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**HYDROGEN PERMEATION RESISTANT HEAT PIPE
FOR BI-MODAL REACTORS**

FINAL REPORT

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1.0 EXECUTIVE SUMMARY

The principal objective of this program was to demonstrate technology that will make a sodium heat pipe tolerant of hydrogen permeation for a bimodal space reactor application. Special focus was placed on techniques which enhance the permeation of hydrogen out of the heat pipe. Specific objectives include:

1. Define the detailed requirements for the bimodal reactor application.
2. Design and fabricate a prototype heat pipe tolerant of hydrogen permeation.
3. Test the prototype heat pipe and demonstrate that hydrogen which permeates into the heat pipe is removed or reduced to acceptable levels.

The results of the program were fully successful. Analyses were performed on two different heat pipe designs and an experimental heat pipe was fabricated and tested. The experimental heat pipe was constructed of nickel with sodium as the working fluid. The heat pipe was operated between 720°C and 800°C. While operating in a steady state, hydrogen gas was injected into the heat pipe. The response of the heat pipe was monitored while the hydrogen permeated out of the heat pipe in the condenser section. A specially designed condenser which included a re-entrant annular gas cavity and an array of small diameter, thin-walled tubes enhanced the rate at which hydrogen permeated out of the heat pipe. For each of the tests run, the hydrogen gas was removed from the heat pipe in approximately 5 to 10 minutes.

A model of the experimental heat pipe was developed to predict the enhancement in the hydrogen permeation rate out of the heat pipe. A significant improvement in the rate at which hydrogen permeates out of a heat pipe was predicted for the use of the special condenser geometry developed here. Agreement between the model and the experimental results was qualitatively good. Inclusion of the additional effects of fluid flow in the heat pipe are recommended for future work.

Based on the successful results of this program, it is recommended that future experimental work be performed using the high temperature materials proposed for NEBA-1. Testing of such a heat pipe could be done at NEBA-1 design conditions.

2.0 INTRODUCTION

Bimodal space nuclear systems which combine both thermal propulsion and electric generation capabilities in a single integrated power and propulsion platform have recently emerged as candidates for future military and commercial space applications. For propulsion, the reactor heats hydrogen which is exhausted through a nozzle to provide propulsive thrust. Electrical energy is also generated by either an in-core or out-of-core thermal-to-electric conversion system.

Hydrogen permeation presents potential problems in these bimodal systems. For example, the proposed design of the NEBA-1 bimodal reactor uses a series of high temperature sodium/molybdenum heat pipes to transfer heat from the reactor to an array of ex-core thermoelectric converters. The design involves flowing high pressure hydrogen gas over the outside of the evaporator of a high temperature molybdenum/sodium heat pipe during propulsion, with no hydrogen between burns. The large difference in hydrogen partial pressure across the evaporator wall combined with the high temperature of the evaporator wall makes it possible for a significant flow of hydrogen to occur through the evaporator wall. If unchecked, this hydrogen would rapidly fill the condenser section of the heat pipes, preventing electrical energy generation by the thermoelectric converters.

What is needed in this system is a heat pipe which is capable of venting any hydrogen which permeates into it. This venting process must proceed quickly enough so that the heat pipe is still capable of transferring heat when required. This will vary with different heat pipe applications and mission profiles. For instance, if hydrogen were to continuously permeate into the heat pipe, the venting process would be required to remove hydrogen at the same rate to prevent accumulation of noncondensable gas. The bimodal reactor heat pipe, on the other hand, has a cyclic hydrogen loading in which hydrogen permeates into the heat pipe rapidly for a short period of time which is followed by a long period in which there is no permeation into the heat pipe. In this application, the rate at which hydrogen is vented from the heat pipe needs to be high enough that there is no net accumulation of gas over many cycles and that the heat pipe is capable of transferring some minimum amount of heat at the end of the permeation phase of the cycle. This program aims to analytically and experimentally investigate such a venting scheme, with the bimodal reactor as a baseline application.

Hydrogen Permeation

To understand how hydrogen permeation is a problem in systems which operate with both high temperature heat pipes and hydrogen, and to devise a solution to the permeation problems encountered, it is useful to understand what factors influence hydrogen permeation. This section gives a brief overview of this process.

Hydrogen permeation through a metal is the overall process of moving hydrogen from the bulk gas phase on the high pressure side of the metal to the bulk gas phase on the low pressure side. The process consists of interactions at both the entry and exit surfaces of the metal, as well as the actual transport through the metal. Hydrogen enters and diffuses through the metals as monatomic atoms; the hydrogen molecule (H_2) does not diffuse in metals. The hydrogen diffuses through the metal as atoms and recombines into molecules at the exit surface [Kohl, 1960].

The process of hydrogen permeation through metals has been well studied [Bhattacharyya, et al., 1981, Fraunfelder, 1967, Svedburg, 1980]. When diffusion through the metal is the limiting process, the hydrogen permeation rate is given by the following:

$$j_t = \Phi \frac{A}{\Delta x} (P_{\text{source}}^{\frac{1}{2}} - P_{\text{sink}}^{\frac{1}{2}}) \quad (1)$$

where

j_t = volumetric hydrogen flow normal to the surface at STP of 1 torr and 0°C, liter/s

Φ = permeability, liter/cm·s·torr^{1/2}

A = surface area for hydrogen flow, cm²

Δx = material thickness, cm

P_{source} = hydrogen source pressure (or partial pressure), torr

P_{sink} = hydrogen sink pressure (or partial pressure), torr.

Examining equation (1), the permeation rate depends on three factors: a) the hydrogen partial pressure difference, which provides the driving potential for the hydrogen flow, b) the geometry, with permeation varying linearly with surface area and inversely with solid thickness, and c) a proportionality constant, the permeability, which depends only on the solid material. For a given material, the permeability is an exponential function of temperature. For nickel, the permeability is [Fraunfelder, 1967]:

$$\Phi = 6.610^{-4} e^{-\frac{54.81 \text{ kJ/kmol}}{RT}} \frac{\text{liter}}{\text{cm} \cdot \text{s} \cdot \text{torr}^{1/2}} \quad (2)$$

For molybdenum, the permeability is [Fraunfelder, 1967]:

$$\Phi = 7.1 \times 10^{-4} e^{-\frac{84,573 \text{ kJ/kmol}}{RT}} \frac{\text{liter}}{\text{cm} \cdot \text{s} \cdot \text{torr}^{1/2}} \quad (3)$$

These are plotted as a function of temperature in Figure 1 below.

Figure 1: Temperature Dependence of Hydrogen Permeability for Nickel and Molybdenum

Hydrogen permeabilities have been determined for many materials of engineering interest. These cover a range from tungsten, with a very low permeability (approx. 5×10^{-11} $\text{torr}^{1/2} \cdot \text{liter/cm} \cdot \text{sec}$ @ 1000K) to nickel (approx. 10^{-6}) to palladium (approx. 10^{-4}).

The strong temperature dependence of the permeability exacerbates the problem of hydrogen accumulation in a heat pipe condenser. By blocking the flow of hot working fluid vapor, the presence of gas in an ordinary heat pipe condenser causes the end of the heat pipe to be colder than the working fluid vapor, sometimes by as much as 100-300°C. This decrease in temperature causes the permeability of the condenser wall to be significantly reduced in exactly the region where a high permeability is desired for the purpose of allowing the hydrogen to permeate out of the heat pipe.

