred to as the "freezing blow-by" mechanism. In order to enable to simulate the consequences of such an event, the ice plug model has been developed and is incorporated as an optional operating VCHP mode in subroutine.

It is assumed that in zero-gravity the bulk of the excess fluid resides as a liquid mass in the gas reservoir and as a slug blocking the vapor spaces in the pipe section at and near the coupling between pipe and reservoir. During a transient
analysis, formation of an ice plug in the condenser is subject to the following constraints: a) the gas reservoir temperature must be above freezing conditions, b) the history of the VCHP must show a net decrease of the gas inventory in the reservoir, i.e., gas front and, presumably liquid slug, movement toward the evaporator, and c) subfreezing temperatures at some point along the condenser and/or adiabatic sections.

These conditions for ice plug formation can be stated mathematically,

\[ T_R > T_{F.P.} \] (Freezing temperature of working fluid) \hspace{1cm} (14)
\[ \frac{\partial N_g}{\partial t} < 0 \] \hspace{1cm} (15)
\[ T(z) < T_{F.P.} \] \hspace{1cm} (16)

Provided conditions given by Equations (14) and (15) are met, the solution procedure subsequently involves stepping out from the reservoir end of the pipe in \( \Delta z \) increments in the search of subfreezing temperatures. The point at which conditions, Equation (16), is first satisfied establishes the location of the ice plug which is currently modelled as a barrier of negligible thickness. The location of the ice plug, \( L_p \), is defined by

\[ L_p = z \text{ where } T(z) < T_{F.P.} \text{ and } T(z-dz) > T_{F.P.} \] \hspace{1cm} (17)

Concurrent calculation of the cumulative gas inventory up to \( z = L_p \) (Equation (1)) enables to determine the distribution of the gas on the reservoir and evaporator side of the ice plug which respectively are:

\[ N_{gR} = N_g(L_p) \] \hspace{1cm} (18)
\[ N_{gE} = N_T - N_{gR} \] \hspace{1cm} (19)
These parameters are stored and used at subsequent times to analyze the characteristics of the two decoupled regions in the VCHP. The solution procedure is then as follows:

I. Calculate total pressure on reservoir side of the plug using Equation (12)

$$P_{T,R} = \left\{ \text{Ng}_R - \text{Nv}(L_p) \right\} / F(L_p) \quad (20)$$

II. Perform analysis on the evaporator side of the ice plug resorting to Equations (1) through (12) and appropriately changing the limits of integration and reservoir volume in order to account for the decoupling of the two regions, i.e., set $V_R = 0$ and evaluate the integrals from $L_p$ to $z \leq L_T$.

In addition, calculation of the effective length by Equation (11) requires the condenser length, $L_c$, be reduced by $L_p$.

$$L_c' = L_c - L_p \quad (21)$$

The above procedure yields an estimate of a significant new parameter: the pressure differential across the ice plug, which can result in depriming of an arterial VCHP when liquid contact is established between the two regions.

Open-Artery Capacity Modelling

If a mechanism such as the one described above leads to depriming of an arterial VCHP, the designed capacity of the heat pipe will diminish to what is usually referred to as "open-artery" capacity. In such an operating mode, capillary pumping pressure is substantially reduced and continuous operation of a VCHP at high loads after depriming will generally result in partial evaporator dry-out.

In order to simulate the performance of a VCHP with
open-artery capacity, a model was developed and is currently incorporated in subroutine VCHPDA as an operational operating mode. To exercise this option, a new constraint is introduced into the overall VCHP model. The power-length capacity of the heat pipe, defined in Equation (10), cannot now exceed a prescribed capacity \( Q_L \), i.e.,

\[
Q_L(z) \leq (Q_L)_0
\]  

(22)

The effective length, \( L_{\text{eff}} \), is calculated by

\[
L_{\text{eff}} = \frac{L'}{2} + L_a + L'_{\text{e}}
\]

(23)

where, \( L'_{\text{e}} \), is the evaporator length reduced by the length of the dry section of the pipe, \( L_d \).

\[
L'_{\text{e}} = L_{\text{e}} - L_d
\]

(24)

\[
L_d = L_T - L_w
\]

(25)

where, \( L_w \), is the length of the wet portion of the pipe measured from the reservoir end.

Rearranging Equation (22) as

\[
Q_L(z) - (Q_L)_0 = 0.0
\]

(26)

the solution procedure involves finding the root of the above Equation which establishes the length of the wet section of the pipe and hence, the dry portion of the evaporator.

The general procedure which requires a double-iterative scheme is as follows:

I. Initialize \( L_w = L_T \) (i.e., \( L_d = 0.0 \)) and \( L_g = L_T \)

II. Calculate gas length, \( L_g \), and vapor temperature, \( T_v \), such that Equation (7) is satisfied
III. Calculate $Q_L(z)$ for $z = L_w$ and check whether constraint (Equation (22)) is met. If the check is positive, the solution to the model has been attained and the procedure is terminated; otherwise, perform step IV.

IV. Holding $L_g$ fixed, and stepping out from the evaporator end of the pipe in $(-\Delta z)$ increments, recalculate the vapor temperature using

$$T_v(z) = \frac{\int_{L_g}^{z} h(z') T(z') P(z') \, dz' / \int_{L_g}^{z} h(z') P(z') \, dz'}{L_g}$$

and subsequently calculate $Q_L(z)$.

Repeat steps III and IV until $Q_L(z) \leq (Q_L)_0$ at which point a quadratic binary search is performed to find the root of Equation (26). To initiate the iterative process, the wet length, $L_w$, is approximated by linear interpolations as follows:

$$L_w = z + \left\{ \frac{(Q_L)_0 - Q_L(z)}{(Q_L(z+\Delta z) - Q_L(z))} \right\} \cdot \Delta z$$

The search is terminated when a prescribed convergence criterion is satisfied which is defaulted in VCHPDA to

$$\left| \frac{Q_L(z) - (Q_L)_0}{(Q_L)_0} \right| \leq 10^{-4}$$

With the wet length, $L_w = z$, held fixed a new gas length and vapor temperature are calculated confining the calculations to the wet section of the pipe. Steps II through IV are repeated until

$$\left| \frac{W^{k+1} - W^k}{W^{k+1}} \right| \leq 10^{-4}$$

where $W^k$ is either the gas length, $L_g$, or wet length, $L_w$, and superscript 'k' is the iteration number.
Having established the domain of the active section of the pipe, i.e., \( L_g \leq z \leq L_w \), the local thermal conductance, \( G(z) \), between the vapor and adjacent pipe walls can now be redefined to reflect the reduction of effective heat transfer area resulting from gas blockage and wall dry-out.

\[
G(z) = H(z) h(z) P(z)
\]  

(29)

where \( H(z) \) is defined as

\[
H = \begin{cases} 
0.0 & \text{for} \ (z-L_g) < 0.0 \ \text{or} \ (L_w-z) < 0.0 \\
1.0 & \text{for} \ (z-L_g) \geq 0 \ \text{and} \ (L_w-z) \geq 0.0 
\end{cases}
\]  

(30)

**VCHPDA: Computer Program Subroutine Usage**

This subroutine solves numerically the analytical VCHP model described in preceding sections of this report. Its solution logic is written in FORTRAN IV language which is compatible with current Control Data Corporation (CDC) compilers. As a program subroutine, VCHPDA is meant to interact with general lumped parameter thermal systems. In its present form, however, VCHPDA is only suitable for usage in conjunction with TRW's Systems Improved Numerical Differencing Analyzer (SINDA).

The subroutine must be called in VARIABLES 1. In the process of generating a thermal model which involves a VCHP subsystem, the following convention must be followed.

a) Heat pipe wall nodes are numbered and input sequentially stepping out from reservoir end.

b) Wall to vapor conductors are numbered and input sequentially as in a).

c) The vapor of each heat pipe must be declared a boundary node.
Figure 2. VCHPDA Solution Flowchart
The calling sequence is:

```
VCHPDA (A1, TN, A3, GN, A2, TJ, TK, QN, K1, K2, TL, A4)
```

where:

- **A1** = Working fluid saturation pressure vs temperature array
- **TN** = First node in wall temperature array
- **A3** = Array of vapor to wall conductors values with no gas input in order starting from the reservoir end (Btu/hr-°F)
- **GN** = First conductor in vapor to wall conductor array
- **A2** = Heat pipe characteristics array (see below)
- **TJ** = Reservoir temperature node
- **TK** = Control temperature node (for a heated reservoir system)
- **QN** = Reservoir heat input identification (for a heated reservoir system)
- **K1** = Constant set to zero, fixed point
- **K2** = Usage flag; 1 = normal calculation usage, -1 = printout of heat pipe data only, used in output calls
- **TL** = Vapor temperature node
- **A4** = Array of node lengths starting from reservoir end (inches)

Inputs for the heat pipe characteristics array are:

- **A2(1)** = Total heat pipe length (inches)
- **A2(2)** = Number of heat pipe wall nodes (Floating Point Number)
- **A2(3)** = Heat pipe free volume to length ratio (in³/in)
- **A2(4)** = Reservoir volume (in³)
- **A2(5)** = Gas NR value (Ft-lb/°R)
- **A2(6)** = Reservoir wick flag: 1.0 wicked reservoir, -1.0 un-wicked reservoir
- **A2(7)** = 1.0
- **A2(8)** = Reservoir heater power (Btu/Hr), -1.0 if unheated
- **A2(9)** = Upper temperature setting (°F), (Heated reservoir system)
- **A2(10)** = Lower temperature setting (°F), (Heated reservoir system)
Sample Problem

To illustrate the usage of subroutine VCHPDA, the steps involved in solving a simple problem are described in what follows. Consider the heat pipe radiator system sketched in Figure 3. The heat source is a 0.32 Kg aluminum block in which power is uniformly generated. The block is mounted on a 2.5 cm wide saddle that is attached to the heat pipe over a 30 cm-long section. The VCHP is a 1.25 cm O.D., 1.0 meter long aluminum/ammonia heat pipe with a 0.1 litter gas reservoir. Short Al/SS/Al and Al/SS transition sections minimize the coupling by conduction between evaporator and condenser and between reservoir and condenser respectively. Power generated in the heater block is transported by the heat pipe to a radiator panel from which is radiated to a sink.

The system is nodalized as shown in Figure 4. A listing of this model is shown in pages 45 through 47. The FORTRAN version of subroutine VCHPDA is listed in pages 48 through 67.

Steady state, followed by transient solutions were obtained using SINDA. Selected solution outputs are shown in pages 68 through 79. The results are summarized in Figure 5.
Figure 4. Sample Problem Nodalization
FIGURE 5 VCHP Model Transient Response

- Power (Watt)
- Sink Temp
- Vapor Temp
- Pressure (N/m² x 10⁻⁴)
- Temperature (°K)
- Time (Hr)
- Wet Length
- Gas Length
- Freezes Location
- Ice Plug
- Thaws Location
For the first hour after steady state conditions were reached, the VCHP system is shown to transfer 40 watts from the heat source to a sink at 190°K. The vapor temperature remains constant at 296°K and the gas blocks about 3/5 of the condenser section. When the power is turned off, the vapor temperature can be seen to drop resulting in increase gas blockage of the pipe. At 1.5 hours the entire length of the pipe is gas blocked. Although the rate of heat leak from the evaporator is of only one percent of peak power (i.e., 0.40 watt), the vapor, and consequently, the source temperatures undergo a substantial drop during the off period. This is the result of the rather small mass (0.32 Kg) of the source modelled in this problem.

Continuous operation at low sink conditions, causes the inactive condenser to reach subfreezing temperatures and, as shown on the bottom of Figure 5, an ice plug forms at 2 hours, 0.10 meters from the reservoir end of the pipe. The center sketch on the figure shows that a pressure differential develops across the ice plug. This barrier effectively decouples the evaporator from the gas reservoir and causes a large temperature overshoot when the power is turned on at 4.0 hours. Between this time and 5.0 hours, a large pressure differential is sustained across the ice plug. Following a step increase in sink temperature (5.0 hours) simulating a change in solar view factor, the ice plug thaws giving rise to the freezing blow by mechanism which is postulated to cause wicking failure. Continuous operation of the VCHP at 40 watts and deprimed capacity (7.5 watt-meter) is seen to result in partial evaporator dry-out as the heatpipe adjusts its effective length to satisfy the imposed capacity constraint.

References:

BLANK CARD

BCD 3THERMAL LPC

BCD 9 VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

END

BCD 3NODE DATA

GEN 1,20,1,70,-1.
22,70,-1.

GEN 23,9,1,70,174,,211,7.4-4,1.
32,70,-1.
33,90,,15
34,-100,,0.04
-35,-120,,1.
-36,70,,1.

END

RELATIVE NODE NUMBERS

1 THRU 16 23 24 25 26 27 28 29 30 31 33
11 THRU 20 34 1 2 3 4 5 6 7 8 9
21 THRU 30 10 11 12 13 14 15 16 17 18 19
31 THRU 35 26 22 32 35 36

NODE ANALYSIS... DIFFUSION = 11, ARITHMETIC = 22, BOUNDARY = 2, TOTAL = 35

ELAPSED TIME = 2.34, PHASE TIME = 0.08

BCD 3CONDUCTOR DATA

GEN -19,1,23,1,35,0,,0346E-8,1,1,1.
-10,22,35,0,00822E-8
-11,3,35,0,00492E-8
12,2,2,,5

GEN 13,9,1,3,1,23,1,2,,1,1,,1.
22,12,32,5

GFN 73,6,1,15,1,33,0,2.56,1,,1,1.
29,1,34,0.0305
30,1,2,0.0252
31,2,3,0.0777

GEN 32,8,1,3,1,4,1,0,252,1,1,,1.
40,11,17,0.0777
41,12,13,0.0252
42,13,14,0.0252
43,14,15,0.0498

GEN 44,5,1,15,1,16,1,,252
CAL -49,34,35, C.5,64, 1714E-9,1.
GEN 50,20,1,1,1,36, C.5,1,1,1,1.
END

RELATIVE CONDUCTOR NUMBERS

<p>| | | | | | | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>THRU</td>
<td>10</td>
<td></td>
<td>2</td>
<td></td>
<td>3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>THRU</td>
<td>20</td>
<td>11</td>
<td>17</td>
<td>13</td>
<td>14</td>
<td>15</td>
<td>16</td>
</tr>
<tr>
<td>21</td>
<td>THRU</td>
<td>30</td>
<td>21</td>
<td>22</td>
<td>23</td>
<td>24</td>
<td>25</td>
<td>26</td>
</tr>
<tr>
<td>31</td>
<td>THRU</td>
<td>40</td>
<td>31</td>
<td>32</td>
<td>33</td>
<td>34</td>
<td>35</td>
<td>36</td>
</tr>
<tr>
<td>41</td>
<td>THRU</td>
<td>50</td>
<td>41</td>
<td>42</td>
<td>43</td>
<td>44</td>
<td>45</td>
<td>46</td>
</tr>
<tr>
<td>51</td>
<td>THRU</td>
<td>60</td>
<td>51</td>
<td>52</td>
<td>53</td>
<td>54</td>
<td>55</td>
<td>56</td>
</tr>
<tr>
<td>61</td>
<td>THRU</td>
<td>69</td>
<td>61</td>
<td>62</td>
<td>63</td>
<td>64</td>
<td>65</td>
<td>66</td>
</tr>
</tbody>
</table>