The current development effort aims to demonstrate a heat pipe in which the permeation rate from the condenser can be increased enough so that the condenser will become free of hydrogen within a short time, typically on the order of a few minutes.

2.1 SOLUTION CONCEPTS

Two heat pipe concepts were proposed which can mitigate the hydrogen permeation problem in the bimodal reactor. The first is a two chambered heat pipe which continuously vents hydrogen through a sonic orifice. The second is a heat pipe which has a high permeability hydrogen "window" which allows the hydrogen in the condenser to permeate out at a higher rate. Each of these concepts is described below.

2.1.1 Sonic Orifice Heat Pipe

The sonic orifice heat pipe concept is shown schematically in Figure 2. This heat pipe has a main heat pipe chamber that is connected to a second chamber via a orifice. The main chamber operates at a higher pressure than the second chamber. As a result, as hydrogen permeates into the main heat pipe through the evaporator wall, it is convected to the condenser and forced through the orifice and into the lower pressure chamber. Because of the low pressure in the second chamber, any working fluid which enters the second chamber is readily condensed and returned to the main heat pipe chamber via a capillary wick. Hydrogen passes through the secondary chamber and is vented to space. In summary, the purpose of the sonic orifice in this design is to allow the hydrogen to escape from the heat pipe, and the purpose of the second chamber is to trap working fluid that escapes and return it to the heat pipe.

The orifice heat pipe will not purge itself of noncondensable gas until a significant pressure difference has been created between the two chambers. This is created by simultaneously heating the first (heat pipe) chamber and evacuating the second chamber. Because the noncondensable gas is ultimately exhausted through the second chamber, the sonic orifice heat pipe depends on having a vacuum source available for operation. This vacuum can be supplied either from a vacuum pump for terrestrial applications or from the vacuum of space for space applications. In addition, to minimize the loss of working fluid from the system, the temperature of the second chamber must be maintained low enough that only a small vapor pressure of working fluid exists; if the vapor pressure of the working fluid is too high in the evacuated chamber, working fluid will be lost from the system over time and the heat pipe will cease to function due to lack of working fluid. Both of

these point to the need to seal the exhaust port and the orifice of the heat pipe during periods when the heat pipe is not in use.

Figure 2: Sonic Orifice Heat Pipe Concept. Arrows show flow of hydrogen or (lithium) working fluid.

This concept was demonstrated by Thermacore under Phase I Contract No. DE-FG05-93ER81644, "Innovative Cesium Supply for Bimodal Thermionic Reactors". In the experimental portion of this program, hydrogen and cesium were substituted by helium and water for the purpose of demonstrating the sonic orifice concept. The helium was removed from a water heat pipe as the helium was introduced at a known rate. The operating system removed an 800 Pa partial pressure of helium in approximately 20 seconds, thus demonstrating the capability to rapidly remove a large quantity of noncondensable gas. The Phase II for this program, however, was not funded.

Questions which remain for implementation of this concept in the bimodal reactor include sealing of the orifice and vent during storage, launch, and startup of the heat pipe. In addition, there is concern about the possibility of venting working fluid to space, ultimately resulting in the complete failure of the heat pipe; this gas purging technique has not been demonstrated over periods of time comparable in length to proposed space missions.

2.1.2 Hydrogen Window Heat Pipe

A second concept for venting hydrogen from heat pipes is based on placing a hydrogen "window" in the heat pipe condenser that would increase the hydrogen efflux rate in the condenser. In its simplest form, this can be done either by increasing the surface area available for hydrogen

efflux or using a material with a much higher hydrogen permeability, or both. This concept is shown schematically in Figure 3.

Figure 3: Hydrogen window heat pipe concept. Hydrogen permeates in through the evaporator wall and out through the permeation "windows" in the condenser section.

Although the hydrogen efflux rate in the window design is not likely to be as high as the efflux rate in the sonic orifice design, preliminary calculations showed that it was likely to be high enough to be useful in some applications where the hydrogen permeation rate was relatively low or where the hydrogen permeation rate was high for short times (5-10 minutes) followed by longer recovery periods in which the hydrogen can permeate out of the heat pipes (1-3 hours), such as the bimodal reactor. The properly sized hydrogen window must prevent the hydrogen from blocking the entire condenser during the hydrogen permeation periods so that the heat pipe may still dissipate heat and maintain the evaporator within desired temperature limits.

Because all of the walls of the hydrogen window heat pipe are solid, there is no risk of losing working fluid during operation and no need for special sealing during transport and launch. For this reason, and with the concurrence of the technical monitor, the hydrogen window approach was selected for experimental study during this program.

3.0 DESCRIPTION OF WORK PERFORMED

This section describes the work performed under each of the four technical tasks. These tasks included: establishment of requirements and estimation of hydrogen permeation rates, a proof-of-concept heat pipe design, fabrication and testing of the proof-of-concept heat pipe hardware, and modelling of the experimental hardware. Each of these is described below.

3.1 Task 1: Establish Requirements/Estimate Permeation Rates

Task 1 included two subtasks. The first was definition of requirements, the second was estimation of permeation rates.

3.1.1 Establish Requirements

A kick-off meeting was held at Thermacore for this program on October 28, 1994. In attendance were representatives from DOE (S. Brown), INEL (M. Jacox), and USAF/Phillips Lab (F. Kennedy), as well as Thermacore personnel. The two potential solutions to the hydrogen permeation problem were discussed at the meeting. It was decided at the meeting to concentrate effort on the hydrogen window concept, due to its simplicity (as compared to the sonic orifice approach) and the results of preliminary calculations which showed that its performance would most likely satisfy the needs of the NEBA-1 bimodal reactor. It was felt that the advantages offered by the hydrogen window concept outweighed those of the sonic orifice concept. In addition, the hydrogen window heat pipe would mitigate the need for developing sealing methods for the vented heat pipe. Effort which was proposed to be devoted toward this sealing problem was instead devoted toward additional numerical modelling of the experimental hardware.

Through continuing discussions with the Bimodal Design Team at Phillips Laboratory, the original design parameters evolved during the first two months of the program. The final design goals were set in late December, 1994 based on NEBA-1 design specifications at that time. These are shown in Table 1 below.

Although the goal of this program is not specifically to design a heat pipe for the NEBA-1 reactor, the technology under development in this program would be most directly applicable to the NEBA-1 design. The general goal of the current program is to develop a heat pipe which will allow hydrogen to permeate out of the condenser of a heat pipe under conditions similar to that which would be encountered in the NEBA-1 mission.

TABLE 1: Design Requirements for Hydrogen Tolerant Bimodal Reactor Heat Pipes

<u>Parameter</u>	<u>Design Requirement</u>
Evaporator ID	1.2 cm
Condenser OD	2.54 cm
Evaporator Length	49 cm
Heat Pipe Length	1-2 m
Condenser Length	40 cm
Hydrogen Pressure	2 MPa
Heat Pipe Temperature	1350-1400K (operating) 800-1400K (during cooling)
Heat Pipe Material	Molybdenum
Heat Load	8.4kW (160kW/19 Heat Pipes)
Heat Pipe Life	5-10 years
Duration of Hydrogen Exposure	3 hours (cumulative from 16 cycles over 60 hours)

3.1.2 Estimation of Permeation Rates

The second subtask of Task 1 was to estimate the permeation rate of hydrogen into a heat pipe subject to the conditions outlined in Table 1. Two models were developed for this purpose. The first model was a first order model to estimate the permeation rate in the heat pipes proposed for the NEBA-1 preliminary design. The second model was a slightly more complex model which modelled the heat pipes built and tested during the experimental phase of this program. Details of the first model is discussed in this section; the second model is discussed under Task 4.

A first order model was developed to predict the hydrogen permeation rate in and out of the NEBA-1 sodium/molybdenum heat pipes. The purpose of this model was to quickly generate an estimate for the permeation rate in the NEBA-1 heat pipes. The results of this model were intended to give an estimate of the magnitude of the hydrogen permeation problem and to give an indication

of general areas where improvement was possible. Some of the simplifying assumptions made in this first order model were later relaxed in the second generation model.

The geometry considered is consistent with the values in Table 1 above. The model assumes that hydrogen permeates into the heat pipe through the evaporator and permeates out through the wall of the condenser. Several simplifying assumptions were incorporated into the model; these are listed below.

- 1) Permeation rates through the heat pipe walls and through the hydrogen windows obey Eq. (1).
- 2) Hydrogen which permeates into the evaporator is immediately swept into the condenser and the partial pressure of hydrogen inside the evaporator is zero.
- 3) The partial pressure of hydrogen outside of the condenser is zero.
- 4) The pressure of the hydrogen inside the heat pipe is the same as the vapor pressure of the working fluid at the heat pipe evaporator temperature.
- 5) The entire annulus of the evaporator is exposed to 2MPa hydrogen during propulsion (i.e. during burns), and is not exposed to hydrogen between burns.
- 6) There is a step temperature change between the working fluid vapor and the hydrogen gas pocket in the condenser of 100-200°C.
- 7) The duration of each burn cycle is fixed at 2 hours 35 min., and the duration of each burn is 13.23 min.
- 8) No hydrogen permeation window in the condenser.
- 9) No solution of hydrogen in the liquid sodium.
- 10) Infinite permeability of hydrogen through the liquid sodium saturated wick.
- 11) No hydrogen in the heat pipe at $t = 0$.

Calculations for this model were performed on MathCad. A representative printout of the MathCad document is included as Appendix A. Results from this model are shown below in Figures 4-6. Figure 4 shows the predicted length, normalized by the 40 cm condenser length, of the pocket of hydrogen in the end of the condenser vs. time for a single burn cycle with the heat pipe vapor temperature held fixed at 1400K. A step temperature change of 200°C was assumed for this case between the working fluid vapor and the hydrogen pocket, i.e. the hydrogen pocket was at 1200K. The plot shows that the heat pipe condenser rapidly fills with hydrogen gas during the burn, the time the heat pipe evaporator is actually in contact with hydrogen, due to the high permeation rate of hydrogen at the high temperature and the large hydrogen pressure difference across the evaporator wall. In this case, the model predicts that the length of the hydrogen pocket will decrease gradually over the duration of the burn cycle. At the end of the cycle, approximately

15% of the condenser would be blocked by gas. The total hydrogen left in the heat pipe would be expected to accumulate over multiple cycles.

Figure 4: Results from 1D Model with 1400K Heat Pipe Operating Temperature and 1200K Hydrogen Gas Pocket

Figure 5: Results from 1D Model with 800K Heat Pipe Operating Temperature and 700K Hydrogen Gas Pocket

Figure 5 shows a similar prediction for the case where the heat pipe vapor temperature is 800K. Because the driving force for radiative cooling in the gas blocked portion of the condenser

is lower in this case, the temperature in the gas pocket is assumed to be only 100°C lower in this case. At this significantly lower temperature, the rate of hydrogen permeation into the heat pipe is much lower, however, the length of condenser blocked by hydrogen is increased because of the reduced vapor pressure of the sodium working fluid at this temperature. The rate of hydrogen permeation out of the condenser between burns is likewise reduced by the same two factors: the lower permeability at the lower temperature, and the reduced hydrogen partial pressure difference between the inside and outside of the condenser. As a result of the lower permeation rate out of the heat pipe, the condenser remains nearly entirely blocked at the end of the burn cycle.

Figure 6: Results from 1D Model with 800K Heat Pipe Operating Temperature During Burn and 1400K Operating Temperature After Burn with 1300K Hydrogen Gas Pocket

This model suggests that for constant temperature operation, the hydrogen permeation problem is actually reduced by operating at higher temperatures; even though more hydrogen would permeate into the heat pipe during the burn, the rate of permeation out of the condenser between burns is significantly higher.

Figure 6 shows a prediction for a third operating scenario in which the temperature of the heat pipe is varied during the burn cycle so that the temperature during the burn is 800K, which minimizes the permeation rate into the heat pipe, and during the balance of the cycle the temperature is raised to 1400K, which maximizes the condenser wall permeability and the hydrogen partial pressure difference which drives the permeation process. In this case, the length of condenser blocked is predicted to be approximately 65% of the whole condenser at the end of the burn. When the temperature is suddenly raised to 1400K at the end of the burn, the vapor pressure of the

sodium is increased by a factor of 663, which compresses the hydrogen filled length of the condenser to less than a millimeter. The small mass of hydrogen in this length quickly permeates out of the heat pipe at 1200K. The rapid compression of the gas bubble would offer the benefit of restoring the full use of the condenser for rejecting heat from the condenser for generating electrical energy from thermoelectrics. Varying the operating temperature of the heat pipe in this way would apparently mitigate most of the problem of hydrogen permeation into the heat pipe.

3.2 Task 2: Heat Pipe Design

Under this task, a heat pipe was designed to implement the hydrogen window concept discussed in section 1.2.2. Design consideration was given to several details of the design. These are described in detail below.

Working Fluid

Sodium was selected as the working fluid for the heat pipe based on the temperature range of interest (1000-1100K) and its likely use as a working fluid in bimodal reactor heat pipes.

Wall Material

Selection of the wall material was quickly narrowed to nickel or steel superalloys; refractory metals such as tungsten and molybdenum were ruled out on cost considerations. Nickel was preferred because it is available as a pure material in tube form, and because it does not readily oxidize and form a barrier to hydrogen permeation; in this way it is similar in behavior to the refractory metals. The superalloys sometimes contain trace quantities of materials such as aluminum which can diffuse to the surface and form an oxide coating which can act as a hydrogen permeation barrier. In addition, the hydrogen permeability of pure nickel had been well studied, whereas permeability information on superalloys is not always available. For these reasons, nickel was selected as the wall material for the heat pipe design.

The heat pipe operating temperature was chosen so that the permeation rate through the nickel was similar to the permeation rate through molybdenum near 1400K.

Hydrogen Window

The design of the hydrogen window involved consideration of factors, including the window material, window area, window thickness, and control of the wall temperature in the hydrogen filled portion of the heat pipe condenser.

Because of the choice of the working fluid, the materials from which the windows could be made was limited. Ideally, the window material would be a material with a very high hydrogen permeability. Unfortunately, materials with known high permeabilities, such as palladium, are not compatible with sodium. In addition, the Mo-Pd binary system has a eutectic at 1273K, which is near the operating temperature of the NEBA-1 system. Without an identified material with a permeability at least an order of magnitude higher than that of nickel, nickel was chosen for the window material as well.

Using identical materials for the heat pipe envelope and hydrogen windows eliminated the concern over thermal stresses between the window and heat pipe end cap, as well as the need for special joining techniques for installing dissimilar hydrogen windows into the heat pipe end cap. For example, if nickel hydrogen windows were to be used in a molybdenum heat pipe, a special joint between the window and the end cap would be needed to prevent the joint from reaching the melting point of the Ni-Mo eutectic (1590K) in the welded region.