CONDUCTOR ANALYSIS...
LINEAP = 57, RADIATION = 17, TOTAL = 69, CONNECTIONS = 69

ELAPSED TIME = 2.54, PHASE TIME = .20

BCD 3CONSTANTS DATA
NLOOP=300, ARLXCA=0.1, DRLXCA=0.1
END

CONSTANTS ANALYSIS... USFR = 0, ADDFD = 0 0 0, TOTAL = 0

ELAPSED TIME = 2.56, PHASE TIME = .02

BCD 3APRAY DATA
15 -450,9, -170,0,2, -160,0,5, -150,1, -140,1, -130, -3, -120,5
-110,8,9, -100,1,25,90,1,88, -80,2,77, -70,3,99, -60,5,62
-50,4,7,9, -40,1,0,2,30,1,43, -30,2,4,10, -23,92,0, -30,62
10,3,3,7,3, -3,4,7,3, -3,60,0,4,0,7,3,6,9,50,8,9,6,4,6,0,108,15
70,1,29,4,7,8,0,1,53,6,9,90,1,81,6,7,100,2,13,1,110,2,48,49
115,2,67,7,6,1,2,0,2,88,1,2,12,5,3,09,6,1,130,3,32,2,8,135,3,35,6,16
140,3,81,2,9,1,4,5,4,0,7,1,5,0,4,3,5,4,1,55,4,6,4,5,3,160,4,495,0,3
165,5,26,9,7,1,70,5,6,0,3,8,17,5,59,5,3,1,0,6,31,7,1,85,6,69,84
190,7,09,5,3,1,9,5,7,5,8,7,2,00,7,93,92
END

-2,4,0,2,0,115,6,7,1,2,53,1,1,1,0,1,1,1,1,1,1,1,108,0,0
4,6,1000,0,1,12,END & HEAT PIPE CHARACTERISTICS
-3,SPACE=20,END
-4,SPACE=20,END

END

ARRAY ANALYSIS... NUMBER OF APYAYS = 4, TOTAL LENGTH = 156

ELAPSED TIME = 2.66, PHASE TIME = .10

ELAPSED TIME = 2.76, PHASE TIME = .10
RCD 3EXECUTION
F DIMENSION X(300)
F NDIM=300
F NTH=0
  STFSOS(5,20,A3)
  STFSOS(2,20,A4)
  CINDSL
  TIME=M=8.
F OUTPLT=2.
M A(2+12)=1.
M A(2+14)=1.
END
  ELAPSED TIME =  2.87,  PHASE TIME =   .10
RCD 3VARIABLES 1
M IF(TIME<1.0.OR.TIME0.GE.4.0) 033=136.5
M IF(TIME0.GT.5.0) T35=-50.
  VCHPDA(A1,T1,A3,G50,A2,T34,1.,1.,0,1,T36,A4,TIME0)
F 100 CONTINUE
END
  ELAPSED TIME =  2.92,  PHASE TIME =   .05
LEN
RCD 3VARIABLES 2
END
  ELAPSED TIME =  2.94,  PHASE TIME =   .02
RCD 3OUTPUT CALLS
  TOPLIN
  TPRINT
  GPRINT
F 100 CONTINUE
  VCHPDA(A1,T1,A3,G50,A2,T34,1.,1.,0,-1.,T36,A4,TIME0)
END
  ELAPSED TIME =  3.02,  PHASE TIME =   .08
SUBROUTINE VCHFPA(A1,T1,A3,G1,A2,T2,T3,O,N1,N2,N3,T1,A4,FTIM)

C THIS SUBROUTINE IS A NEW VERSION OF THE PREVIOUS VCHF SUBROUTINE.
C THIS SUBROUTINE CONSIDERS THE FORMATION OF AN ICE PLUG WHICH
C EFFECTIVELY DECOUPLES THE RESERVOIR FROM THE REST OF THE HEAT
C PIPE, IN ORDER TO CONSIDER THE EFFECT OF AN ICE PLUG ON THE
C TRANSIENT RESPONSE OF A VARIABLE CONDUCTANCE HEAT PIPE SYSTEM,
C TWO MORE DATA NEED TO BE INCLUDED AT THE END OF ARRAY $=51$, $=52$ AND $=53$ IN THE CTS MODEL.
C THESE DATA ARE:

A1= FREEzing POINT OF THE WORKING FLUID

A2(15)= A FLAG (=1, FOR AND ICE TO BE ALLOWED TO Form,
=0, FOR AN ICE PLUG NOT TO BE CONSIDERED)

A2(16)= LENGTH OF THE ADIABATIC SECTION BETWEEN THE EVAPORATOR
AND THE CONDENSER

A2(17)= SPECIFIED OPEN ARTERY CAPACITY IN BTU/HR-INCH

A2(18)= A FLAG (=1, TO CONSIDER OPERATION OF THE HEAT PIPE
WITH OPEN ARTERY CAPACITY, AND =0, TO CONSIDER OPERATION
OF THE HEAT PIPE WITH NORMAL OR UNLIMITED CAPACITY)

A2(19)= LENGTH OF THE EVAPORATOR SECTION

A2(15)=1. SHOULD BE USED FOR TRANSIENT RUNS ONLY, HOWEVER:
A2(18)=1. COULD BE USED FOR STEADY AS WELL AS FOR TRANSIENT
RUNS. IN ADDITION A2(15)=1. AND A2(18)=1. COULD BE USED
SIMULTANEOUSLY FOR TRANSIENT RUNS.
IN THE PREVIOUS VCHP2 IT WAS REQUIRED THAT A2(13)=1 IF
USEL AS A DAMPING FACTOR FOR STADY STATE RUNS AND THAT
A2(13)=1, BE USED AS A DAMPING FACTOR FOR TRANSIENT SOLUTIONS.
IN THIS NEW VERSION THOSE TWO PARAMETERS SHOULD BE USED
AS BEFORE ALTHOUGH NOW THEY MERELY FUNCTION AS FLAGS.
THE PARAMETER ENBE WHICH WAS USED TO SPECIFY THE NUMBER OF
ITERATIONS DESIRED IN THE LOOP TO CALCULATE THE GAS FRONT
LOCATION, THE NEW VAPOR-TO-WALL NODE CONDUCTORS AND THE
VAPOR TEMPERATURE IS NOT USED IN THIS NEW VERSION AND
HENCE IT CAN BE TREATED AS A DUMMY PARAMETER.
THE USER WILL NOTICE THAT THERE ARE TWO DIFFERENT OUTPUTS
DEPENDING WHETHER AN ICE PLUG HAS FORMED OR WHETHER AN ICE PLUG
HAS NOT BEEN CONSIDERED OR HAS THAWED.
AS BEFORE THE PARAMETER ENBE IS THE PRINTING FLAG.
ANY OTHER INPUT PARAMETER REQUIRED TO CALL #VCHP2# WHICH
HAS NOT BEEN MENTIONED ABOVE SHOULD BE CONSIDERED AS REFOFF.
A1=FLUID P VS T ARRAY, T WALL TEMP ARRAY, A3=WALL-VAPOR CONDUCTORS
C1=WALL-VAPOR CONDUCTORS CALCULATED, A2=HEAT PIPE CHARACTERISTICS
T2=RESERVOIR TEMP, T3=CONTROL TEMPERATURE, Q=RESERVOIR HEATER Q
CIMENSION TW(1),A3(1),G1(1),P2(1),A4(1),NL(80),N2(80),F(80),
L(P(80),PW(80),T(80),FN(10)
REAL L,N1,N2,NGAS,1TOT,LGRE,LGEV,NPEV,NPRE,N23,LP,LPD,LEFF,
L1,L2,L3,L4,LCMD
DATA IFRETE/0/
DATA FN/10*0./
IF(FT1M,CT,4,5,AND.,IFREZE,F0,0)A2(1R)=1.
CALL FX(A2(2),N)
NM1=N-1
NP1=N+1
IF( A2(6).GT.0,) GO TO 102
CALL OIDEGL(TW(1),A1,PVR)
GO TO 101
CALL OIDEGL(T2,A1,PVR)
CONTINUE
A=A2(3)
LCU=D*A2(1)-A2(16)-A2(19)
VR=A2(4)
TP=T2
PVTR=PVP*VR/(TK+460.)/12.*
VTR=VR/(TR+460.)/12.*
NGAS=A2(5)
IF(A2(13).LT.1.) GO TO 500
IF(IFREZ.EQ.1) GO TO 71
SUM15=0.
SUM610=0.
GO 70 J=1,9
I=10-J
FN(I+1)=FN(I)
IF(I.GT.4) SUM610=SUM610+FN(I+1)
SUM15=SUM15+FN(I+1)
CONTINUE
FN(1)=PVR
SUM15=SUM15+FN(1)
SUM15=SUM15-SUM610
IF(SUM15.GT.SUM610) GO TO 500
IF(T2.LT.A2(14)) GO TO 500
GO TO 75
DO 72 I=1,10
FN(I)=0.
SUM15=0.
SUM610=0.
CONTINUE
IF(A2(15).LT.1.) GO TO 500
DO 90 I=1,N
IF(TW(I).LT.A2(14)) GO TO 51
CONTINUE
IF(IFREZE.EQ.1) GO TO 52
IFREZE=0
GO TO 90
CONTINUE
01360 889 CONTINUE
01370 DC 53 J=1,N
01380 I=NP1-J
01390 IF(TW(I).LT.A2(14)) GO TO 54
01400 CONTINUE
01410 54 IFPEZ=1
01420 IP1=I+1
01430 IPC=I
01440 N2(IPO)=PSAT*F(IPO)-N1(IPO)
01450 NPEV=NGAS-N2(IPC)
01460 NPREF=N2(IPO)
01470 CONTINUE
01480 SUM1=0.
01490 SUM2=0.
01500 SUM3=0.
01510 GO 60 I=1,IPG
01520 CALL D1DEG1(TW(I),A1,PW(I))
01530 SUM1=SUM1+PW(I)*A4(I)*A/(TW(I)+460.)/12.
01540 N1(I)=PVTP+SUM1
01550 SUM2=SUM2+A4(I)*A/(TW(I)+460.)/12.
01560 F1(I)=0.
01570 F(I)=VTP+SUM2
01580 SUM3=SUM3+A4(I)
01590 L(I)=SUM3
01600 CONTINUE
01610 LGRE=L(I)
01620 PCTF=(NPFE+N1(IP0))/F(IPO)
01630 PGR=PTOTRE-PVR
01640 RNP=PGR*VF/(TR+460.)/12.
01650 IFULL=0
01660 N0=m*1
01670 N3=a
01680 IO=0
01690 LP=A2(1)
01700 IQO=-3
01710 IB=0
01720 CLCT*A2(17)
01730 CONTINUE
01740 SLV1=0.
01750 SUM2=0.
01760 SUM3=0.
01770 SUMT=0.
01780  SUMG=0.
01790  CT+1 I=I+1,i3
01800  J=J+1+IP-I
01810  CALL DICEG1(TW(I),A1,PW(I))
01820  SUM1=SUM1+P(I)*A1*6/(TW(I)+460.)/12.
01830  N1(I)=SUM1
01840  SUM2=SUM2+A1*6/(TW(I)+460.)/12.
01850  F(I)=SUM2
01860  SUM3=SUM3+A1(I)
01870  L(I)=SUM3
01880  G1(I)=0.
01890  SUMT=SUMT+TW(J)*A3(J)
01900  SUMG=SUMG+A3(J)
01910  T(J)=SUMT/SUMG
01920  CALL DICEG1(T(J),A1,P(J))
01930  61 CONTINUE
01940  P/CTEV=(NPFW+N1(N3))/F(N3)
01950  CALL DICEG1(TW(N3),A1,PSAT)
01960  N2(N3)=PSAT+F(N3)-N1(N3)
01970  ICHK=0
01980  DO 1111 I=I1,NO
01990  N2(I)=P(I)+F(I)-N1(I)
02000  1111 IF(K2(I)-GT,NPFW.AND,.ICHELK.LT.1) ICHK=I
02010  IF(TCHE=CK.EQ.0) GO TO 600
02020  SUMG=0.
02030  SUMGT=0.
02040  DO 2 J=I1,NO
02050  I=NO+IP1-J
02060  SUMGT=SUMGT+TW(I+1)*A3(I+1)
02070  SUMG=SUMG+A3(I+1)
02080  T(I+1)=SUMGT/SUMG
02090  CALL DICEG1(T(I+1),A1,P(T+1))
02100  N2(I)=P(I+1)*F(I)-N1(I)
02110  IF(N2(I).LT.NPFW.AND.1.LT.ICHCHK) GO TO 601
02120  G1(I+1)=A3(I+1)
02130  62 CONTINUE
02140  FPA(C=(N2(I)-NPFW)/N2(I)
02150  G1(I)=FPA(C*A3(I)
02160  L(I)=(1.-FPA(C)*A4(I)
02170  LGFV*L(I)
02180  XMD=(1.-FPA(C)*A4(I
02190  N1(I)=PW(I)*XMD*A/(TW(I)+460.)/12.
02200  \( F(I) = \frac{XNDA}{T(I) + 460} / 12 \).  
02210  \( SUMC = SUMC + T(I) + C1(I) \)  
02220  \( SUMG = SUMG + G1(I) \)  
02230  \( TSAT = \frac{SUMC}{SUMG} \)  
02240  CALL D1NEGL(TSAT, A1, PSAT)  
02250  \( N2(I) = PSAT \cdot 0.1 \cdot N1(I) \)  
02260  \( X4 = N2(I) \)  
02270  \( T1 = TSAT \)  
02280  \( A7(7) = LGEV + LGRE \)  
02290  IF (IFULL.EQ.2) GO TO 1607  
02300  IF (A2(18).LT.1.) GO TO 500  
02310  CALL QMETER(T1, Tw, GL, X5, X6, N3)  
02320  CLC = GL(LP, A7(16), A2(7), LCOND, X6, LEFF)  
02330  IF (10.EQ.-2) GO TO 5002  
02340  IF (CLC.LT.2(17)) GO TO 1353  
02350  \( T0 = -1 \)  
02360  CLC = QLC  
02370  CLC3 = QLG  
02380  \( LP = LN3 + LGRE \)  
02390  \( L1 = LP \)  
02400  \( C1(N3) = 0 \)  
02410  \( NO = NO - 1 \)  
02420  \( N3 = N3 - 1 \)  
02430  IF (IFULL.EQ.1)  
02440  GO TO 999  
02450  IF (10.EQ.0) GO TO 500  
02460  \( A0 = A0 + 1 \)  
02470  \( N3 = N3 + 1 \)  
02480  \( GLL = QLC \)  
02490  \( QLC = QLL \)  
02500  \( A4(N3) = A4(N3) \)  
02510  \( G1(N3) = A3(N3) \)  
02520  \( IQ = -2 \)  
02530  \( LP = LN3 + LGRE \)  
02540  \( L3 = LP \)  
02550  \( CLP = L3 - L1 \)  
02560  GO TO 999  
02570  IF (10.EQ.-0) GO TO 1704  
02580  \( C1(N3) = 0.5 \cdot LN3 \)  
02590  \( CY = 0.5 \cdot A4(N3) \)  
02600  \( LP = LN3 + EX + LGRE \)  
02610  \( L2 = LP \)
03040  FRAC1=FRAC1-SIGN*DF1
03050  FPAC=FRAC1
03060  IT=C
03070  GO TO 927
03080  93 IF(M2(I+1).LT.NPEV) GO TO 94
03090  94 GO TO 95
03100  CF2=DF2/2.
03110  N23=N2(I+1)-NPEV
03120  SIGM=N23/APS(N23)
03130  FRAC2=FRAC2-SIGN*DF2
03140  FPAC=FRAC2
03150  IT=1
03160  GO TO 927
03170  602 L(I+1)=L(I)+YN
03180  LGEV=L(I+1)
03190  A2(7)=LGRE+LGEV
03200  TI=TSAT
03210  X4=N2(I+1)
03220  1607 IF(A2(18).LT.1.) GO TO 500
03230  IF(IFULL.EQ.1) GO TO 1606
03240  GO TO 16CF
03250  1606 N3=N3+1
03260  NO=NO+1
03270  LP=L(N3)+LGRE
03280  IFULL=2
03290  GO TO 999
03300  1608 CONTINUE
03310  CALL OMEETER(T1,Tw,Gl,X5,X6,N3)
03320  CLC=QL(LP,A2(16),A2(7),LCOND,X6,LEFF)
03330  IF(IQ.EQ.-2) GO TO 5001
03340  IF(QLC.LT.A2(17)) GO TO 1453
03350  TO=-1
03360  CLG=QLC
03370  CLC3=QLG
03380  LP=(N(0)+LGRE
03390  L1=LP
03400  NO=NO-1
03410  Gl(X3)=0.
03420  X3=k3-1
03430  GO TO 999
03440  1453 IF(IQ.EQ.0) GO TO 500
03450  NO=NO+1
03460 N3=N3+1
03470 A4=N3=A4(N3)
03480 C1N3=A3(N3)
03490 QLL=QLC
03500 QLC1=QLL
03510 IO=-2
03520 LP=L(N3)+LGPE
03530 L3=LP
03540 ELPL3-11
03550 GO TO 999
03560 5001 IF(IOW.EQ.0) GO TO 1800
03570 G1(N3)=.5*G1N3
03580 DX=.5*A4N3
03590 LP=1(NO)+DX+LGPE
03600 L2=LP
03610 IF(IOW.EQ.3) GO TO 1804
03620 1800 IF(I8.EQ.0) QLC2=QLC
03630 I8=ABS((QLC-A2(17))/A2(17))
03640 IF(I8.LT.1.E-3) GO TO 1805
03650 IF(I8.EQ.0)
03660 1 CALL QUADRA(L1,L2,L3,L0,QLC1,QLC2,QLC3,QLCT)
03670 CALL BINARY(L1,L2,L3,LP,QLC,QLC1,QLC2,QLC3,QLCT,IP)
03680 DX=LP-(L(N0)+LGPE)
03690 G1(N3)=DX/DLP*G1N3
03700 1804 A3(N3)=G1(N3)
03710 A4(N3)=DX
03720 I9C=0
03730 GO TO 999
03740 1805 A4(N3)=A4N3
03750 A3(N3)=G1N3
03760 LPD=A2(1)-LP
03770 GO TO 500
03780 600 CONTINUE
03790 X4=NPEV
03800 LGEV=L(N3)
03810 T1=TW(N3)
03820 PSAT=PTOTEV
03830 500 CONTINUE
03840 LTGT=LGEE+LGTV
03850 A2(7)*LTGT
03860 EPV=PSAT-PTOTRF
03870 LPD=A2(1)-LP
04300  TGSUM=TGSUM+TW(I+1)*A3(I+1)
04310  GSUM=GSUM+A3(I+1)
04320  TSAT=TGSUM/GSUM
04330  CALL D1REF61(TSAT,A1,P(I+1))
04340  N2(I)=P(I+1)*FAS(I)-H1(I)
04350  IF(N2(I).LT.NGAS.AND.1.LT.ICHECK) GO TO 2000
04360  G1(I+1)=A3(I+1)
04370  CONTINUE
04380  C1(1)=(A4(1)-C1)/A4(1)*A2(1)
04390  A2(7)=0,
04400  T1=TSAT
04410  IF(IFULL.EQ.2) GO TO 607
04420  IF(A2(18).LT.1.) GO TO 121
04430  CALL QMETER(T1,TW,G1,X5,X6,N3)
04440  QLC=QL(LP,A2(16),A2(7),LCMD,X6,LEFF)
04450  IF(10.EQ.-2) GL TO 4002
04460  IF(QLC+LT,A2(17)) EO TO 353
04470  IQ=-1
04480  QLC=QLC
04490  LP=L(N0)
04500  G1(N3)=0.
04510  QLC3=QLG
04520  L1=LP
04530  N0=N0-1
04540  N3=N3-1
04550  IFULL=1
04560  GO TO 388
04570  353 IF(I0,EQ.0) GO TO 121
04580  N0=N0+1
04590  N3=N3+1
04600  QLI=QLC
04610  A4M=A4(N3)
04620  G1N3=A3(N3)
04630  IQ=-2
04640  LP=L(N3)
04650  QLC1=QLL
04660  L3=LP
04670  MLP=L3-L1
04680  CT TO 888
04690  4002 IF(I0G.EQ.0) GL TO 704
04700  G1(N3)=.5*G1N3
04710  CX=.5*A4N3
04720 LP=L(M)+CX
04730 L2=L2
04740 IF(I00,F0,-3) GO TO 904
04750 704 IF(I10,F0,0) QLC2=QLC
04760 TEPM=ABS((QLC-A2(17))/A2(17))
04770 IF(TEPM,L1,E=-3) GP TO B03
04780 IF(I16,E0.0)
04790 1 CALL QMCREA(L1,L2,L3,LP,QLC1,QLC2,QLC3,QLC)
04800 CALL PIVARY(L1,L2,L3,LP,QLC,QLC1,QLC2,QLC3,QLCT,IN)
04810 DX=LP-L(N0)
04820 61(N3)=DX/IDL*61N3
04830 904 A3(N3)=G1(N3)
04840 A4(N3)=DX
04850 12C=0
04860 GP TO 8R8
04870 803 LPD=A2(1)-LP
04880 A3(N3)=G1(N3)
04890 A4(N3)=A4(N3)
04900 IFULL*0
04910 X4=-0.
04920 PNR=A2(5)
04930 PGP=PSAT-PVG
04940 GO TO 3000
04950 121 PSAT=P(I1+1)
04960 LPD=0.
04970 X4=-0.
04980 RNR=A2(5)
04990 PGR=PSAT-PVR
05000 GO TO 3000
05010 1000 PSAT=PNTT
05020 LPD=0.
05030 LEFF=0.
05040 A2(7)=A2(1)
05050 T1=TW(N3)
05060 PGR=PSAT-PVR
05070 PNP=PGR*A2(4)*(T2+460.)/17.
05080 X4=A2(5)-RNP
05090 (C0.61)='1*3
05100 N2(1)=PSAT*F(1)-N1(N1)
05110 P1 CONTINUE
05120 GO TO 3000
05130 2000 CONTINUE
05140  IT=2
05150  FFAC=(NGAS-N2(I))/(N2(I+1)-N2(I))
05160  FRAC1*FFAC
05170  FRAC2=FFAC
05180  DF1=FRAC
05190  DF2=1.-FRAC
05200  DN=PW(I+1)*A/(TW(I+1)+460.)/12.
05210  DF=//(TW(I+1)+460.)/12.
05220  G1(I+1)=(1.-FFAC)*A4(I+1)
05230  XMD=F*AC*A4(I+1)
05240  DGT=TW(I+1)*(G1(I+1)-A3(I+1))
05250  DG=G1(I+1)-A3(I+1)
05260  TSAT=(TGSUM+DGT)/(GSUM+DG)
05270  CALL D1DEG1(TS=T,A1,PSAT)
05280  N1(I+1)=N1(I)+CN*XMD
05290  F(I+1)=F(I)+DF*XMD
05300  N2(I+1)=PSAT*F(I+1)-N1(I+1)
05310  IF(ABS(N2(I+1)-NGAS)/NGAS).LT.1.E-5) GO TO 604
05320  IF(IT.EQ.2) GO TO 97
05330  IF(IT.EQ.1) GO TO 93
05340  99  DF1=DF1/2.
05350  N2=N2(I+1)-NGAS
05360  SIGM=N23/ABS(N23)
05370  FFAC1=FFAC1-SIGN*DF1
05380  FRAC=FRAC1
05390  IT=C
05400  GO TO 926
05410  97  IF(N2(I+1).LT.NGAS) GO TO 98
05420  90  GO TO 99
05430  98  DF2=DF2/2.
05440  N2=N2(I+1)-NGAS
05450  SIGM=N23/ABS(N23)
05460  FRAC2=FFAC2-SIGN*DF2
05470  FFAC=FRAC2
05480  IT=1
05490  GO TO 928
05500  604  L(I+1)=L(I)+XMD
05510  TGSU=TGSU*K+DGT
05520  GSUM=GSUM+DG
05530  A2(7)=L(I+1)
05540  T1=TSAT
05550  607  IF(ADF(I).LT.1.) GO TO 605
IF(18,F0,0)
05999 1 CALL QCAPPA(L1,L2,L3,LP,QLC1,QLC2,QLC3,QLCT)
06000 CALL RINXY(L1,L2,L3,LP,QLC,QLC1,QLC2,QLC3,QLCT,IR)
06010 DX=LP-L(N1)
06020 G1(N3)=DX/DLP*GIN3
06030 804 A3(N3)=G1(N3)
06040 A4(N3)=NX
06050 MQ=0
06060 GO TO 866
06070 805 A4(N3)=A4N3
06080 A3(N3)=GIN3
06090 PGR=PSAT-PVR
06100 LPD=A2(1)-LP
06110 RNP=PGR*A2(4)/(T2+460.)/12.
06120 X4=A2(I+1)-FNR
06130 GO TO 3000
06140 605 PGR=PSAT-PVR
06150 LPD=0.
06160 RNP=PGR*A2(4)/(T2+460.)/12.
06170 X4=A2(I+1)-FNR
06180 3000 CONTINUE
06190 IF(IFREZE.EQ.2) GO TO 889
06200 501 CONTINUE
06210 IF(A2(9).LT.-0.5) GO TO 80
06220 C IF(A2(11).GT.0.001) GO TO 90
06230 C CALCULAT FFSEVFDR HEATEP INPUT
06240 IF(T3.LE.A2(10)) A2(11)=A2(8)
06250 IF(T3.GE.A2(9)) A2(11)=0.0
06260 100 Q=A2(11)
06270 80 CONTINUE
06280 210 IF(N4.GT.0) GO TO 220
06290 CALL TMEFL(T1,T2,T1,T2,T1,T2,T1,T2)
06300 PGR=QLC*QLC(LP,A2(16),A2(7),LCOMD,X6,LEFF)
06310 IF(IFREZE.EQ.1) GO TO 219
06320 WRITE(6,221) A2(12)
06330 221 FORMAT(18,H,*19X,HEAT PIPE NUMER ,F2.0,17H CHARACTERISTICS)
06340 WRITE(6,222) A2(11),A2(3),A2(4)
06350 222 FORMAT(18,H,TOTAL LFT6TH=,F6.2,*3/H,11H GAS V/L= ,F5.3,8HIN**3/IN,
06360 C18H RES VOLUME= ,F6.3,5H TV-3)
06370 WRITE(6,223) A2(5)
06380 223 FORMAT(18,H GAS INVENTORY = ,F4.8H FT-1B/R,24H RESERVOIR IS W
06390 CICKFD)
06820 WRITE(6,208) TP,T1
06830 208 FORMAT(18H RES TEMPERATURE=,F7.2,2H F)
06840 122H EVAP SIDE VAPOF TEMP=,F7.2,2H F)
06850 WRITE(6,209) X5,X6,LEFF,CLC
06860 209 FORMAT(1CH SUP QIN=,F8.7,19H BTU/HR, SUM QOUT=,FR.2,7H BTU/HR)
06870 1 4X,5HLEFF= ,F6.7,6H INCHES,2X,6HLEFF=,F8.1,12H BTU-INCH/HR)
06880 420 CONTINUE
06890 220 CONTINUE
06900 RETURN
06910 END
06920 C
06930 C
06940 C THIS SUBROUTINE CALCULATE THE HEAT FLOW RATE INTO AND OUT OF
06950 C THE VAPOR NODE
06960 C
06970 C SUBROUTINE CMETER(TV,T,G,QOUT,QIN,N3)
06980 C DIMENSION T(80),G(80)
06990 C
07000 X5=0.
07010 X6=0.
07020 G0 100 I=1,N3
07030 IF(X7.LE.G.0.) X5=X5-X7
07040 IF(X7.GT.0.) X6=X6+X7
07050 CIN=X5
07060 QOUT=X6
07070 RETURN
07080 END
07090 C
07100 C
07110 C THIS FUNCTION CALCULATES THE Q*EFFECTIVE(POWER-LENGTH)
07120 C CAPACITY, E.G., BTU/HR-INCH OR WATT-METER
07130 C
07140 C
07150 C FUNCTION QL(A,P,C,D,F,Z)
07160 C
07170 0=0.6-C
07180 A2=0.6+B
07190 A3=A-0.6
07200 IF(C.GE.P.AND.C.LT.A) GO TO 100
07210 C
07220 IF(C.GE.P) GO TO 101
GO TO 102