Without the advantage of higher hydrogen permeability through material selection, the hydrogen window in the present design could only enhance hydrogen permeation through proper geometry selection. According to Equation (1), the permeation rate is increased when surface area is increased and thickness is decreased. This implies that the hydrogen window should have a large surface area and a small thickness. In addition, it is desirable to minimize the volume of the hydrogen window because the windows displace hydrogen in the condenser and reduce the hydrogen holding capacity of the condenser. Because an array of small tubes displaces a smaller volume of the condenser vapor space than a single large tube with the same total surface area, optimal hydrogen window was an array of small diameter, thin-walled tubes.

Heated Gas Pocket

In a conventional heat pipe with noncondensable gas, a gas pocket forms at the end of the condenser of the heat pipe. Because the only path for heat transfer into this pocket is by conduction through the heat pipe wall, significant cooling of this section of the heat pipe occurs.

The cold end of such a gas filled heat pipe is commonly known as a "cold finger", and has been seen to be as much as hundreds of degrees colder than the heat pipe saturation temperature. This is shown schematically in Figure 7; because heat enters the gas pocket from only one end, the temperature decreases monotonically in the cold finger.

Because hydrogen permeability is an exponential function of temperature, this temperature drop reduces the permeation rate of hydrogen out of the condenser to low levels. What is needed is a method for keeping the wall temperature in the gas pocket warm so that the wall permeability is kept as high as possible.

Figure 7: Schematic of Heat Pipe with Hydrogen Gas Cold Finger in End of Condenser with Representative Temperature Profile.

The present heat pipe design included a structure that was devised to keep the gas pocket warmer than would be possible with a conventional heat pipe. The structure is a single closed end annular cavity formed by an inner tube and an annular ring which closes the end of the annulus toward the evaporator. The structure and function of this annular cavity are shown schematically in Figure 8. The cavity is closed to the flow of gas or vapor back toward the evaporator thereby allowing the noncondensable gas to be trapped in the cavity; liquid in the wick, however, can pass the plate, preventing the accumulation of working fluid in the gas cavity. This detail is shown in Figure 9.

Figure 8: Schematic of Heat Pipe with Heated Gas Pocket and Representative Temperature Profile.

Figure 9: Detail of Annular Gas Cavity. Because the endplate extends partway into the liquid saturated wick, liquid bypasses the plate but the gas does not.

The primary purpose of this structure was to keep the noncondensable gas near the wall of the heat pipe. Because hot working fluid vapor flows down the center of the annular gas cavity, the gas and the heat pipe wall in the gas pocket will be heated by radiation and by conduction across the gas pocket (i.e. from the heated inner tube to the heat pipe envelope). In addition, the heat pipe wall is warmed by conduction in the wall from both ends of the gas pocket toward the center of the gas pocket. This provides twice the heat transfer that would be available in a conventional heat pipe where wall conduction into the gas filled region is from one end only.

Because this design depends in part on hot working fluid vapor flowing down the center of the gas pocket, it is important to size the gas cavity properly. The annular gas cavity must be large enough to hold the maximum anticipated volume of gas. If the gas cavity is overfilled, gas will fill the end of the heat pipe and begin filling the center tube, blocking the flow of working fluid from the center tube; the behavior of the heat pipe in this case would be expected to be similar to a conventional gas filled heat pipe and a cold finger would be expected to develop.

Finally, in the current heat pipe design, because the hydrogen gas was expected to collect in the gas pocket, the hydrogen window tubes were placed in a ring so that they protruded into the annular gas cavity. The performance of the windows was expected to benefit as well from the conduction from both ends as well as radiative and conductive heating from the inner wall of the gas cavity.

Detailed engineering drawings of the heat pipe and its assembly were prepared. These are included as Appendix B.

3.3 Task 3: Heat Pipe Fabrication and Testing

Under this task, the heat pipe which was designed under Task 2 was fabricated and tested. This section describes the fabrication and testing procedures and the results of the tests.

3.3.1 Heat Pipe Fabrication

The demonstration heat pipe was a nickel/sodium heat pipe. The overall length of the heat pipe was 61cm and the outside diameter was 2.54cm. The wall of the heat pipe was 0.165cm thick except in the region from the gas cavity to the end of the condenser; in this region the wall was machined to 0.140cm thickness to provide a step for the three positioning tabs on the dividing gas cavity end cap to rest upon.

The wick was composed of -50+100 sintered nickel powder. In addition to providing a means for the liquid working fluid to be returned to the evaporator, the sintered wick held the gas cavity structure in place. Following sintering of the wick, the heat pipe was subjected to a long bake out in a vacuum furnace to liberate any hydrogen or other gases which may have been adsorbed on the inside surface of the heat pipe and the wick structure.

The hydrogen windows consisted of twelve 0.025cm wall by 0.16cm diameter tubes 10.2cm in length which were welded closed at one end. The tubes were electron beam welded into the end cap of the heat pipe, and the end cap was in turn welded into the heat pipe.

For the purpose of collecting the hydrogen which permeated out of eleven of the hydrogen windows, an additional cup was welded onto the end of the end cap. This cup was in turn plumbed to a large volume canister and to a Residual Gas Analysis (RGA) system. Analysis of the gas collected in this chamber was to show that the gas evolved was in fact hydrogen.

The twelfth hydrogen window tube extended out through the gas collection cup; this window was also used as a well for a small diameter thermocouple which monitored the temperature at the end of the hydrogen windows. In addition, two more thermocouple wells were welded into both ends of the heat pipe along the centerline. One extended 30.5cm from the condenser endcap into the condenser vapor space, and the other extended 5.1cm from the evaporator endcap into the evaporator vapor space. Thermocouples were placed in these wells to monitor the internal temperature of the heat pipe.

After welding of the endcaps into place, the heat pipe was again vacuum baked so that adsorbed gases would be minimized. The heat pipe was then charged with 50cc of sodium. Because of the need to insure that all noncondensable gases were purged from the heat pipe, the condenser of the heat pipe was heated during the charging procedure to insure that any noncondensable gases were driven from the gas cavity. This was necessary because of the shape of the gas cavity. Filling and purging of the heat pipe were done through one of the two valves installed in the evaporator end cap.

The heat pipe was instrumented with several thermocouples. In addition to the thermocouples placed in the wells discussed above, 15 thermocouples were brazed to the outside of the heat pipe. To measure the temperature profile in the hydrogen pocket region of the condenser, 10 of these thermocouples were brazed to the outside of the heat pipe in that region. A photo of this region of the heat pipe is shown in Figure 10. Table 2 lists the position of each of the thermocouples in inches from the endcap of the heat pipe. Note that the gas cavity extends approximately 5.5 inches from the end cap.

Figure 10: Close-up View of Condenser End of Heat Pipe. Thermocouples were brazed to the outside wall of the heat pipe. The tube on the right end is the hydrogen exhaust and was connected to the residual gas analyzer.

TABLE 2: Position of Thermocouples on the Outside of the Heat Pipe

Thermocouple Number	Inches from Condenser End Cap
1	0.45
2	1.1
3	1.6
4	2.1
5	2.6
6	3.1
7	3.6
8	4.1
9	4.6
10	5.1
11	6.1
12	16.2
13	19.7
14	23.2
15	24.4

Heat was supplied to the heat pipe by a metal sheathed wire heater. Two heaters were wound tightly around the evaporator in a double helix and brazed on to the heat pipe. This provided good contact between the heater and the heat pipe, and provided redundancy in the event that one of the heaters failed during operation. Thermocouples 12, 13, and 14 were brazed onto the evaporator portion of the heat pipe.