2=C
CL*Z+F
RETURN
END

THIS SUBROUTINE CALCULATES THE ROOTS OF A QUADRATIC EQUATION
WHICH RESULTS FROM A QUADRATIC FIT OF THREE VALUES OF
CALCULATED POWER-LENGTH CAPACITIES.

SURROGATE QUADPA(L1,L2,L3,LP,QLC1,QLC2,QLC3,QLCT)
REAL L1,L2,L3,LP,L5
F1=QLC1-QLCT
F2=QLC2-QLCT
F3=QLC3-QLCT
CALCULATE THE COEFFICIENTS OF F(Z)=A+B*Z+C*Z**2

A1=(F1*(L2-L3)+F2*(L3-L1)+F3*(L1-L2))
A2*(L1**2*(L2-L3)+L2**2*(L3-L1)+L3**2*(L1-L2))
IF(A2.FE.0.) PRINT 10,L1,L2,L3,F1,F2,F3
FOR F=1X/3{F22.15,2X})
A=1/2
E={F2-F3)-A*(L2**2-L3**2))/(L2-L3)
C=F3-A*L3**2-B*L3
A2=2.*A
Y=SQRT(B**2-4.*A*C)
Z1=(-B+Y)/2
Z2=(-B-Y)/2
IF(Z1.LT.0.OR.Z1.GT.L3) GO TO 100
IF(Z1.LT.L1) GO TO 100
LP=Z1
GO TO 101
LP=Z2
CONTINUE
RETURN
END

THIS SUBROUTINE PERFORMS A QUADRATIC BINARY SEARCH OF THE
ECLIPSED POWER-LENGTH CAPACITY.

SURROGATE BINARY(L1,L2,L3,LP,QLC1,QLC2,QLC3,QLCT,IR)
REAL L1,L2,L3,L4,L5,LP
07660 IF(IA.EQ.0) GO TO 200
07670 IA=0
07680 F1=CLC1-QLCT
07690 F2=CLC2-QLCT
07700 F3=QLC3-QLCT
07710 F4=QLC-QLCT
07720 IF(LP .GT. L2) GO TO 201
07730 L5=L3
07740 L3=L2
07750 L2=LP
07760 L4=L5
07770 F2=F4
07780 F3=QLC2-QLCT
07790 F4=QLC3-QLCT
07800 SIGN1=F1*F2
07810 SIGN2=F3*F2
07820 SIGN3=F2*F4
07830 IF(SIGN1.LT.0) GO TO 202
07840 IF(SIGN2.LT.0) GO TO 203
07850 IF(SIGN3.LT.0) GO TO 204
07860 202 EX=(L7-L1)/2.
07870 L3=L2
07880 L2=L1+DX
07890 QLC3=QLC
07900 GO TO 300
07910 203 CY=(L3-L2)/2.
07920 L1=L2
07930 L2=L2+DX
07940 QLC1=QLC
07950 QLC3=QLC2
07960 GO TO 300
07970 204 PX=(L4-L2)/2.
07980 L1=L2
07990 L2=L2+DX
08000 L3=L4
08010 QLC1=QLC
08020 QLC3=QLC3
08030 GO TO 300
08040 201 L5=L3
08050 L3=LP
08060 L4=L5
08070 F3=QLC-QLCT
### VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>Conductance (G)</th>
<th>Gains</th>
<th>Heat (W)</th>
<th>Cooling (W)</th>
</tr>
</thead>
</table>

#### Heat Pipe Number 1 Characteristics

- Total Length: 40.000 m
- Gas V/L: .1151 in³/in³
- Res Volume: 6.710 in³
- Vapor Temp: 74.91 °F
- Total Press: 141.49 psia
- Gas LH: 11.87 in
- Dry Evap LH: 0.00 in
- Reservoir Temp: -97.69 °F
- Res Gas Press: 140.05 psia
- Sum Qin: 136.32 BTU/hr
- Sum Qout: 136.32 BTU/hr
- Eff: 0.50000
- Qref: 2224.1 BTU-in/HR
- Reservoir Gas Nr: 142164
- Heat Pipe Nr: 0.36852
- Residual: 0.00001

#### Total Heat Exchange to Boundaries: -1.35907E+02

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>QB</td>
<td>-1.35907E+02</td>
</tr>
<tr>
<td>QB</td>
<td>36</td>
</tr>
</tbody>
</table>

---

**Note:** The table and calculations are based on thermal and heat pipe test case data, indicating a complex system simulation or testing scenario.
### VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

<table>
<thead>
<tr>
<th>Time</th>
<th>DTimeU</th>
<th>CSnum</th>
<th>TempCC(</th>
<th>RlCC(</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.2000E+00</td>
<td>7.22656E-03</td>
<td>9.76562E-03</td>
<td>3.24291E+00</td>
<td>9.04152E-02</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>T</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
</tr>
</thead>
<tbody>
<tr>
<td>23</td>
<td>-3.4600E+10</td>
<td>-2</td>
<td>-3</td>
<td>-4</td>
<td>-5</td>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>28</td>
<td>3.4600E+10</td>
<td>-7</td>
<td>-8</td>
<td>-9</td>
<td>-10</td>
<td>3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>34</td>
<td>2.0000E+00</td>
<td>17</td>
<td>18</td>
<td>19</td>
<td>20</td>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>2.5600E+00</td>
<td>22</td>
<td>23</td>
<td>24</td>
<td>25</td>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>7.7700E+00</td>
<td>32</td>
<td>33</td>
<td>34</td>
<td>35</td>
<td>7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>2.5200E+00</td>
<td>37</td>
<td>38</td>
<td>39</td>
<td>40</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>2.5200E+00</td>
<td>42</td>
<td>43</td>
<td>44</td>
<td>45</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>46</td>
<td>2.5200E+00</td>
<td>47</td>
<td>48</td>
<td>49</td>
<td>50</td>
<td>0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>51</td>
<td>0.0000</td>
<td>52</td>
<td>53</td>
<td>54</td>
<td>55</td>
<td>0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>56</td>
<td>0.0000</td>
<td>57</td>
<td>58</td>
<td>59</td>
<td>60</td>
<td>5.61818E-03</td>
<td></td>
<td></td>
</tr>
<tr>
<td>61</td>
<td>5.0000E+00</td>
<td>62</td>
<td>63</td>
<td>64</td>
<td>65</td>
<td>5.0000E+00</td>
<td></td>
<td></td>
</tr>
<tr>
<td>66</td>
<td>5.0000E+00</td>
<td>67</td>
<td>68</td>
<td>69</td>
<td>70</td>
<td>5.0000E+00</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**HEAT PIPE NUMBER 1 CHARACTERISTICS**

- TOTAL LENGTH = 40.000IN
- GAS V/L = .115IN**3/IN
- RES VOLUME = 6.710 IN-3
- GAS INVENTORY = .2530 FT-18/R
- RESERVOIR IS WICKED
- VAPOR TEMP = 68.88 F
- TOTAL PRESS = 127.08 PSIA
- GAS LGTH = 22.000IN
- DRY EVAP LGTH = 0.00 IN
- RESERVOIR TEMP = -97.62 F
- RES GAS PRESS = 125.68 PSIA
- SUM QIN = 8.55 BTU/HR
- SUM QOUT = 8.55 BTU/HR
- LEFF = 11.25 INCHES
- QLEFF = 96.2 BTU-INCH/HR
- RESERVOIR GAS NR = .293926
- HEAT PIPE NR = .059075
- RESIDUAL = -.000001

**TOTAL HEAT EXCHANGE TO BOUNDARIES**
- Q = -6.65372E+01
- QB = 36 = -6.09930E-11
**TRW SYSTEMS IMPROVED NUMERICAL DIFFERENCING ANALYZER (SINDA) CDC 6000 VERSION**

**VARIABLE CONDUCTANCE HEAT PIPE TEST CASE**

<table>
<thead>
<tr>
<th>T</th>
<th>23 = 1.0576E+02 T</th>
<th>24 = 1.0587E+02 T</th>
<th>25 = -1.0476E+02 T</th>
<th>26 = -1.0333E+02 T</th>
<th>27 = -1.0169E+02 T</th>
</tr>
</thead>
<tbody>
<tr>
<td>G</td>
<td>23 = 1.0556E+02 T</td>
<td>24 = 1.0265E+02 T</td>
<td>25 = -1.0700E+02 T</td>
<td>26 = -1.0650E+02 T</td>
<td>27 = 9.0821E+02 T</td>
</tr>
<tr>
<td>G</td>
<td>23 = 1.0472E+02 T</td>
<td>24 = 1.0330E+02 T</td>
<td>25 = -1.0167E+02 T</td>
<td>26 = -9.9891E+01 T</td>
<td>27 = -9.8001E+01 T</td>
</tr>
<tr>
<td>T</td>
<td>23 = 9.5048E+01 T</td>
<td>24 = 9.4561E+01 T</td>
<td>25 = -8.1798E+01 T</td>
<td>26 = -2.5301E+01 T</td>
<td>27 = 3.1162E+01 T</td>
</tr>
</tbody>
</table>

**HEAT PIPE NUMBER 1 CHARACTERISTICS**

<table>
<thead>
<tr>
<th>TOTAL LENGTH</th>
<th>40.0000 IN</th>
<th>GAS V/L</th>
<th>0.115 IN**3/IN</th>
<th>PES VOLUME</th>
<th>6.710 IN-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>GAS INVENTORY</td>
<td>2530 FT-18/R INVENTORY IS WICKED</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>VAPOR TEMP</td>
<td>62.5 F</td>
<td>TOTAL PRESS</td>
<td>113.62 PSIA</td>
<td>GAS LGTH</td>
<td>29.6000 IN</td>
</tr>
<tr>
<td>RESERVOIR TEMP</td>
<td>-100.56 F</td>
<td>RESERVOIR PRESS</td>
<td>112.40 PSIA</td>
<td></td>
<td></td>
</tr>
<tr>
<td>SUM QIN</td>
<td>0.34 BTU/HR</td>
<td>SUM QOUT</td>
<td>0.34 BTU/HR</td>
<td>LEFF</td>
<td>5.20 INCHES</td>
</tr>
<tr>
<td>RESERVOIR GAS NR</td>
<td>174851</td>
<td>HEAT PIPE NR</td>
<td>0.078148</td>
<td>RESIDUAL</td>
<td>0.000000</td>
</tr>
<tr>
<td>TOTAL HEAT EXCHANGE TO BOUNDARIES</td>
<td>-1.10248E+01</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>QB</td>
<td>35 = -1.10248E+01 QB</td>
<td>36 = -1.80425E-11</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### TRW SYSTEMS IMPROVED NUMERICAL DIFFERENCING ANALYZER (SINDA) CDC 6600 VERSION

**VARIABLE CONDUCTANCE HEAT PIPE TEST CASE**

<table>
<thead>
<tr>
<th>TIME</th>
<th>DTME</th>
<th>CSmin</th>
<th>TEMPCC</th>
<th>RELXCC</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>23</td>
<td>-1.10685E+02</td>
<td>24</td>
<td>-1.10423E+02</td>
<td>25</td>
<td>-1.09866E+02</td>
</tr>
<tr>
<td>28</td>
<td>-1.07016E+02</td>
<td>29</td>
<td>-1.03645E+02</td>
<td>30</td>
<td>-1.03885E+02</td>
</tr>
<tr>
<td>34</td>
<td>-1.02045E+02</td>
<td>35</td>
<td>-1.01572E+02</td>
<td>36</td>
<td>-1.10159E+02</td>
</tr>
<tr>
<td>50</td>
<td>-1.09535E+02</td>
<td>51</td>
<td>-1.09068E+02</td>
<td>52</td>
<td>-1.08105E+02</td>
</tr>
<tr>
<td>10</td>
<td>-1.03783E+02</td>
<td>11</td>
<td>-1.01152E+02</td>
<td>12</td>
<td>-6.77315E+01</td>
</tr>
<tr>
<td>15</td>
<td>6.09146E+01</td>
<td>16</td>
<td>6.08954E+01</td>
<td>17</td>
<td>6.09352E+01</td>
</tr>
<tr>
<td>20</td>
<td>6.09382E+01</td>
<td>21</td>
<td>-1.10416E+02</td>
<td>22</td>
<td>-8.86609E+01</td>
</tr>
</tbody>
</table>

### HEAT PIPE NUMBER 1 CHARACTERISTICS

- **TOTAL LENGTH**: 40.00 IN
- **GAS V/L**: .115 IN**3**/IN
- **RES VOLUME**: 6.710 IN**3**
- **GAS INVENTORY**: .2530 FT-LB/RESERVOIR IS WICKED
- **LOCATION OF ICE PLUG**: 4.00 IN POSITION OF GAS FRONT 40.00 IN DRY EVAP LNGTH 0.00 IN
- **GAS IN RES SIDE**: 1.186676 GAS IN EVAP SIDE 0.066324 GAS RESIDUAL 0.000000
- **TOTAL PRESSURE IN RES SIDE**: 112.87 PSIA
- **TOTAL PRESSURE ON EVAP SIDE**: 111.32 PSIA
- **PEV-PRE**: -1.55 PSIA
- **RES TEMPERATURE**: -102.04 F
- **EVAP SIDE VAPOR TEMP**: 60.94 F
- **SUM QIN**: 0.00 BTU/HR, SUM QOUT: 0.00 BTU/HR
- **LEFF**: 0.00 INCHES QLEFF: 0.0 BTU-INCH/HR
- **TOTAL HEAT EXCHANGE TO BOUNDARIES**: -7.33479E+00
- **Q1**: 35 = -7.33479E+00
- **Q2**: 36 = 0.00000
TRW SYSTEMS IMPROVED NUMERICAL DIFFERENCING ANALYZER (SINDA)
CDC 6600 VERSION

VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

*********** TIME= 4.00000E+00
DTIME= 7.2265E-03 CSGMIN( 33)= 9.76562E-03 TEMPC( 33)= 6.29288E-02 RELXCC( 18)= 6.29812E-02

T 23=-1.19025E+02 T 24=-1.19121E+02 T 25=-1.19083E+02 T 26=-1.18897E+02 T 27=-1.18521E+02
T 28=-1.17876E+02 T 29=-1.16824E+02 T 30=-1.15145E+02 T 31=-1.12481E+02 T 33= 4.28057E+01
T 34=-1.13432E+02 T 1=1.19018E+02 T 2=1.18206E+02 T 3=1.19006E+02 T 4=1.19104E+02
T 5=-1.19063E+02 T 6=-1.18872E+02 T 7=-1.18486E+02 T 6=-1.17823E+02 T 9=-1.16742E+02
T 10=-1.15016E+02 T 11=-1.12278E+02 T 12=-9.81778E+01 T 13=-4.30292E+01 T 14= 1.20647E+01
T 15= 4.23155E+01 T 16= 4.27708E+01 T 17= 4.28085E+01 T 18= 4.28116E+01 T 19= 4.28114E+01
T 20= 4.28626E+01 T 22=-1.18252E+02 T 32=-9.67798E+01 T 35=-1.20000E+02 T 36= 4.28691E+01

G -11= 8.22000E-11 G -12= 5.00000E-01 G -13= 2.00000E+00 G -14= 2.00000E+00 G -15= 2.00000E+00
G -16= 2.00000E+00 G -17= 2.00000E+00 G -18= 2.00000E+00 G -19= 2.00000E+00 G -20= 2.00000E+00
G -21= 2.00000E+00 G -22= 2.00000E+00 G -23= 2.00000E+00 G -24= 2.00000E+00 G -25= 2.00000E+00
G -26= 2.00000E+00 G -27= 2.00000E+00 G -28= 2.00000E+00 G -29= 2.00000E+00 G -30= 2.00000E+00
G -31= 2.00000E+00 G -32= 2.00000E+00 G -33= 2.00000E+00 G -34= 2.00000E+00 G -35= 2.00000E+00
G -36= 2.00000E+00 G -37= 2.00000E+00 G -38= 2.00000E+00 G -39= 2.00000E+00 G -40= 7.77000E-02
G -51= 0.00000 G -52= 0.00000 G -53= 0.00000 G -54= 0.00000 G -55= 0.00000
G -56= 0.00000 G -57= 0.00000 G -58= 0.00000 G -59= 0.00000 G -60= 0.00000
G -61= 0.00000 G -62= 0.00000 G -63= 0.00000 G -64= 0.00000 G -65= 0.00000
G -66= 0.00000 G -67= 0.00000 G -68= 0.00000 G -69= 0.00000

HEAT PIPE NUMBER 1 CHARACTERISTICS
TOTAL LENGTH= 40.000IN GAS V/L= 115IN**3/IN RES VOLUME= 6.710IN**3
GAS INVENTORY= .2530 FT-LB/RESERVOIR IS WICKED
LOCATION OF ICE PLUG= 4.00 IN POSITION OF GAS FRONT= 40.00 IN DRY EVAP LGTH= 0.00 IN
GAS IN RES SIDE= 186676 GAS IN EVAP SIDE= .066324 GAS RESIDUAL= 0.00000
TOTAL PRESSURE IN RES SIDE= 108.90 PSIA TOTAL PRESSURE ON EVAP SIDE= 99.38 PSIA PEV-PRE= -9.53 PSIA
RES TEMPERATURE= -113.43 F EVAP SIDE VAPOR TEMP= 42.81 F
SUM QIN= 0.00 BTU/HR SUM QOUT= 0.00 BTU/HR LEFF= 0.00 INCHES QLEFF= 0.0 BTU-INCH/HR

TOTAL HEAT EXCHANGE TO BOUNDARIES =-1.65910E+00
QB 35=-1.65910E+00 QB 36= 0.00000
## VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>Temperature (°C)</th>
<th>Gas Mass Flow (kg/s)</th>
<th>Heat Transfer Rate (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>23</td>
<td>-1.19138E+02</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
<tr>
<td>28</td>
<td>-1.14101E+02</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
<tr>
<td>34</td>
<td>-1.14093E+02</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
<tr>
<td>5</td>
<td>1.19084E+02</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
<tr>
<td>10</td>
<td>5.39366E+01</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
<tr>
<td>15</td>
<td>8.33012E+01</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
<tr>
<td>20</td>
<td>8.33249E+01</td>
<td>3.46000E+00</td>
<td>4.20000E+00</td>
</tr>
</tbody>
</table>

## HEAT PIPE NUMBER 1 CHARACTERISTICS

- **Total Length**: 40.00 in
- **Gas V/L**: 0.115 in**3**/in
- **Gas Volume**: 6.710 in**3**
- **Gas Inventory**: 0.2530 ft-lb
- **Reservoir**: WICKED
- **Location of Ice Plug**: 4.00 in position of gas front
- **Gas in Res Side**: 186676 ft-lb in evap side
- **Gas Residual**: 0.06632 ft-lb
- **Total Pressure in Res Side**: 108.68 psia
- **Total Pressure on Evap Side**: 135.45 psia
- **Evap Pre**: 44.77 psia
- **Res Temp**: -114.09°F
- **Evap Side Vapor Temp**: 79.82°F
- **Sum Qin**: 105.03 Btu/hr
- **Sum Gout**: 105.03 Btu/hr
- **LEFF**: 12.25 inches
- **QLEFF**: 1286.8 Btu/inch/hr

## Heat Transfer

- **Total Heat Exchange to Boundaries**: -2.86253E+01
- **Q8**: 35×-2.86253E+01
- **LEFF**: 12.25 inches
- **QLEFF**: 1286.8 Btu/inch/hr
### VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

<table>
<thead>
<tr>
<th>T</th>
<th>DTIMEU</th>
<th>CGMIN</th>
<th>TEMPCC(</th>
<th>RELXCC(</th>
</tr>
</thead>
<tbody>
<tr>
<td>23</td>
<td>7.22566E-03</td>
<td>9.76562E-03</td>
<td>6.18198E-01</td>
<td>8.71805E-02</td>
</tr>
<tr>
<td>28</td>
<td>3.76151E+01</td>
<td>3.14528E+01</td>
<td>3.09649E+01</td>
<td>3.09649E+01</td>
</tr>
<tr>
<td>34</td>
<td>1.11414E+02</td>
<td>1.11414E+02</td>
<td>1.11414E+02</td>
<td>1.11414E+02</td>
</tr>
<tr>
<td>5</td>
<td>9.1596E+01</td>
<td>6.8508E+01</td>
<td>7.7775E+01</td>
<td>4.71821E+01</td>
</tr>
<tr>
<td>10</td>
<td>1.03067E+02</td>
<td>1.03067E+02</td>
<td>1.03067E+02</td>
<td>1.03067E+02</td>
</tr>
<tr>
<td>15</td>
<td>1.13073E+02</td>
<td>1.13073E+02</td>
<td>1.13073E+02</td>
<td>1.13073E+02</td>
</tr>
<tr>
<td>20</td>
<td>1.13099E+02</td>
<td>1.13099E+02</td>
<td>1.13099E+02</td>
<td>1.13099E+02</td>
</tr>
</tbody>
</table>