A photo of the completed heat pipe is shown in Figure 11.

Figure 11: Completed Heat Pipe Test Article. Evaporator is to the left with brazed on cable heater, and condenser is on the right with brazed thermocouples. The valve on the lower left was used for processing the heat pipe, the other valve was for injection of hydrogen.

3.3.2 Testing Procedure

After being charged with working fluid, the heat pipe was installed in the test apparatus. The apparatus performed several functions in testing of the heat pipe: heating, injection and collection of hydrogen, and temperature measurement. The heat pipe was plumbed into this system through two connections; the first was a connection to the hydrogen charging system through a tube which penetrated the evaporator end cap, the second was between the hydrogen collection cup and the RGA system. A schematic of the experimental apparatus is shown in Figure 12. A photo of the heat pipe installed in the test facility is shown in Figure 13.

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Figure 12: Schematic of Heat Pipe Testing Facility

Figure 13: Heat Pipe Installed in Test Facility. Except for the condenser, the heat pipe was insulated with several wraps of stainless foil.

The procedure for testing the heat pipe is included as Appendix C. In general terms, each test involved running the heat pipe at a constant power input, injecting a known mass of hydrogen gas, and monitoring the temperature profile in the heat pipe as the hydrogen gas permeated out of the heat pipe. Two series of tests were performed: the first involved injecting a quantity of gas which would fill a fixed length of the condenser at the operating temperature, and the second involved injecting varying quantities of gas at the same temperatures. A matrix of the tests performed is shown below in Table 3. For runs 1 through 10, the injected mass of gas was sufficient to fill approximately 80% of the 10cm long gas cavity, assuming that none of the gas went into solution in the sodium; this mass was determined by calculation. Typical output from a given run was a transient temperature profile in the condenser section of the heat pipe. The run was considered complete when the pressure in the line leading to the RGA no longer increased with time and when the measured temperature profile in the condenser returned to its pre-injection level.

TABLE 3: Matrix of Hydrogen Permeation Tests Performed on Proof-of-Concept Heat Pipe

Run #	Heat Pipe Operating Temp (°C)	Mass of H ₂ Injected (mg)
1	780	0.46
2	740	0.19
3	740	0.19
4	800	0.55
5	800	0.54
6	760	0.38
7	760	0.35
8	720	0.31
9	720	0.29
10	780	0.50
11	740	0.32
12	740	0.42
13	740	0.55

3.3.3 Experimental Results

Experimental results are reported for the conditions shown in Table 3 above. Figures 14-23 show the normalized temperature profile in the condenser section of the heat pipe at various times after injection of the known mass of hydrogen into the heat pipe ($t=0$) for runs 1-10. The temperatures were normalized by the steady state absolute temperature measured at each position just before $t=0$. The figures are arranged in order of decreasing temperature. Experiments at each temperature were repeated at least once to demonstrate the repeatability of the results. Good repeatability was seen between runs at the same temperature.

The gas pocket appears as a depression in the normalized temperature profile in the condenser region of the heat pipe. In each case except for runs 2 and 3 (740°C), the initial length of the hydrogen charge appears to be approximately 8cm in length, as was expected. At increasing time, the hydrogen pocket appears to grow shorter and to move forward in the gas cavity (toward the closed end). This effect was seen in every case, and is most likely due to the balance between the momentum of the condensing vapor entering the gas cavity and heat conduction down the wall of the gas cavity which tends to elevate the working fluid vapor pressure on the closed end of the gas cavity and prevent the gas pocket from being pushed all the way to the closed end of the gas cavity.

Runs 2 and 3 were apparently undercharged due to an error in determining the correct gas charge early in the experiment sequence. The correct charge would have been twice as large. The 0.19mg of hydrogen filled less than 80% of the gas cavity, however the runs were also quite repeatable and the same trend in gas pocket movement with time was seen. In every case, the hydrogen gas introduced into the heat pipe was seen to completely and rapidly permeate out of the heat pipe. Typically the effect of the gas on the condenser temperature profile was minimized within 10 minutes.

Figure 14: Transient Temperature Profiles for Run #4, 0.55mg H₂ @ 800°C

Figure 15: Transient Temperature Profiles for Run #5, 0.54mg H₂ @ 800°C

Figure 16: Transient Temperature Profiles for Run #1, 0.46mg H_2 @ $780^\circ C$

Figure 17: Transient Temperature Profiles for Run #10, 0.50mg H_2 @ $780^\circ C$

Figure 18: Transient Temperature Profiles for Run #6, 0.38mg H_2 @ $760^\circ C$

Figure 19: Transient Temperature Profiles for Run #7, 0.35mg H_2 @ $760^\circ C$

Figure 20: Transient Temperature Profiles for Run #2, 0.19mg H_2 @ $740^\circ C$

Figure 21: Transient Temperature Profiles for Run #3, 0.19mg H_2 @ $740^\circ C$

Figure 22: Transient Temperature Profiles for Run #8, 0.31mg H_2 @ 720°C

Figure 23: Transient Temperature Profiles for Run #9, 0.29mg H_2 @ 720°C

Runs 3, 11, 12, and 13 were all conducted at 740°C with increasing quantities of injected hydrogen. Results for these runs are plotted in Figures 24-27 with the same ordinate range. As expected, the size of the gas bubble increased with increasing mass of hydrogen injected. The size of the gas bubble is evidenced by the width and depth of the temperature profile depression in Figures 24-27. The length of time for recovery of the heat pipe, i.e., for the temperature profile to return to the $t=0$ state, also increased as more hydrogen was introduced.

The 0.55mg charge of hydrogen in run 13 (Fig.27) was deliberately too large to fit in the gas cavity of the heat pipe. With more hydrogen gas than the gas cavity could hold, the gas filled the end of the heat pipe and backed up the inside of the inner tube of the gas cavity. Because the gas cavity was the only active condenser region of the heat pipe, this caused the blockage of the condenser. Continued addition of heat at the evaporator caused the heat pipe temperature to rise, increasing the vapor pressure of the working fluid. The increased vapor pressure compressed the hydrogen gas until it did fit in the gas cavity, allowing normal operation of the heat pipe to resume and the hydrogen to permeate out through the hydrogen windows in the gas cavity and through the outer wall of the heat pipe. Recovery took a longer period of time with this large charge of hydrogen, but it was complete and it did proceed as expected.

Figure 24: Transient Temperature Profiles for Run #3, 0.19mg H₂ @ 740°C
(same data as Figure 21)

Figure 25: Transient Temperature Profiles for Run #11, 0.32mg H₂ @ 740°C

Figure 26: Transient Temperature Profiles for Run #12, 0.42mg H₂ @ 740°C

Figure 27: Transient Temperature Profiles for Run #13, 0.55mg H₂ @ 740°C

3.4 Task 4: Numerical Model of Experimental Heat Pipes

As a result of selection of the hydrogen window concept, the originally proposed Task 4, development of a method for sealing the sonic orifice and vent, was no longer necessary. With the concurrence of the technical monitor, Task 4 was redefined to include additional modelling of the experimental hardware. This modelling was in addition to that performed under Task 1.