### HEAT PIPE NUMBER 1 CHARACTERISTICS

- **Total Length**: 40.0001N
- **Gas V/L**: 0.11515**3**
- **In Res Volume**: 6.710 IN**3**
- **Gas Inventory**: 0.2530 FT-LB/RESERVOIR
- **WICKED**
- **Location of Ice Plug**: 4.00 IN POSITION OF GAS FRONT
- **Total Pressure in Res Side**: 108.35 PSIA
- **Total Pressure in Evap Side**: 244.10 PSIA
- **Evap Side Vapor Temp**: 108.76 F
- **Sum Qin**: 13G.09 BTU/HR
- **Sum Qout**: 130.09 BTU/HR
- **Leff**: 14.33 INCHES
- **Qleff**: 1864.4 BTU-INCH/HR

**Total Heat Exchange to Boundaries**: -1.17181E+02
**QB**: 35 = -1.17181E+02
**QB**: 36 = -1.11868E-10
### VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

<table>
<thead>
<tr>
<th>TIME</th>
<th>DTIME</th>
<th>CSGMIN</th>
<th>TEMPCC</th>
<th>RELXCC</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.260E+00</td>
<td>7.2265E+03</td>
<td>9.7656E+03</td>
<td>1.5893E+00</td>
<td>9.0378E+02</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>T</th>
<th>23 = -5.563E+01</th>
<th>24 = -4.3238E+01</th>
<th>25 = -2.3456E+01</th>
<th>26 = 4.50673E+00</th>
<th>27 = 5.28458E+01</th>
</tr>
</thead>
<tbody>
<tr>
<td>28</td>
<td>6.15086E+01</td>
<td>29 = 6.18917E+01</td>
<td>30 = 6.19099E+01</td>
<td>31 = 6.19401E+01</td>
<td>32 = 6.1914E+01</td>
</tr>
<tr>
<td>34</td>
<td>-1.09175E+02</td>
<td>1 = 9.34923E+01</td>
<td>2 = 6.19749E+01</td>
<td>3 = 5.43996E+01</td>
<td>4 = 4.22482E+01</td>
</tr>
<tr>
<td>5</td>
<td>6.22415E+01</td>
<td>6 = 7.40668E+00</td>
<td>7 = 6.00703E+01</td>
<td>8 = 6.95037E+01</td>
<td>9 = 6.99232E+01</td>
</tr>
<tr>
<td>10</td>
<td>6.09433E+01</td>
<td>11 = 6.99765E+01</td>
<td>12 = 7.23433E+01</td>
<td>13 = 7.31535E+01</td>
<td>14 = 7.34602E+01</td>
</tr>
<tr>
<td>15</td>
<td>1.06835E+02</td>
<td>16 = 1.40737E+02</td>
<td>17 = 1.43545E+02</td>
<td>18 = 1.43777E+02</td>
<td>19 = 1.43797E+02</td>
</tr>
<tr>
<td>20</td>
<td>1.43798E+02</td>
<td>22 = -6.14764E+01</td>
<td>32 = 6.45429E+01</td>
<td>35 = -5.0000E+01</td>
<td>36 = 7.31560E+01</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
<th>G</th>
</tr>
</thead>
<tbody>
<tr>
<td>-1 = 3.46000E-10</td>
<td>-2 = 3.46000E-10</td>
<td>-3 = 3.46000E-10</td>
<td>-4 = 3.46000E-10</td>
<td>-5 = 3.46000E-10</td>
<td></td>
</tr>
<tr>
<td>-6 = 3.46000E-10</td>
<td>-7 = 3.46000E-10</td>
<td>-8 = 3.46000E-10</td>
<td>-9 = 3.46000E-10</td>
<td>-10 = 3.46000E-10</td>
<td></td>
</tr>
<tr>
<td>-11 = 3.46000E-10</td>
<td>12 = 5.00000E-01</td>
<td>13 = 2.00000E+00</td>
<td>14 = 2.00000E+00</td>
<td>15 = 2.00000E+00</td>
<td></td>
</tr>
<tr>
<td>16 = 2.00000E+00</td>
<td>17 = 2.00000E+00</td>
<td>18 = 2.00000E+00</td>
<td>19 = 2.00000E+00</td>
<td>20 = 2.00000E+00</td>
<td></td>
</tr>
<tr>
<td>21 = 2.00000E+00</td>
<td>22 = 5.00000E-01</td>
<td>23 = 2.56000E+00</td>
<td>24 = 2.56000E+00</td>
<td>25 = 2.56000E+00</td>
<td></td>
</tr>
<tr>
<td>26 = 2.56000E+00</td>
<td>27 = 2.56000E+00</td>
<td>28 = 2.56000E+00</td>
<td>29 = 5.05000E+02</td>
<td>30 = 5.25000E+02</td>
<td></td>
</tr>
<tr>
<td>31 = 7.77000E-02</td>
<td>32 = 2.52000E-01</td>
<td>33 = 2.52000E-01</td>
<td>34 = 2.52000E-01</td>
<td>35 = 2.52000E-01</td>
<td></td>
</tr>
<tr>
<td>41 = 2.52000E-01</td>
<td>42 = 2.52000E-01</td>
<td>43 = 2.52000E-02</td>
<td>44 = 2.52000E-01</td>
<td>45 = 2.52000E-01</td>
<td></td>
</tr>
<tr>
<td>46 = 2.52000E-01</td>
<td>47 = 2.52000E-01</td>
<td>48 = 2.52000E-01</td>
<td>49 = 5.48480E-11</td>
<td>50 = 0.00000</td>
<td></td>
</tr>
<tr>
<td>51 = 0.00000</td>
<td>52 = 0.00000</td>
<td>53 = 0.00000</td>
<td>54 = 0.00000</td>
<td>55 = 0.00000</td>
<td></td>
</tr>
<tr>
<td>56 = 1.96346E+00</td>
<td>57 = 5.00000E+00</td>
<td>58 = 5.00000E+00</td>
<td>59 = 5.00000E+00</td>
<td>60 = 5.00000E+00</td>
<td></td>
</tr>
<tr>
<td>61 = 5.00000E+00</td>
<td>62 = 5.00000E+00</td>
<td>63 = 5.00000E+00</td>
<td>64 = 2.87321E+00</td>
<td>65 = 0.00000</td>
<td></td>
</tr>
<tr>
<td>66 = 0.00000</td>
<td>67 = 0.00000</td>
<td>68 = 0.00000</td>
<td>69 = 0.00000</td>
<td>70 = 0.00000</td>
<td></td>
</tr>
</tbody>
</table>

### HEAT PIPE NUMBER 1 CHARACTERISTICS

TOTAL LENGTH = 40.001N  GAS V/L = .115IN**3/IN  RES VOLUME = 6.710 IN-3  GAS INVENTORY = .2530 Ft-18/R RESERVOIR IS WICKED  VAPOR TEMP = 73.21 F  TOTAL PRESS = 137.31 PSIA  GAS LGTH = 13.21IN  DRY EVAP LGTH = 10.85 IN  RESERVOIR TEMP = -109.17 F  RES GAS PRESS = 136.47 PSIA  SUM QIN = 97.87 BTU/HR  SUM QOUT = 97.87 BTU/HR  LEFF = 10.22 INCHES  QLEFF = 1000.0 BTU-INCH/HR  RESERVOIR GAS NR = .217514 HEAT PIPE NR = .035488 RESIDUAL = -.00002  

TOTAL HEAT EXCHANGE TO BOUNDARIES = -8.98458E+01  QB = 35 = -8.98456E+01  QB = 36 = 5.86624E-11
VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

**TIME** = 6.000000E+00  
**DTIME** = 7.22E-03  
**CSGMIN** ( 33) = 9.76562E-03  
**TEMPCC** ( 33) = 1.62094E+00  
**RELCXCC** ( 18) = 1.27446E-02

| T | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       | G       |
|---|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|---------|
| 23 | 3.90734E+01 | 24 | -3.22309E+01 | 25 | -1.84603E+01 | 26 | 7.90362E+00 | 27 | 5.88763E+01 | 28 | 6.57098E+01 | 29 | 6.60123E+01 | 30 | 6.60271E+01 | 31 | 6.60584E+01 | 32 | 6.60851E+01 | 33 | 3.22351E+02 |
| 34 | 8.28823E+01 | 54 | 1.67551E+01 | 6 | 1.13274E+01 | 7 | 6.65825E+01 | 8 | 7.40897E+01 | 9 | 7.44230E+01 | 10 | 7.44393E+01 | 11 | 7.44738E+01 | 12 | 7.69487E+01 | 13 | 7.78084E+01 | 14 | 7.96794E+01 | 15 | 2.85532E+02 |
| 16 | 3.19301E+02 | 22 | 4.72502E+01 | 32 | 6.87448E+01 | 35 | 5.00000E+01 | 36 | 7.78033E+01 |

HEAT PIPE NUMBER 1 CHARACTERISTICS

TOTAL LENGTH = 40.00 IN  
GAS V/L = .115 IN**3/IN  
RES VOLUME = 6.710 IN-3  
GAS INVENTORY = .2530 FT-18/IN  
RESERVOIR TS = WICKED  
VAPOR TEMP = -77.84 F  
TOTAL PRESS = 146.61 PSIA  
GAS LGTH = 13.02 IN  
DRY EVAP LGTH = 11.82 IN  
RESERVOIR TEMP = -82.88 F  
RES GAS PRESS = 146.09 PSIA  
SUM QIN = 101.74 BTU/HR  
SUM QOUT = 101.74 BTU/HR  
LEFF = 9.83 INCHES  
QLEFF = 999.9 BTU-INCH/HR  
RESERVOIR GAS NR = .216617  
HEAT PIPE NR = .036383  
RESIDUAL = -.00000  

TOTAL HEAT EXCHANGE TO BOUNDARIES = -9.88990E+01  
QB = -9.88990E+01  
QB = -9.54969E-12
### VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

<table>
<thead>
<tr>
<th>Time</th>
<th>Temp</th>
<th>Relax</th>
</tr>
</thead>
<tbody>
<tr>
<td>T</td>
<td></td>
<td></td>
</tr>
<tr>
<td>DTIME</td>
<td>7.2265E-03</td>
<td>1.5662E+00</td>
</tr>
<tr>
<td>CGSMIN(33)</td>
<td>9.76562E-03</td>
<td></td>
</tr>
<tr>
<td>TEMPCC(33)</td>
<td></td>
<td>1.23151E-02</td>
</tr>
</tbody>
</table>

#### HEAT PIPE NUMBER 1 CHARACTERISTICS

- **Total Length**: 40.0000 ft
- **GAS V/L**: 0.115
- **RES Volume**: 6.710 in³
- **GAS INVENTORY**: 0.2530 ft³
- **RESERVOIR IS**: WICKED
- **VAPOR TEMP**: 81.16 F
- **TOTAL PRESS**: 157.12 psia
- **GAS LNGTH**: 13.251 in
- **DRY EVAP LNGTH**: 11.92 in
- **RESERVOIR TEMP**: -63.71 F
- **RES GAS PRESS**: 152.10 psia
- **SUM QIN**: 103.37 BTU/HR
- **SUM QOUT**: 103.37 BTU/HR
- **LEFF**: 9.67 INCHES
- **QLEFF**: 999.1 BTU-INCH/HR
- **RESERVOIR GAS NR**: 0.214618
- **HEAT PIPE NR**: 0.038382
- **RESIDUAL**: -0.00000

#### HEAT EXCHANGE TO BOUNDARIES

- **Qb**: 35 = -1.02141E+02
- **Qb**: 36 = -4.72937E-11
### VARIABLE CONDUCTANCE HEAT PIPE TEST CASE

**TIME** = 8.00000E+00  
**DTIME** = 7.22656E-03  
**SGMIN** = 33  
**TEMPCC** = 9.76562E-03  
**RELC** = 1.53594E+00  
**RELCC** = 18  
**RELXCC** = 1.20770E-02

| T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       | T      | G       |
|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|--------|---------|
| 23     | -3.7470E+01 | 24     | -3.1269E+01 | 25     | -1.7890E+01 | 26     | -8.4998E+00 | 27     | 5.8925E+01 |
| 28     | 6.9981E+01  | 29     | 7.0469E+01  | 30     | 7.0493E+01  | 31     | 7.0526E+01  | 32     | 7.0554E+01  |
| 34     | -5.4035E+01 | 35     | -4.9975E+01 | 36     | -4.1836E+01 | 37     | -3.6840E+01 | 38     | -3.0307E+01 |
| 5      | -1.6169E+01 | 6      | 1.1623E+01  | 7      | 6.5585E+00  | 8      | 7.8755E+00  | 9      | 7.9295E+00  |
| 10     | 7.9320E+01  | 11     | 7.9357E+01  | 12     | 8.1950E+01  | 13     | 8.2875E+01  | 14     | 8.8584E+01  |
| 15     | 7.1767E+02  | 16     | 7.52037E+02 | 17     | 7.5488E+02  | 18     | 7.5512E+02  | 19     | 7.5513E+02  |
| 20     | 7.5514E+02  | 22     | 6.2200E+02  | 22     | 6.2200E+02  | 22     | 6.2200E+02  | 22     | 6.2200E+02  |

#### HEAT PIPE NUMBER 1 CHARACTERISTICS

- **TOTAL LENGTH**: 40.00IN  
- **GAS V/L**: .115IN**3*/IN  
- **RES VOLUME**: 6.710 IN=3

- **GAS INVENTORY**: .2530 FT=18/R  
- **RESERVOIR IS WICKED**

- **VAPOR TEMP**: 82.86 F  
- **TOTAL PRESS**: 161.83 PSIA  
- **GAS LGTH**: 13.36IN  
- **DRY EVAP LGTH**: 11.95 IN

- **RESERVOIR TEMP**: -54.04 F  
- **RES GAS PRESS**: 154.94 PSIA

- **SUM QIN**: 104.22 BTU/HR  
- **SUM QOUT**: 104.22 BTU/HR  
- **LEFF**: 9.59 INCHES  
- **QLEFF**: 999.9 BTU-INCH/HR

- **RESERVOIR GAS NR**: 213417  
- **HEAT PIPE NR**: .039533  
- **RESIDUAL**: .000000

**TOTAL HEAT EXCHANGE TO BOUNDARIES**: -1.03615E+02  
**QB**: 35 = -1.03615E+02  
**QB**: 36 = -2.50111E-11
APPENDIX A.5.1

POTENTIAL FOR BUBBLE FORMATION IN CTS HEAT PIPES:
FUNDAMENTALS
INTRODUCTION

In the Anomalies Review and Program Plan of 19 October 1978, there is promised the following analysis: "Calculate number of critical size bubbles which can be generated due to supersaturation in rapid chilldown. Consider both temperature and pressure effects". In what follows the basis for making such a calculation is reviewed.

GAS SATURATION

For dilute solutions of gas in liquid, Henry's Law can be invoked: The partial pressure gas \( i \) in the vapor phase \( P_i \) and the mole fraction \( X_i \) or concentration \( c_i \) of gas \( i \) in the liquid phase are linearly related

\[
P_i = C_i(T) X_i = \left[ c_i(T) / c \right] c_i, \quad i > 2
\]

In the vapor phase the total pressure \( P \) is given by Dalton's Law for an ideal gas mixture

\[
P = \sum_{i=1}^{n} P_i
\]

where the vapor pressure of the solute species is given by

Raoult's Law

\[
P_1 = P_{sat}(T) X_1 = C_1(T) X_1
\]
Often Henry's constant $C_i(T)$ is so large that all $X_i$'s other than $X_1$ are very small, and $X_1 = 1$. The mole fractions in the liquid sum to unity, of course.

$$1 = \sum_{i=1}^{n} X_i \quad \quad c = \sum_{i=1}^{n} c_i$$  \hspace{1cm} (4)

If the liquid contacts a gas mixture of specified total pressure $P$ and specified ratios of noncondensible gas mole fraction,

$$Y_i = P_i / \sum_{i=2}^{n} P_i = P_i / P_g$$  \hspace{1cm} (5)

First, one finds the mole fraction of noncondensibles in the liquid

$$X_g = \sum_{i=2}^{n} \frac{(Y_i / C_i) (P - C_1)}{1 - \sum_{i=2}^{n} (Y_i / C_i) C_1}$$  \hspace{1cm} (6)

The partial pressures in the vapor phase are

$$P_i = Y_i \left[ P - C_1 (1 - X_g) \right]$$  \hspace{1cm} (7)

The mole fractions in the liquid phase are

$$X_i = (Y_i / C_i) \left[ P - C_1 (1 - X_g) \right]$$  \hspace{1cm} (8)
CRITICAL BUBBLE SIZE

The preceding relations are assumed to apply to gas at equilibrium within a critically-sized bubble when the total pressure in the bubble exceeds that in the surrounding liquid by an amount related to the surface tension $\sigma$ and bubble size $r$

$$P - P_{\infty} = \frac{2\sigma(T)}{r} \quad (9)$$

Given a value of $P_{\infty}$, $\sigma(T)$, and $r$ one can find $P$ from Eq. (9) and then proceed as before to establish the set of $X_i$ mole fractions in the liquid immediately surrounding the bubble (the effect of $X_i$ on $\sigma$ is neglected). Conversely given a set of $X_i$, one can use Eqs. (1), (2), and (3) to find $P$ and Eq. (9) to find $r$.

PREVIOUS AND PRESENT STATE VARIABLES

For convenience we choose to set the composition of a (bubbly) liquid by its "Previous state variables" $T'_e$ and $T'_c$ plus the $Y_i$ ratios. The quantity $T'_e$ is the "Previous evaporator temperature". It sets the previous total pressure $P'$

$$P' = P_{\text{sat}}(T'_e) \quad (10)$$

The "Previous (gas-blocked) condenser temperature is $T'_c$". It sets the previous vapor pressure according to Eq. (3) and, as explained below Eq. (4) in Eqs. (5) - (8), with $P'$ and $P'_1$ one can find the previous liquid composition $X'_1$. 

A-83
Present state variables are $T_e$ and $T_c$. The quantity $T_e$ is understood to fix pressure $P_\infty$ by the relation

$$P_\infty = P_{\text{sat}}(T_e) \quad (11)$$

The quantity $T_c$ fixes $\sigma$ and $P_1$.

CRITICAL BUBBLE SIZE IN AN INFINITE LIQUID

Specification of the previous state variables, present state variables, and nonisothermal gas composition leads immediately to the critical size of a single bubble in contact with an infinite liquid. Variables $T_e'$, $T_c'$, and $Y_i'$ leads to a set of $X_i'$ which, in an infinite liquid, are identical to $X_i$. Then, as explained below Eq. (9), one can find $P$ and $r$ using the set of $X_i$ and $P_\infty$ (from $T_e$) and $\sigma$ and $P_1$ (from $T_c$).

$$r_{cr} = \frac{2 \sigma (T_c')}{P - P_\infty} \quad (12)$$

Where $P_\infty$ comes from Eq. (11) and $P$ from Eq. (2) with $P_1$ from Eqs. (1) and (3).

NUMBER OF BUBBLES OF A SPECIFIED SIZE IN A FINITE LIQUID

Let the mole fractions in one mole of liquid mixture be specified by the previous state variables. Imagine that the liquid is supersaturated, that is, there exists a finite $r_{cr}$. Now choose a value $r$ greater than $r_{cr}$. Then Eq. (9), with $P_\infty$ fixed by $T_e$ and $\sigma$ by $T_c$, fixes total pressure $P$. Unfortunately the gas composition in the bubble is unknown so that Eq. (7) cannot be directly applied. However, one can write that the moles of species $i$ remaining in the liquid plus
those in the gas sum to the original amount of species $i$.

$$X_i (1 - N) + N Y_i = X_i'$$  \hspace{1cm} (13)$$

where $N$ is the number of moles in the vapor phase.

Introducing Eqs. (1), (2), and (3) gives

$$X_i + N P_i / P = X_i'$$

$$X_i + (N/P) C_i = X_i$$  \hspace{1cm} (14)$$

where $C_i$ is understood to be $P_{sat}(T_c)$. Solving for $X_i$ gives

$$X_i = \frac{X_i'}{1 + (N/P) C_i}$$  \hspace{1cm} (15)$$

Multiplying both sides by $C_i(T_c)/P$ and summing over $i$ gives a single equation which fixes the unknown $N$.

$$1 = \sum_{i=1}^{n} \frac{C_i X_i'}{P + N C_i}$$  \hspace{1cm} (16)$$

For a single-species noncondensible gas Eq. (16) may be solved explicitly via the quadratic equation. For a binary noncondensible gas mixture, one can employ a computer-aided binary search for the normalizing value of $N$. Once $N$ is determined the number of bubbles follows immediately

$$N_b = \frac{N R u T_c}{P (4/3 \pi r^3)}$$  \hspace{1cm} (17)$$

where $R_u$ is the universal gas constant.
INTRODUCTION

It is desired to calculate the number and size of critical size bubbles which can be generated due to supersaturation. In Part I the fundamentals were developed. Here sample calculations are made for two test cases (1) a condenser depressurization case where evaporator temperature is dropped at constant gas-blocked condenser temperature and (2) a condenser chilldown where the gas-blocked condenser is cooled at constant total pressure.

HENRY'S CONSTANTS

Henry's constants are calculated from extrapolations of a solubility curve from Saaski. Saaski plots mole fraction of the gas in the liquid versus temperature on a log-log scale. Presumably the gas pressure is one atmosphere. Thus one has \( X_i \) for \( P_i = 1 \), and since \( P_i = C_i X_i \), \( C_i = P_i / X_i = 1 / X_i \).

As a check, an Ostwald coefficient reported by Saaski for helium in methanol at 25°C is compared to the graphed value. Ostwald coefficient \( \alpha \) at 25°C is reported to be 0.036. The graph shows \( X_i = 6 \times 10^{-5} \). Hence we expect from the graph that \( C_i = 1/6 \times 10^{-5} = 16670 \text{ atm} \). Ostwald coefficient and Henry number are related by

\[
C_i = \frac{C_i \rho_i R_i T}{\alpha} = \frac{\rho_i R_i T}{M \alpha} \]

\[
C_i = \frac{(0.785)(82.05)(298.15)}{(32)(0.036)} = 16670 \text{ atm} \]

The check is very satisfactory.

Table 1 gives Henry constant for helium and nitrogen at -100°C and -40°C. The values are hypothetical and extrapolated, hypothetical in that liquid is
freeze at a temperature of -98°C, and extrapolated in that Saaski's curves are extrapolated. The values are chosen merely to exemplify trends.

**TEST CASES**

Table 2 shows the test-case conditions selected. Table 3 shows the property values needed for the test cases.

**PREVIOUS STATE COMPOSITIONS**

As detailed in Part I the calculation commences with Eq.(I-6) and proceeds to Eq.(I-8).

\[ x_g = \frac{\left( \frac{y_2}{C_2} + \frac{y_3}{C_3} \right) P_{C_1}}{1 - \left( \frac{y_2}{C_2} + \frac{y_3}{C_3} \right) C_1} \]

\[ x_i = \frac{\left( \frac{y_1}{C_i} \right) P_{C_1}(1-x_g)}{1-x_g} \]

Table 4 gives the values.
TABLE 1

Extraction of Henry's Constants From A Solubility Curve

\[ C = \frac{1}{x_i} \text{ atm} \]

<table>
<thead>
<tr>
<th>Gas</th>
<th>Temperature</th>
<th>Solubility in Methanol</th>
<th>Henry's Constant C atm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Helium</td>
<td>-100°C</td>
<td>1 \times 10^{-5}</td>
<td>100000</td>
</tr>
<tr>
<td></td>
<td>(Hypothetical)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>-40°C</td>
<td>2.6 \times 10^{-5}</td>
<td>38500</td>
</tr>
<tr>
<td></td>
<td>(Extrapolated)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Nitrogen</td>
<td>-100°C</td>
<td>2.82 \times 10^{-4}</td>
<td>3550</td>
</tr>
<tr>
<td></td>
<td>(Hypothetical)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>-40°C</td>
<td>2.69 \times 10^{-4}</td>
<td>3720</td>
</tr>
<tr>
<td></td>
<td>(Extrapolated)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
## TABLE 2

### Hypothetical Test Cases

#### Test Case 1  Depressurization from Drop in Evaporator Temperature

**Previous State Variables**
- Evaporator Temperature, $T_e'$: 49°C (120°F)
- Gas Blocked Condenser Temperature, $T_c'$: -40°C (-40°F)
- Noncondensible Gas Composition, $y_{n,i}$: 90% N₂ - 10% He

**Present State Variables**
- Evaporator Temperature, $T_e$: 21°C (70°F)
- Gas Blocked Condenser Temperature, $T_c$: -40°C (-40°F)

#### Test Case 2  Chilldown in Gas-Blocked Condenser

**Previous State Variables**
- Evaporator Temperature, $T_e'$: 21°C (70°F)
- Gas Blocked Condenser Temperature, $T_c'$: -40°C (-40°F)
- Noncondensible Gas Composition, $y_{n,i}$: 90% N₂ - 10% He

**Present State Variables**
- Evaporator Temperature, $T_e$: 21°C (70°F)
- Gas Blocked Condenser Temperature, $T_c$: -100°C (-148°F)
<table>
<thead>
<tr>
<th>T (°C)</th>
<th>P_{\text{sat}} (\text{psia})</th>
<th>T (°C)</th>
<th>\sigma (\text{lb}_g/\text{ft})</th>
</tr>
</thead>
<tbody>
<tr>
<td>49</td>
<td>7.62</td>
<td>-40</td>
<td>2.1 \times 10^{-3}</td>
</tr>
<tr>
<td>21</td>
<td>1.96</td>
<td>-100</td>
<td>2.4 \times 10^{-3}</td>
</tr>
<tr>
<td>-40</td>
<td>0.02870</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-100</td>
<td>0.00012</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### Table 4

*Previous State Conditions*

<table>
<thead>
<tr>
<th>( \gamma_2 )</th>
<th>( \gamma_3 )</th>
<th>( p ) atm</th>
<th>( C_1 ) atm</th>
<th>( C_2 ) atm</th>
<th>( C_3 ) atm</th>
<th>( x_g )</th>
<th>( x_1 )</th>
<th>( x_2 )</th>
<th>( x_3 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>.90</td>
<td>.10</td>
<td>0.519</td>
<td>.001953</td>
<td>3720</td>
<td>38500</td>
<td>1.264 ( \times ) 10(^{-4} )</td>
<td>1.000</td>
<td>1.251 ( \times ) 10(^{-4} )</td>
<td>1.34 ( \times ) 10(^{-6} )</td>
</tr>
<tr>
<td>.90</td>
<td>.10</td>
<td>0.133</td>
<td>.001953</td>
<td>3720</td>
<td>38500</td>
<td>3.204 ( \times ) 10(^{-5} )</td>
<td>1.000</td>
<td>3.170 ( \times ) 10(^{-5} )</td>
<td>3.40 ( \times ) 10(^{-7} )</td>
</tr>
</tbody>
</table>

### Table 5

*Critical Bubble Size in an Infinite Liquid*

<table>
<thead>
<tr>
<th>( x_1 )</th>
<th>( x_2 )</th>
<th>( x_3 )</th>
<th>( C_1 ) atm</th>
<th>( C_2 ) atm</th>
<th>( C_3 ) atm</th>
<th>( \sigma ) ( \text{lb}_f/\text{ft}^2 )</th>
<th>( p ) atm</th>
<th>( p_l ) atm</th>
<th>( r_{cr} ) ( \mu \text{m} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.000</td>
<td>1.251 ( \times ) 10(^{-4} )</td>
<td>1.34 ( \times ) 10(^{-6} )</td>
<td>.001953</td>
<td>3720</td>
<td>38500</td>
<td>2.1 ( \times ) 10(^{-3} )</td>
<td>0.519</td>
<td>0.133</td>
<td>1.6 ( (0.6 \times 10^{-4} \text{ in.}) )</td>
</tr>
<tr>
<td>1.000</td>
<td>3.170 ( \times ) 10(^{-5} )</td>
<td>3.40 ( \times ) 10(^{-7} )</td>
<td>8.17 ( \times ) 10(^{-5} )</td>
<td>3550</td>
<td>10(^5 )</td>
<td>2.4 ( \times ) 10(^{-3} )</td>
<td>0.1466</td>
<td>0.133</td>
<td>50.8 ( (2 \times 10^{-3} \text{ in.}) )</td>
</tr>
</tbody>
</table>
Critical Bubble Size in an Infinite Liquid

As explained in Part I, to compute the critical bubble size, one computes

\[ P = \sum_{i=1}^{n} x_i' C_i(T_c) \]

and then

\[ r_{cr} = \frac{2\sigma(T_e)}{P - P_{sat}(T_e)} \]

Table 5 shown the results. For depressurization (case 1) the critical bubble is small, 1.6 \( \mu \)m in radius, the size of nuclei expected to be active in nucleate surface boiling. However, the gas composition near the surface exposed to the gas would be more nearly in equilibrium. That is contact with the wick might be close to the composition assumed. For chilldown (case 2) the increase in Henry's constant for helium results in supersaturation, but the potential for nucleation is much less. A fifty-micron-radius nucleus would be required for bubble formation.

Number of Bubbles of a Specified Size

From Part I the equation which governs the amount of gas \( N \) within bubbles is

\[ 1 = \frac{C_1 x_1'}{P + NC_1} + \frac{C_2 x_2'}{P + NC_2} + \frac{C_3 x_3'}{P + NC_3} \]

where the total pressure within the bubble is

\[ P = P_{sat}(T_e) + \frac{2\sigma}{r} \]

A-93
For case 1 and r = 50.8 μm the following values pertain:

\[
P = 1.96 + \frac{2(2.1 \times 10^{-3}/12}{50.8/25400} = 2.135 \text{ psia} = 0.1453 \text{ atm}
\]

\[
C_1 x_1' = (0.001963)(1.000) = 0.001953 \text{ atm}
\]

\[
C_2 x_2' = (3720)(1.251 \times 10^{-4}) = 0.465 \text{ atm}
\]

\[
C_3 x_3' = (38500)(1.34 \times 10^{-6}) = 0.052 \text{ atm}
\]

\[
N = 8.96 \times 10^{-5} \text{ moles of bubbles/mole of original liquid}
\]

The number of bubbles per mole is then

\[
N_b = \frac{N R T c}{P(\frac{4}{3} \pi r^3)} = \frac{(8.96 \times 10^{-5})(82.05)(233.15)}{(0.1453)(\frac{4}{3} \pi)(50.8 \times 10^{-4})^3}
\]

\[
N_b = 2.15 \times 10^7 \text{ Bubbles/g-mole of liquid}
\]

Taking \(c_L\) to be approximately \(0.8 \text{ [g/cm}^3\)/32[g/g-mole] gives

\[
N_b c_L = 537000 \text{ Bubbles/cm}^3 \text{ of liquid}
\]

**Conclusions**

The sample calculations show the possibility for bubble formation upon (1) depressurization by reduction in heat pipe loading and reservoir chilling and (2) condenser chilldown. Depressurization shows greater potential, for 537000 bubbles/cm\(^3\) are found possible, compared to only one of equal size from condenser chilldown.
APPENDIX A.5.3

BUBBLE NUCLEATION EXPERIMENTS
As part of the experimental program to investigate the potential of various mechanisms to cause artery depriming, a series of experiments have been conducted to examine bubble nucleation in methanol. The experiments were to determine whether gas bubbles could be generated in the bulk of liquid methanol saturated with either helium or nitrogen gas as it undergoes temperature and/or pressure reduction.