The purpose of this model was to relax several of the simplifying assumptions which were made in the earlier model to more accurately predict the response of the heat pipe to the introduction of hydrogen gas and the ability of the hydrogen window to purge this gas from the heat pipe. Some of the features included in this model which were not built into the model from Task 1 included solution of hydrogen gas in the liquid sodium working fluid, variable concentration of

hydrogen gas in the gas pocket, and conduction of heat in the walls of the heat pipe. A complete list of assumptions that were included in the numerical model follows:

- 1) One-dimensional gas and working fluid flow in the heat pipe.
- 2) Solution of hydrogen gas in the liquid sodium working fluid.
- 3) Hydrogen in the gas phase behaves as an ideal gas.
- 4) At $t=0$, the gas pocket suddenly occupies a steady state position in the condenser.
- 5) The partial pressure of hydrogen in the gas pocket drives the permeation through the condenser wall and the hydrogen window (if present). The presence of the liquid sodium or of the hydrogen dissolved in it does not impede the permeation of hydrogen through the wall.
- 6) The flow of liquid working fluid in the wick does not affect the temperature profile in the wall nor the conduction of the heat in the wall, i.e. Pe is assumed to be zero.
- 7) The partial pressure of hydrogen outside the heat pipe is zero.
- 8) The thermal conductivity of liquid sodium is approximately the same as the thermal conductivity of nickel; therefore, the wick/working fluid matrix acts as a homogeneous solid material for the purpose of computing heat conduction.
- 9) The temperature profile in the wall of the heat pipe is determined by conduction from the ends of the gas pocket toward its center and radiation from the outside wall of the condenser to the ambient surroundings. The surroundings act as a blackbody; the heat pipe wall acts as a gray body.
- 10) The temperature profile in the hydrogen windows is the same as the heat pipe wall.
- 11) Outside of the gas filled region, the temperature of the heat pipe is uniform.
- 12) No radiative or conduction heat transfer inside the tube to other portions of the tube.

Figure 28 below shows a schematic of the modelled section of the heat pipe and the processes which were considered. Fluid and gas flow within the heat pipe were again assumed to be one dimensional along the axis of the heat pipe; however, the temperature of the wall of the heat pipe was allowed to vary due to the conduction of heat along the wall and radiation from the outer surface of the wall to the surroundings. Convection of heat by the liquid sodium was assumed to be negligible, however the liquid was assumed to be a path for conduction. The wall temperature of the hydrogen windows was assumed to vary as the condenser wall temperature due to their proximity to the outer wall and to the fact that both ends of the window are immersed in the working fluid vapor at the operating temperature of the heat pipe thus allowing the same double ended conduction process as in the wall to take place.

Figure 28: Model Geometry for the Experimental Gas Cavity . The model considers conduction and radiation in the wall of the heat pipe as well as conduction in the hydrogen window tube. The gas cavity is assumed to be symmetric about the centerline of the gas pocket.

Because the temperature of the sodium varied along the length of the condenser, the vapor pressure of the sodium was also assumed to vary in the axial direction. The pressure inside the heat pipe was assumed to be uniform throughout, i.e. the small pressure differences which are required to circulate the working fluid were assumed to be negligible, so that the partial pressure of hydrogen at any axial position is the difference between the sodium vapor pressure at the heat pipe (evaporator) operating temperature and the vapor pressure of sodium at the local wall temperature. This is represented by equation 4:

$$P_{H_2}(x) = P_{sat}(T_{evap}) - P_{sat}(T(x)) \quad (4)$$

The gas pocket was assumed to be symmetric about the center of the gas pocket, i.e. that heat was conducted toward the center of the gas pocket equally from both ends. From a computational standpoint, this allowed the problem to be divided in half, with the calculated result from the first half simply being reflected over the center of the gas filled region.

The vapor pressure of sodium over the temperature range of interest bounds the maximum partial pressure of hydrogen present in the heat pipe. Because these pressures are too low for hydride formation, the hydrogen is present in the sodium as a dissolved species only. The solution of hydrogen in the liquid sodium was based on the work of McClure and Halsey⁵. The concentration of dissolved hydrogen in liquid sodium is related to the partial pressure of hydrogen by the following equation

$$\chi = \left(\frac{P_{H_2} e^{\frac{1007 \text{ K}}{T}}}{8.2914 \times 10^9} \right)^{\frac{1}{2}} \quad (5)$$

where χ is the number of moles of H_2 dissolved in each mole of Na, P is the partial pressure of hydrogen in Pa, and T is the absolute temperature of the sodium.

The temperature profile in the radiatively cooled cylindrical tube is governed by the following:

$$\frac{d}{dx} \left(k \frac{dT}{dx} \right) - \frac{\epsilon \sigma D}{\delta(D-\delta)} (T^4 - T_{amb}^4) = 0 \quad (6)$$

where T is the wall temperature, k the wall/wick/liquid thermal conductivity (assumed constant), ϵ the wall emissivity, σ the Stefan-Boltzman constant, D the outer diameter of the heat pipe, δ the thickness of the wall and wick, and T_{amb} is the absolute temperature of the surroundings. The boundary conditions applied to equation (6) were $T = T_{evaporator}$ at the edge of the gas bubble and $dT/dx = 0$ at the center of the bubble (or the end cap). The problem is thus a cylindrical hollow fin with a known base temperature and an adiabatic tip. However, because the length of the bubble is initially unknown and is changing with time, the length of the fin is not known *a priori*. The solution of the problem then involves guessing the length of the fin, solving the temperature profile, computing the mass of hydrogen which is accommodated by such a length, and then adjusting the length up or down to satisfy the hydrogen mass balance. The nonlinear ordinary differential equation (6) was solved numerically using the source term linearization technique described by Patankar⁶.

The complete algorithm used to solve the above system of equations is given below.

- 1) For the known initial mass of hydrogen, guess the length of the bubble.
- 2) For the guessed length, solve for the temperature distribution using Eqn. (6).
- 3) From the calculated temperature profile, calculate the sodium vapor pressure profile.
- 4) Calculate the hydrogen partial pressure profile, using Equation (4).
- 5) Determine the mass of hydrogen in solution, using Equation (5).
- 6) Add the total mass of hydrogen

- 7) If the mass of hydrogen is less than the initial mass of hydrogen, increase the guessed length. Go to step 2.
If the mass of hydrogen is more than the initial mass of hydrogen, decrease the guessed length. Go to step 2.
- 8) Calculate the permeation rate out of the heat pipe based on the hydrogen pressure and temperature profiles.
- 9) Adjust the mass of hydrogen in the heat pipe, and increment time.
- 10) Go to 1 until little or no hydrogen is left in the heat pipe.

The above algorithm was implemented through a FutureBasic (Macintosh) computer program. A printout of the computer code is included in Appendix D. All dimensions and material properties were based on the prototype heat pipe, except for the length of the gas cavity-- the model assumes that the cavity is long enough to hold whatever charge of gas is present in the heat pipe at $t = 0$.

Figure 29 shows calculated results for the mass of hydrogen remaining inside the heat pipe vs. time at various temperatures with an initial charge of 0.5mg of hydrogen. As expected, the rate at which hydrogen permeates out of the heat pipe increases with increasing temperature. At 740°C, approximately 1600 seconds are required to remove 99% of the gas, whereas at 800°C only 1000 seconds are required.

The mass of gas in the heat pipe corresponds to a length of gas bubble at a given temperature. Because it is the length of the gas slug which is often of greater interest, the corresponding length versus time plot is shown in Figure 31. The general trend with temperature is the same with length as it was for mass, however because of the nonlinear solubility effect the length does not decrease by 99% at the same rate that was seen above for mass.