Prior to the experiments, a theoretical analysis of the potential for bubble formation in CTS heat pipes had shown that large numbers of bubbles could be generated in methanol due to supersaturation resulting from temperature and pressure reduction under conditions similar to those prevailing prior to the anomalies. An objective of these experiments was to verify, at least qualitatively, the theoretical results.

In addition the experiments considered the potential of a mesh screen artery to provide bubble nucleating sites.

The experimental set-up consisted of a glass flask half filled with 50cc of spectral grade methanol and instrumented to allow continuous monitoring of temperature and pressure. A sketch of the apparatus is attached. A needle valve located between the flask and a vacuum pump was used to control the pressure level or the rate of pressure reduction of the liquid. Cooling of the liquid was attained by immersion of the test flask in liquid nitrogen. This technique, however, did not allow for arbitrary control of the cooling rate. The liquid was saturated by bubbling either nitrogen or helium gas through the liquid using a frit glass tube. This process was allowed to be continued for at least two hours.

A typical pressure reduction experimental sequence was as follows:

1. Saturate methanol with either nitrogen or helium at ambient conditions.
2. Reduce temperature at atmospheric pressure. Two temperature levels, 21°C and -40°C, were used.
3. Drop into the liquid a dry section of mesh screen artery.
4. Reduce pressure from 14.7 psia to 4 psia. In some cases this pressure drop was accomplished in 10 seconds and in others in 5 minutes.

A typical temperature reduction experimental sequence was as follows:

1. Saturate methanol at ambient conditions.
2. Drop isoto the liquid a dry section of mesh screen artery.
3. Reduce pressure at ambient temperature. Pressure levels of 14.7 psia, 10 psia, and 4 psia were used.
4. Reduce temperature. In some cases the temperature was reduced from 21°C to -40°C in approximately 20 minutes. This was accomplished by placing the flask near the liquid nitrogen surface contained in a Dewar flask where cooling of the methanol occurred by natural convection. In other cases, the liquid was rapidly chilled by immersing the flask in liquid nitrogen for approximately 10 seconds at which point the liquid on the bottom of the test flask and inside the artery froze.

The experiments were repeated several times and the results were found reproducible. No bubbles were observed as the liquid, originally at a set temperature level, underwent pressure reduction. Similar results were obtained from temperature reduction experiments provided the liquid temperature did not drop below the freezing point (-98°C).

Experiments in which the liquid partially froze however, yielded significantly different results. A large number of small bubbles were observed streaming from the surface of the thawing ice. As the melting process went to completion the bubbles were reduced to a few originating from the bottom surface of the flask and from the outer surface of the mesh screen artery. About a dozen small bubbles were observed trapped inside the arteries after the ice melted. These bubbles were observed later to coalesce forming fewer but larger bubbles which continue to grow. The ultimate size of these bubbles depended on the pressure of the system. For example, at 4 psia the remaining bubbles inside the artery continue to grow for approximately 10 minutes as the liquid temperature rose from -40°C to 0°C, at which point the size of the bubbles was such that essentially all the liquid in the artery was displaced.

As the result of these experiments, freezing of supersaturated methanol in the arteries has been identified as a potential mechanism for bubble formation in the arteries owing to the fact that the ice surface is an excellent source of nucleating sites.
TEMPERATURE RECORDER

VACUUM PUMP

LIQUID TRAP

NEEDLE VALVE

PRESSURE TRANSDUCER

MILLIVOLT RECORDER

He OR N₂ GAS

GLASS FRIT TUBE

METHANOL

MESH SCREEN ARTERY SECTION

LN₂
APPENDIX A.5.4
GLASS HEAT PIPE BUBBLE NUCLEATION/MIGRATION EXPERIMENTS

A series of experiments were performed with an existing 1.07 meter long glass heat pipe. A cross section of this heat pipe is shown in Figure 1. The pipe contains a slab wick with a CTS-type artery attached to one side. A heater and cooling loop are attached to the other side of the wick in the evaporator and condenser sections, respectively. This arrangement permits observations on the behavior of the artery in an operational heat pipe.

The heat pipe was gas loaded with a 90% nitrogen – 10% helium mixture at a pressure equivalent to the conditions in the CTS pipes. The experiments simply involved visually observing the artery behavior as a result of freezing the methanol within it by passing liquid nitrogen through the cooling loop, and subsequently thawing the methanol by terminating the LN$_2$ flow.

The experiment was repeated several times with the following results:

- The methanol froze to an opaque white solid (frost), indicating it was full of gas bubbles.
- As the methanol thawed and warmed, numerous (up to 33) gas bubbles of varying size were observed along a 12-inch length of the artery.
- The bubbles slowly shrank and ultimately disappeared as the gas within them redissolved into the liquid and diffused through the artery wall into the surrounding vapor core under the pressure gradient established by their curvature and surface tension.
- The time required for the bubbles to disappear varied enormously, depending on their size. Very small bubbles (d ~1/4 artery dia) dissolved within a few hours, while a large sausage-shaped bubble which filled the artery (2 artery diameters long) required several weeks.

It was believed that incorporation of these ice-generated bubbles into the active condenser could cause artery derimming. Accordingly, additional tests were conducted and a number of deprimings were observed, but they were sporadic (probabilistic in nature). On some occasions, as the active

A-99
Figure 1. Cross Section of the Glass Heat Pipe in the Condenser and Evaporator Regions

- GLASS TUBE {1.9 cm O.D., 1.3 cm I.D.}
- SHEATHED THERMOCOUPLE
- DOUBLE LAYER SCREEN CASING
- ROD HEATER
- 0.127 cm THICK SLAB WICK
- COOLANT LOOP
- ARTERY

CONDENSER CROSS SECTION
EVAPORATOR CROSS SECTION

A-100
condensation front was brought into the previously gas-blocked condenser, no depriming occurred. On two occasions, freezing generated a bubble in the adiabatic section; on both these occasions, advancement of the front to the bubble caused depriming. On another occasion, a rather large bubble formed in the center of the condenser, and advancement of the front to this bubble caused depriming.

These results conclusively demonstrated that: 1) control gas is liberated from saturated methanol every time the heat pipe goes through a freeze-thaw cycle, and 2) the number, size, and durability of gas bubbles generated within the arteries is a statistical phenomena influenced by bubble coalescence. Arterial bubbles are known to lead to depriming if they are convected into the active region of the pipe under high load conditions. Thus, it is clear that freezing and thawing of the condenser can, but does not necessarily, lead to artery depriming, depending on subsequent history. That is, do the bubbles dissolve and diffuse away before they are convected into a high stress region where they would deprime the artery?

This mechanism appears to be a prime candidate for explaining the CTS anomalies. It is consistent with their seasonal occurrence (eclipse seasons may be necessary to experience condenser freezing), sporadic occurrence (statistical nature of bubble population) and the lack of freezing prior to the anomaly on day 253 (bubbles were generated during day 252).
APPENDIX B.1

SPHERICAL BUBBLE MODEL COMPUTER PROGRAM LISTING

This appendix contains the main program and sample input.
PROGRAM SMLUB (INPUT, OUTPUT, TAPE5=INPUT, TAPE6=OUTPUT, TAPE9)

--- FILE: SMLUB INPUT FILE: SUBDAT ---

DIMENSION A(100), B(100), C(100), ASTR(100), BSTR(100), A2(3),
1B2(3), C2(3), XBBC(3), XEBC(3), FG1(100), FG2(100),
2FG1(100), FG2(100), DIFF(3)

COMMON/MAIN/X(100,3), XSTR(100,3), XSTR2(100,3), R(100), Y(100),
1Z(100), R1(100), R2(100), TIME, TC, TE, PVC, PVE, WG1, WG2, YE1, YE2, PB, TCN0T,
2XB1, XB2, XE1, XE2, YB1, YB2, IPRINT, KPL0T, YBO, IPLOT, IDPLOT, TENOT, RATE

COMMON/SUBDAT/CMDLE, RHOL, RHQV, RU, EMW

COMMON/DATPLO/FTIME(1000), FTE(1000), FTC(1000), FPVE(1000),
1FPVC(1000), FYBO(1000), FYB1(1000), FYB2(1000),
2FR1(1000), FPBUBLE(1000)

COMMON/SC0DIF/DF11, DF12, DF12, DF22, COS11, COS12, COS21, COS22,
1NG1, NG2

DATA PI, RU/3.1416, 82.06/

NAMESLIST/SUBDAT/R10, R20, NDIV, GAMA, DT, IPRINT, RFLILLET, FYB1, FYB2,
1FEY1, FEY2, EMW, TCN0T, TENOT, NSTPMX, IDPLOT, RATE,
2DF11, DF12, DF21, DF22, COS11, COS12, COS21, COS22, NG1, NG2

INPUT DATA

READ(5, BUBDAT)
WRITE(6, BUBDAT)

CALCULATE CONSTANT COEFFICIENTS

TIME=-1.0

CALL TPLCNDI
CALL SUBRHQ(TC, RHOL, RHQV)
CMDLE=RHOL/EMW
PI(3)=.4/.3*PI
VLIQ2*R2C**3-R10**3
VLIQ1=VLIQ2*PI(3)
N=NDIV+2
DZ=1./FLOAT(NDIV)
DZ2=DZ*DZ
DZD2=DZ2*DZ
BIGGAM=EXP(GAMA)-1.0

INITIALIZE SPECIES AND BUBBLE RADIUS

CALL TENSION(TC, SIGMA)
PLIC=PVE-PVE. SIGMA/RFILLET
YEO=PVC/PVE
YE1=(1.-YEO)*FEY1
YE2=(1.-YEO)*FEY2
CALL OSWALT(TC,HENRY1,HENRY2)
X1NOT=YE1*PVE/HENRY1
X2NOT=YE2*PVE/HENRY2
X1PX2=X1NOT*X1PX2
XBY1=X1NOT/X1PX2
XBY2=X2NOT/X1PX2
CO 100  I=2,N
X(I,1)=X1NOT
X(I,2)=X2NOT
XSTR(I,1)=X1NOT
XSTR(I,2)=X2NOT
XSTR2(I,1)=X1NOT
XSTR2(I,2)=X2NOT
100 CONTINUE
R1=R10
R1STR=R1C
R1STR2=R10
DR1DT=0.
DOR1T=0.
R1MIN=0.1*R1
INITIALIZE TIME AND FLAGS
TIME=0.
DTU=0.
IPRINT=0
IPLT=0
KPLT=0
NSTEP=0
CALCULATE INITIAL AMOUNT OF GAS IN BUBBLE
CALL TPCCNDI
CALL TENSION(TC,SIGMA)
PLIO=PVE=PVE. SIGMA/RFILLET
PG=PLIO+PVE. SIGMA/R1-PVC
PB=PG+PVC
WGT=PG*PI43*R1**3/RU/TC
00940  WG1=WGT*FBY1
00950  WG2=WGT*FBY2
00960  C
00970  C  CALCULATE CONSTANT GEOMETRIC FUNCTIONALS
00980  C
00990    Z(I)=-DZ
01000    DO 200  I=2,N
01010    Z(I)=Z(I-1)+DZ
01020   FG1(I)=BIGGAM*Z(I)+1.0
01030   FG2(I)=ALOG(FG1(I))
01040   Y(I)=FG2(I)/GAMA
01050 200  CONTINUE
01060  C
01070  C  ENTER TIME LOOP
01080  C
01090  CALL PRINT
01100  9999 CONTINUE
01110  TIME=TIME+DT
01120  C
01130  C  SET COEFFICIENTS FOR TEMPORAL DERIVATIVES APPROXIMATION
01140  C
01150  IF(DTU.NE.DT) GO TO 300
01160  COT1=1.5
01170  COT2=-2.
01180  COT3=0.5
01190  GO TO 400
01200  300 CONTINUE
01210  COT1=1.0
01220  COT2=-1.0
01230  COT3=0.
01240  400 CONTINUE
01250  C  DTU=DT
01260  IPRINT=IPRINT+1
01270  C
01280  C  CALCULATE EXTERNAL PRESSURE, TEMPERATURE AND GAS CONCENTRATION
01290  C
01300  CALL TPCONDI
01310  CALL SUBRHO(TC,RHOL,RHOV)
01320  CALL TENSION(TC,SIGMA)
01330  P=SIGMA/RFILLET
01340  YL=VC/PVE
01350  YE=.-YEO)*FEY1
01360 YE2=(1.-YE0)*FEY2
01370 CALL Oswalt(TC,HENRY1,HENRY2)
01360 XE1=YE1*PVE/HENRY1
01390 XE2=YE2*PVE/HENRY2
01400 C
01410 C SET B.C.'S AT EXTERNAL INTERFACE
01420 C
01430 XEBC(1)=XE1
01440 XEBC(2)=XE2
01450 C
01460 C CALCULATE B.C.'S AT BUBBLE INTERFACE
01470 C
01480 WG1=WG1+DG1DT*DT
01490 WG2=WG2+DG2DT*DT
01500 WG12=WG1+WG2
01510 C
01520 C CALCULATE NEW BUBBLE RADIUS
01530 C SOLVE A*X**3+B*X+C=0 BY QUASILINEARIZATION
01540 C
01550 CAB1=RU*TC*(WG1+WG2)/PI43
01560 CAB2=-2.*SIGMA
01570 CAB3=-[PIQ-PVC]
01580 ZR=1./R1
01590 DO 500 ITER=1,50
01600 ZRU=ZR
01610 ZR2=ZR*ZR
01620 ZR3=ZR2*ZR
01630 FACTOR=CAB3+CAB2*ZR+CAB1*ZR3
01640 FACTOR=FACTOR/(3.*CAB1*ZR2+CAB2)
01650 ZR=ZR+FACTOR
01660 ERROR=ABS(1.0-ZR/ZRU)
01670 IF(ERROR.LT.1.E-4) GO TO 501
01680 500 CONTINUE
01690 501 R1=1./ZR
01700 IF(R1.LE.0.) GO TO 51
01710 VDL=PI43*R1**3
01720 PG1=WG1*RU*TC/VOL
01730 PG2=WG2*RU*TC/VOL
01740 PB=PVC+PG1+PG2
01750 YB0=PVC/PB
01760 YB1=PG1/PB
01770 YB2=PG2/PB
01780       XB1=PG1/HENRY1
01790       XB2=PG2/HENRY2
01800 C
01810 C SET B.C. S AT BUBBLE INTERFACE
01820 C
01830       XBBC(1)=XB1
01840       XBBC(2)=XB2
01850 C
01860 C CALCULATE BUBBLE RADIUS TEMPORAL DERIVATIVES
01870 C
01880       DR1DT=(CCT1*Q1+CDT2*R1STR+COT3*R1STR2)/DT
01890       R1SQ=R1*R1
01900       R1CU=R1SQ*R1
01910       R2=(VLIQ2+R1CU)**(1./3.)
01920       R2SQ=R2*R2
01930       DR2DT=R1SQ/R2SQ*DR1DT
01940       DR=R2-R1
01950       DDRDT=DR2DT-DR1DT
01960       COF1=GAMA/DR/BIGGAM
01970       COF2=DR/GAMA
01980       COF2P=DDRDT/GAMA
01990       COF4=DR1DT*R1**2*COF1
02000 C
02010 C CALCULATE COEFFICIENTS FOR PARTIAL DIFFERENTIAL
02020 C EQUATIONS
02030 C
02040       DO 600 K=NG1,NG2
02050       CALL DIFFCO(K,TC,DIFF(K))
02060       COF3=2.*DIFF(K)*COF1
02070       COF5=-DIFF(K)*GAMA/DR*COF1
02080       COF6=-DIFF(K)*COF1*COF1
02090       CD 700 I=2,N
02100       FGG1(I)=COF1*FG1(I)*(COF2P*FG2(I)+DR1DT)
02110       1 +COF3*FG1(I)*1.0/(COF2*FG2(I)+R1)
02120       2 -COF4*FG1(I)*1.0/((COF2*FG2(I)+R1)**2)
02130       3 -COF5*FG1(I)
02140       FGG2(I)=-COF6*FG1(I)*FG1(I)
02150       R(I)=DR*Y(I)+R1
02160       700 CONTINUE
02170 C
02180 C
02190 C RESET FSTR#S AND FSRT2#S
02200 C
02210 DO 800 I=2,N
02220 XSTR2(I,K)=XSTR(I,K)
02230 XSTR(I,K)=X(I,K)
02240 800 CONTINUE
02250 C
02260 C CALCULATE COEFFICIENTS A(I),B(I),C(I) FOR KTH ALGEBRAIC
02270 C SPECIES EQUATION. X(I)=A(I)*X(I+1)+B(I)*X(I-1)+C(I)
02280 C
02290 DO 900 I=2,N
02300 DENOM=CDT1/DT+2.0*FGG2(I)/DZDZ
02310 A(I)=(FGG1(I)/DZ2+FGG2(I)/DZDZ)/DENOM
02320 B(I)=(FGG2(I)/DZDZ-FGG1(I)/DZ2)/DENOM
02330 C(I)=(-CCT2*XSTR(I,K)/DT+CDT3*XSTR2(I,K)/DT)/DENOM
02340 900 CONTINUE
02350 C
02360 C AT NODE =2=
02370 C 1) SAVE COEFFICIENTS A(2),B(2) AND C(2)
02380 C 2) RESET THE ABOVE
02390 C
02400 A22(K)=A(2)
02410 B22(K)=B(2)
02420 C22(K)=C(2)
02430 A(2)=.0.
02440 B(2)=.0.
02450 C(2)=XBBC(K)
02460 C
02470 C AT NODE =N=
02480 C RESET COEFFICIENTS A(N),B(N),AND C(N)
02490 C
02500 A(N)=.0.
02510 B(N)=.0.
02520 C(N)=XBBC(K)
02530 C
02540 C CALCULATE COEFFICIENTS ASTR(I) AND BSTR(I) FOR
02550 C X(I)=ASTR(I)*X(I+1)+BSTR(I)
02560 C
02570 C AT NODE =2=
02580 C
02590 ASTR(2)=.0.
02600 BSTR(2)=C(2)
02610 C
GO TO 9999
CALL SETPLOT
CALL TRACE(FTIME,FTE,KPLOT,1.0)
CALL TRACE(FTIME,FTC,KPLOT,2.0)
CALL ENDPLOT(9HTE AND TC,9HTIME(SEC),7HTEMP(K),5HPARAM)
CALL TRACE(FTIME,FPVE,KPLOT,3.0)
CALL TRACE(FTIME,FPVC,KPLOT,4.0)
CALL TRACE(FTIME,FPBUBLE,KPLOT,5.0)
CALL ENDPLOT(10HPE,PC,PBUB,9HTIME(SEC),9HPRES(ATM),5HPARAM)
CALL TRACE(FTIME,FR1,KPLOT,6.0)
CALL ENDPLOT(10HBUBLE SIZE,9HTIME(SEC),10HRADIUS(CM),
15HPARAM)
CALL TRACE(FTIME,FYBO,KPLOT,7.0)
CALL TRACE(FTIME,FYB1,KPLOT,8.0)
CALL TRACE(FTIME,FYB2,KPLOT,9.0)
CALL ENDPLOT(8HXD,X1,X2,9HTIME(SEC),10HMOLE FRCTN,5HPARAM)
CALL ENALL
CALL PRINT
STOP
END
SUBROUTINE SETPLOT
SAVE VARIABLES FOR PLOTTING
COMMON MAIN/X(100,3),XSTR(100,3),XSTR2(100,3),R(100),Y(100),
IZ(100),R1,R2,NTIME,TC,TE,PC,PE,WE,WG1,WG2,Y1,Y2,PB,TCNT,
2XB1,XB2,XE1,XE2,YB1,YB2,IPRINT,KPLOT,YBO,IPLOT,IDPLOT,TEND,T,RATE
COMMON DATPLO/FTIME:1000,FTC:1000,FPVE:1000,
1FPVC:1000,FYBO:1000,FYB1:1000,FYB2:1000,
2FR1:1000,FPBUBLE:1000
IPlot=IPlot+IdPlot
KPlot=KPlot+1
FTIME(KPLOT)=TIME
FTE(KPLOT)=TE
FTC(KPLOT)=TC
FPVE(KPLOT)=PV
FPVC(KPLOT)=PV
FYBO(KPLOT)=YBO
FYB1(KPLOT)=YB1
FYB2(KPLOT)=YB2
FR1(KPLOT)=R1
FPBUBLE(KPLOT)=PB
03460 100  RETURN
03470   END
03480  SUBROUTINE OSWALT(XT,CH1,CH2)
03490  C
03500  C  SUBROUTINE TO CALCULATE GAS SOLUBILITY IN A LIQUID
03510  C
03520  COMMON/SUBDAT/CMOLE,RHOL,RHOM,RU,EMW
03530  COMMON/SCLDIF/D11,D12,D12,D22,COS11,COS12,COS21,COS22,
03540  1NG1,NG2
03550  C
03560  C  COEFFICIENTS FOR SOLUBILITY EQUATION
03570  C  F(T)=COS1*T*COS2 IN 1./ATM
03580  C  WHERE T=TEMPERATURE IN DEGREES KELVIN
03590  C
03600  CH1=COS11*XT*COS21
03610  CH1=1./CH1
03620  CH2=COS12*XT*COS22
03630  CH2=1./CH2
03640  100  RETURN
03650   END
03660  SUBROUTINE TPCONDI
03670  C
03680  C  UPDATE EXTERNAL PRESSURE AND TEMPERATURE
03690  C
03700  COMMON/MAIN/X(100,3),XSTR(100,3),XSTR2(100,3),X(100),Y(100),
03710  1Z(100),R1,R2,N,TIME,TC,TE,PVC,PVE,WG1,WG2,YE1,YE2,PB,TCN0T,
03720  2XB1,XB2,XE1,XE2,YB1,YB2,IPRINT,KPLOT,YBO,IPLOT,IDPLOT,TENGT,RATE
03730  TC=180.+RATE*TIME
03740  TE=300.
03750  IF(TIME.GT.0.)AND.TC.GE.TCN0T) TC=TCN0T
03760  IF(TIME.LE.0.) GO TO 50
03770  TC=TCN0T
03780  TE=TENGT
03790  50  CONTINUE
03800  C
03810  C  COEFFICIENTS FOR METHANOL VAPOR PRESSURE
03820  C
03830  DATA CP1,CP2,CP3,CP4,CP5
03840  1 /15.05411,-9.24063,3.366136,-1.9692,3.389658E-1/
03850  TRC=1000./1.8/TC
03860  TRC2=TRC2*TRC
03870  TRC3=TRC2*TRC
03880 TRC4=TRC3*TRC
03890 PVC=CP1+CP2*TRC+CP3*TRC2+CP4*TRC3+CP5*TRC4
03900 PVC=EXP(PVC)/14.696
03910 TRE=1000./1.8/TE
03920 TRE2=TRE*TRE
03930 TRE3=TRE2*TRE
03940 TRE4=TRE3*TRE
03950 PVE=CP1+CP2*TRE+CP3*TRE2+CP4*TRE3+CP5*TRE4
03960 PVE=EXP(PVE)/14.696
03970 100 RETURN
03980 END
02990 SUBROUTINE DIFFCO(ISPICE,TEMP,DIFCO)
04000 COMMON/SOLDIF/DF11,DF21,DF12,DF22,COS11,COS12,COS21,COS22,
04010 NG1,NG2
04020 C COEFFICIENTS FOR DIFFUSIVITY EQUATION OF GASES IN METHANOL
04030 C D(T)=DF1*EXP(DF2/T) IN SCCM/SEC
04040 C WHERE T=TEMPERATURE IN DEGREES KELVIN
04050 IF(ISPICE,GT,1) GO TO 10
04060 DIFCO=DF11*EXP(DF21/TEMP)
04070 GO TO 10C
04080 10 DIFCO=DF12*EXP(DF22/TEMP)
04090 100 RETURN
04100 END
04110 SUBROUTINE TENSION(TEMP,FTEMP)
04120 C COEFFICIENT FOR SURFACE TENSION EQN FOR METHANOL
04130 C DATA CT1,CT2,CT3,CT4,CT5
04140 1 5.79025E-3,-1.404494E-5,1.20562E-8,3.516629E-12,
04150 2 -8.67029E-15/
04160 TR=TEMP*1.8
04170 TR2=TR*TR
04180 TR3=TR2*TR
04190 TR4=TR3*TR
04200 FTEMP=CT1+CT2*TR+CT3*TR2+CT4*TR3+CT5*TR4
04210 FTEMP=FTEMP*1.44E-2
04220 100 RETURN
04230 END
04240 SUBROUTINE PRINT
04250 COMMON/MAIN/X(100),X3,XSTR(100),XSTR2(160),R(100),Y(100),
04260 1Z(100),R1,R2,N,TIME,TC,TE,PVC,PVE,W1,W2,Y1,Y2,PB,TCNOT,
04270 2XB1,XB2,XE1,XE2,YB1,YB2,IPRINT,KPLOT,YBO,IPLOT,IDPLOT,TCNOT,RATE
YV = CRV1 + CRV2/TR + CRV3/TR2 + CRV4/TR3 + CRV5/TR4
04730 YV = 1.6E-2 * EXP(YV)
04740 100 RETURN
04750 END
04760 SUBROUTINE TRACE(V, F, NV, PVAL)
04770 C SAVE 1 TRACE (SET OF (V(I), F(I))) OF DATA ON TAPE9.
04780 C ALSO COLLECT EXTREMA INFO
04790 COMMON /PLOT/C, TITLE, VNAME, FNAME, PNAME, VMIN, FMIN, PMIN
04800 1, VMAX, FMAX, PMAX, NP, NPLTS, LOCHED(63), PA(256), NA(256)
04810 DIMENSION V(1), F(1), BLK(1026), INDX(1080)
04820 DATA VMIN, FMIN, PMIN, VMAX, FMAX, PMAX /3*1.E30, 3*-1.E30/
04830 1, NP, NPLTS, LOCHED(1) /0, 1, 1/
04840 2, IPASS1/1/
04850 C SKIP IF NOT 1ST PASS
04860 C IF (IPASS1 .EQ. 0) GO TO 10
04870 IF (IPASS1 = 0)
04880 IPASS1 = 0
04890 C DON'T WANT TO ADD RECORDS TO AN EXISTING FILE. RETURN OLD
04900 C CALL RETFILE(9)
04910 CALL OPENMS(9, INDX, 1080, 0)
04920 C AFTER INC, LOCHED(NPLTS) = RECCRD NUMBER TO STORE THIS TRACE
04930 C 10 LOCHED(NPLTS) = LOCHED(NPLTS) + 1
04940 IF (LOCHED(NPLTS) .GT. 1077) X = SQRT(-1.)
04950 C NUMBER OF PARAMETER VALUES (TRACES) IN THIS PLOT
04960 C NP = NP + 1
04970 IF (NP .GE. 256) X = ACOS(2.)
04980 C SAVE RECCRD SIZE(/2)
04990 NA(NP) = NV
05000 IF (NV .GT. 513) X = ASIN(2.)
05010 C SAVE PARAMETER VALUE
05020 C FIND EXTREMA OF PARAMETER, VARIABLE, FUNCTION
05030 C PMIN = AMIN1(PMIN, PVAL)
05040 C PMAX = AMAX1(PMAX, PVAL)
05050 C DO 20 I = 1, NV
05060 X = V(I)
05070 DO 20 I = 1, NV
05080 X = V(I)
05090 VMIN = AMIN1(VMIN, X)
05100 VMAX = AMAX1(VMAX, X)
05110 C PACK VARIABLE, THEN FUNCTION PRIOR TO WRITE
05120 BLK (I) = X
05130 Y = F(I)
05140  FMN = AMNX(FMN, Y)
05150  FMAX = AMX1(FMAX, Y)
05160  20  BLK(I+NV) = Y
05170  C  WRITE (V(I),I=1,NV),(F(I),I=1,NV)
05180  CALL WRITMS(9, BLK, 2*NV, LOCHED(NPLTS))
05190  RETURN
05200  END
05210  SUBROUTINE ENDPLT(TI, VN, FN, PN)
05220  C  TERMINATE CURRENT PLOT, SET UP FOR NEXT
05230  COMMON /PLOTC/TITLE,VNAME,FNAME,PNAM3,VMIN,FMIN,PMIN
05240  1, VMAX,FMAX,PMAX,NP,NPLTS,LOCHED(63),PA(256),NA(256)
05250  EQUVALENCE (IPA,PA)
05260  DIMENSION IPA(1)
05270  C
05280  C  MOVE CALLING SEG INTO STORAGE BLOCK
05290  TITLE = TI
05300  VNAME = VN
05310  FNAME = FN
05320  PNAME = PN
05330  C  NEXT IS FINAL SETTING OF LOCHED(NPLTS).  NOTE LOCHED(I) IS
05340  C  RECORD NUMBER OF HEADER FOR PLOT I
05350  LOCHED(NPLTS) = LOCHED(NPLTS) + 1
05360  C  WRITE HEADER REC--INCLUDES VARIABLES IN /PLOTC/ FROM TITLE TC NP
05370  CALL WRITMS(9, TITLE, 11, LOCHED(NPLTS))
05380  C  PACK LIST OF RECORD LEN/2 RIGHT AFTER LIST OF PARAMETER VALUES
05390  C  NOTE EQU (IPA,PA)
05400  DO 10 I=1,NP
05410     IPA(I+NP) = NA(I)
05420  C  WRITE PARAMETER, LENGTH LISTS AFTER HEADER
05430  CALL WRITMS(9, PA, 2*NP, LOCHED(NPLTS)+1)
05440  C  NUMBER OF NEXT PLOT
05450  NPLTS = NPLTS+1
05460  IF (NPLTS GT 63) X = ALOG(-1.)
05470  C  (CHECK TRACE) NOTE TRACES FOR NEXT PLOT START RIGHT AFTER
05480  C  HEADER, LIST RECORDS FOR THIS ONE
05490  LOCHED(NPLTS) = LOCHED(NPLTS-1) + 1
05500  C  INITIALIZE SCALING, NUMBER OF PARAM VALUES FOR NEXT PLOT
05510  BIG = 1.E30
05520  VMIN = BIG
05530  FMIN = BIG
05540  PMIN = BIG
05550  VMAX = -BIG
05560 FMAX = -BIG
05570 PMAX = -BIG
05580 NP = 0
05590 RETURN
05600 END
05610 SUBROUTINE ENDAALL
05620 C WRAP UP PLOT DATA FILE
05630 COMMON /PLOTC/ TITLE,VNAME,FNAME,PNAME,VMIN,FMIN,PMIN
05640 1, VMAX,FMAX,PMAX,NP,NPLTS,LOCHED(63),PA(256),NA(256)
05650 C NPLTS = NPLTS-1
05660 C WRITE NUMBER OF PLOTS ON FILE AND REC NUMBER OF HEADER FOR EACH
05670 CALL WRITHS(9, NPLTS, 64, 1)
05680 CALL CLOSEMS(9)
05690 RETURN
05700 END

B-15
Sample Input

-BUBDAT
COS11=2.5091E-12
COS21=2.9665
COS12=2.6367E-4
COS22=0.0
DF11=5.92E-3
DF12=2.94E-2
DF12=2.09E-3
DF22=-1070.5
NG1=1
NG2=2
NSTPMX=8000
IDPLOT=30
R10=0.02
R20=0.08
NDIV=90
GAMA=-2.0
DT=5.0
IDPRNT=200
RFILLET=10.
FHY1=1.0
FHY2=0.
FEY1=0.1
FEY2=0.9
EMW=32.0
TCNOT=250.
TENOT=300.
RATE=0.0
$END
APPENDIX B.2

SNO09 HEAT PIPE CYCLIC TEST DATA

This appendix presents plots of temperatures along the SNO09 heat pipe versus time for selected freeze/thaw test cycles not shown in Section 3.3.
Figure B.2-2 Temperature Histories During ST009 Heat Pipe Cyclic Test No. 5

B-19
Figure B.2.11 Temperature Histories During SN009 Heat Pipe Cyclic Test No. 25