Figure 29: Calculated Hydrogen Mass in the Heat Pipe with Hydrogen Windows vs. Time. The initial charge of hydrogen was 0.5mg for each temperature.

Figure 30: Calculated Length of Hydrogen Bubble in the Heat Pipe with Hydrogen Windows vs. Time. Because the initial charge of hydrogen was 0.5mg for each temperature, the initial lengths are different.

Temperature profiles in the condenser were also calculated using the model. Figures 31 and 32 show the predicted temperature profiles at various times for 740°C and 800°C, respectively. In Figure 31, the initial mass of hydrogen was 0.50mg and in Figure 32 the initial mass was 0.72mg; these masses both result in an initial length of hydrogen of 8 cm. These masses are slightly higher than those that were used during the experiments due to the added consideration of hydrogen dissolving in the liquid sodium within the gas filled region. The model predicted that the greater mass of hydrogen permeated out of the heat pipe more quickly due the strong influence of the higher temperature. Interestingly, there was little predicted difference in the depth of the temperature profile despite significantly different operating temperatures and total gas masses. This was also seen experimentally, where in most cases the minimum in the normalized temperature profile was approximately 0.97; the difference between these values remains unexplained. One possible source is that heat transfer between the inner and outer tubes is more significant than was assumed. The shift in the center of the gas pocket with time which was observed experimentally was not predicted by the model; the cause of this effect is evidently due to physics which were not included in the model.

Figure 31: Normalized Condenser Temperature Profile at Various Times (time in seconds).
Initial H₂ mass = 0.50 mg, heat pipe operating temperature is 740°C.

Figure 32: Normalized Condenser Temperature Profile at Various Times (time in seconds).
Initial H_2 mass = 0.72 mg, heat pipe operating temperature is 800°C .

The model was also used to predict the effect of the hydrogen windows and the gas cavity on the rate at which hydrogen was removed from the heat pipe. Figure 33 shows the results of three calculations in which the same mass of hydrogen was present at $t=0$ at the same temperature for three different geometries: the first was an ordinary straight heat pipe without hydrogen windows and without a gas cavity, the second was a straight heat pipe with hydrogen windows (12 small tubes in the same configuration as the experimental prototype), and the third was the model of the prototype, i.e., with hydrogen windows and a gas cavity.

As seen in Figure 33, the ordinary heat pipe removes hydrogen at the slowest rate while the prototype geometry removes hydrogen nearly twice as fast. A significant portion of the gas cavity/hydrogen window advantage apparently comes from the hydrogen windows alone, as evidenced by the middle curve. The factor of two improvement in the rate at which hydrogen can be removed from the hydrogen window/gas cavity is a significant result for systems such as the NEBA-1 reactor where rapid removal of large quantities of hydrogen is likely to be important.

Figure 33: Predicted Mass of Hydrogen vs. Time for Three Different Heat Pipe Geometries. Initial mass of hydrogen is 0.5mg and heat pipe temperature is 800°C for all cases.

4.0 CONCLUSIONS AND RECOMMENDATIONS

The program was fully successful. All technical goals were met or exceeded. The following conclusions can be made based on the above described work.

- Permeation of hydrogen into the NEBA-1 heat pipes will fill the condenser end of the heat pipes before the end of the mission unless steps are taken to minimize permeation into the heat pipes or maximize permeation out of the heat pipes or both. One possible solution is to vary the temperature of the heat pipe so that the heat pipe is cooler during burns than between burns.
- Putting a hydrogen window in a heat pipe in the form of an array of thin-walled, small diameter tubes will increase the rate at which hydrogen can permeate out of the heat pipe.
- Installation of a method to keep the gas pocket hot, such as a closed ended annular gas cavity, can also help to increase the rate at which hydrogen permeates out of the heat pipe.
- A heat pipe with an annular gas cavity and hydrogen windows was successfully fabricated and tested. The heat pipe typically removed the hydrogen in 5 to 10 minutes.

The following recommendations are made for further work in this area.

- Based on the favorable results using nickel at 720-800°C, fabrication and testing of a high temperature heat pipe in the materials under consideration for NEBA-1 is recommended.
- A more detailed thermal/fluid model of the heat pipe should be constructed which builds on the present work. This future model would include the effects of heat transfer between the inner and outer walls of the gas cavity, and convective heat transfer between the working fluid, the heat pipe wall, and the noncondensable gas.
- Alternative geometries which might more efficiently keep the hydrogen gas pocket(s) hot might also be investigated. Also, investigation of structures which could efficiently pack large hydrogen window surface areas into a small volume would also be useful.

5.0 REFERENCES

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APPENDIX A

PRINTOUT OF MATHCAD MODEL FOR ESTIMATING PERMEATION RATES

Hydrogen Resistant Heat Pipe

11-1272

December 27, 1994

Objective

The NEBA-1 bimodal reactor design has a molybdenum/sodium heat pipe evaporator that is exposed to hydrogen during the propulsion mode. Hydrogen which permeates into the heat pipe will start to fill up the condenser, decreasing the power generation capabilities. Jacox has tested a molybdenum heat pipe with 50 psi of hydrogen. He finds that the hydrogen rapidly fills the heat pipe condenser. However, venting is very slow, since the portion of the condenser that contains the gas is colder than the rest of the heat pipe.

One option to increase the venting rate is to add a single tube (or multiple tubes) inside the heat pipe, possibly using a different material with a higher permeation rate. The tube would run the length of the condenser, but not extend into the evaporator. Palladium is one possible material for the tube, since its permeation rate is several orders of magnitude higher than molybdenum. However, palladium is probably not compatible with sodium; it is not compatible with lithium. In addition, platinum - same column in the periodic chart - is not compatible with sodium.

This analysis will examine the use of multiple vent tubes in a sodium heat pipe for the purpose of allowing the hydrogen to permeate out through the condenser wall and vent tube walls during the periods between propulsion firings. The NEBA-1 heat pipes will be made out of molybdenum. We will perform baseline calculations to see how this material will behave. In addition, we will examine other materials such as nickel or iron, to see if those can form a hydrogen window. A potential problem with nickel is the molybdenum/nickel eutectic has a relatively low melting temperature. Finally, we will look at a stainless steel 304L heat pipe, because this is the material the demonstration heat pipe will be made of.

Units and Constants Definitions

Zero $\equiv$ 0

K $\equiv$ 1   C $\equiv$ K   K_To_C $\equiv$ 273.2 · K   Define needed temperature units

MPa $\equiv$ $10^6 \cdot \text{Pa}$ kPa $\equiv$ $10^3 \cdot \text{Pa}$ Define additional units

s $\equiv$ sec   J $\equiv$ joule   kJ $\equiv$ 1000 · J   N $\equiv$ newton   W $\equiv$ watt

$g = 9.807 \cdot \frac{\text{m}}{\text{s}^2}$ standard gravitational acceleration

BoltzmannConstant $\equiv 1.3805 \cdot 10^{-23} \cdot \frac{\text{J}}{\text{K}}$ gas constant per molecule

StefanBoltzmannConstant $\equiv 5.6696 \cdot 10^{-8} \cdot \frac{\text{W}}{\text{m}^2 \cdot \text{K}^4}$ constant for radiative heat transfer

Avogadro $\equiv 6.0225 \cdot 10^{26} \cdot \frac{1}{\text{mole}}$ note: this is for a kilogram-mole

$$\text{GasConstant} \equiv 8314.4 \cdot \frac{\text{J}}{\text{mole} \cdot \text{K}}$$

note: also for kilogram-mole

$$\text{GasConstantGramMole} := \frac{\text{GasConstant}}{1000}$$

gram-mole gas constant

$$\text{AtmosphericPressure} \equiv 101325 \cdot \text{Pa}$$

standard atmospheric pressure

Hydrogen Properties

$$\text{H2MolecWeight} \equiv 2.016 \cdot \frac{\text{kg}}{\text{mole}}$$

$$\text{H2GasConstant} \equiv \frac{\text{GasConstant}}{\text{H2MolecWeight}}$$

$$\text{H2GasConstant} = 4.124 \cdot 10^3 \cdot \frac{\text{joule}}{\text{kg} \cdot \text{K}}$$

$$\gamma_{\text{H2}} \equiv 1.67$$

hydrogen specific heat ratio

$$\text{H2MoleculeMass} \equiv \frac{\text{H2MolecWeight}}{\text{Avogadro}}$$

$$\text{H2MoleculeMass} = 3.347 \cdot 10^{-27} \cdot \text{kg}$$

Hydrogen Critical Properties: R. C. Reid, J. M. Prausnitz, and T. K. Sherwood, *The Properties of Gases and Liquids*, McGraw Hill, N.Y., N.Y., 1977.

$$\text{H2CritTemp} \equiv 33.2 \cdot \text{K}$$

$$\text{H2CriticalPressure} \equiv 12.8 \cdot \text{AtmosphericPressure}$$

$$\text{H2VCritical} \equiv 0.065 \cdot \frac{\text{m}^3}{\text{mole}}$$

kg-mole, or 65 cm³/gram-mole

$$\text{H2ZCritical} \equiv 0.305$$

Collision Cross-Section Reference: P. W. Atkins, *Physical Chemistry*, W. H. Freeman and Company, San Francisco, California, 1978.

$$\text{H2CollisionCrossSection} \equiv 1.5 \cdot 10^{-19} \cdot \text{m}^2$$

Hydrogen Sound Speed, m/s

$$\text{H2SoundSpeed}(\text{Kelvin}) \equiv \sqrt{\gamma_{\text{H2}} \cdot \frac{\text{GasConstant}}{\text{H2MolecWeight}} \cdot \text{Kelvin}}$$

$$\text{H2SoundSpeed}(800 \cdot \text{K}) = 2.347 \cdot 10^3 \cdot \frac{\text{m}}{\text{s}}$$

$$i := 0, 1..12$$

$$\text{Temp}_i := 800 \cdot \text{K} + i \cdot 50 \cdot \text{K}$$

$$\text{H2SoundSpeed}(1400 \cdot \text{K}) = 3.105 \cdot 10^3 \cdot \frac{\text{m}}{\text{s}}$$

$$\text{SoundSpeed}_i := \text{H2SoundSpeed}(\text{Temp}_i)$$

Sodium Properties

May 24, 1994

Reference: *Handbook of Thermodynamic and Transport Properties of Alkali Metals*, 1985.

, Roland W. Ohse, Editor, Blackwell Scientific Publications, Boston, MA,

O. J. Foust, Editor, *Sodium-NaK Engineering Handbook, Volume 1: Sodium Chemistry and Physical Properties*, York, 1972.

Gordon and Breach, Inc., New York, New

$$\text{NaMolecWeight} \equiv 22.9898 \cdot \frac{\text{kg}}{\text{mole}}$$

$$\text{NaMoleculeMass} \equiv \frac{\text{NaMolecWeight}}{\text{Avogadro}}$$

$$\text{NaGasConstant} \equiv \frac{\text{GasConstant}}{\text{NaMolecWeight}}$$

$$\text{NaMoleculeMass} = 0.382 \cdot 10^{-25} \cdot \text{kg}$$

$$\gamma_{\text{Na}} \equiv 1.67$$

sodium specific heat ratio

$$T_{\text{Namelt}} \equiv 370.90 \cdot \text{K}$$

Melting temperature, Ohse, pg. 334

Critical Properties, Ohse, page 99

$$\text{NaCriticalTemperature} \equiv 2507 \cdot \text{K}$$

$$\text{NaCriticalPressure} \equiv 26.16 \cdot 10^6 \cdot \text{Pa}$$

$$\text{NaCriticalDensity} \equiv 209.7 \cdot \frac{\text{kg}}{\text{m}^3}$$

$$\text{NaZCritical} \equiv \frac{\text{NaCriticalPressure}}{\text{NaCriticalDensity} \cdot \text{NaGasConstant} \cdot \text{NaCriticalTemperature}}$$

$$\text{NaZCritical} = 0.138$$

$$\text{NaVCritical} \equiv \frac{1 \cdot \text{kg} \cdot \text{NaZCritical} \cdot \text{NaGasConstant} \cdot \text{NaCriticalTemperature}}{\text{NaCriticalPressure}}$$

$$\text{NaVCritical} = 4.7687 \cdot 10^3 \cdot \text{cm}^3$$

Sodium Vapor Pressure Curve. Ohse, pg. 351, $864 < \text{Kelvin} < 2361$

$$\text{NaVaporPressure}(\text{Kelvin}) \equiv \left(\frac{\text{K}}{\text{Kelvin}} \right)^{0.3869} \cdot \exp \left[11.2916 - \left(\frac{12532.694 \cdot \text{K}}{\text{Kelvin}} \right) \right] \cdot 10^6 \cdot \text{Pa}$$

Na Surface Tension: Ohse, pg 692

$$\text{NaSurfaceTension}(\text{Kelvin}) \equiv 0.1979 \cdot \frac{\text{N}}{\text{m}} - 1.01 \cdot 10^{-4} \cdot \frac{\text{N}}{\text{m}} \cdot \frac{1}{\text{K}} \cdot (\text{Kelvin} - T_{\text{Namelt}})$$

Sodium Liquid Density: Ohse, pg. 459, $T_{\text{melt}} < T(\text{K}) < 2500$

$$\text{NaLiquidDensity}(\text{Kelvin}) \equiv 1000 \cdot \left[0.89660679 + 0.51613430 \cdot \frac{1}{\text{K}} \cdot \left(\frac{\text{Kelvin}}{1000} \right) - 1.8297218 \cdot \frac{1}{\text{K}^2} \cdot \left(\frac{\text{Kelvin}}{1000} \right)^2 \dots \right. \\ \left. + 2.2016247 \cdot \frac{1}{\text{K}^3} \cdot \left(\frac{\text{Kelvin}}{1000} \right)^3 - 1.3975634 \cdot \frac{1}{\text{K}^4} \cdot \left(\frac{\text{Kelvin}}{1000} \right)^4 \dots \right. \\ \left. + 0.44866894 \cdot \frac{1}{\text{K}^5} \cdot \left(\frac{\text{Kelvin}}{1000} \right)^5 - 0.057963628 \cdot \frac{1}{\text{K}^6} \cdot \left(\frac{\text{Kelvin}}{1000} \right)^6 \right] \cdot \frac{\text{kg}}{\text{m}^3}$$

Thermal conductivity from Foust, pg 46, Evangelisti and Isacchini Equation, 105 to 832°C

$$\text{NaThermalCond(Kelvin)} \equiv 100 \cdot \frac{\text{W}}{\text{m} \cdot \text{K}} \cdot \left[0.918 - 4.9 \cdot 10^{-4} \cdot \frac{1}{\text{K}} \cdot (\text{Kelvin} - \text{K\_To\_C}) \right]$$

Sodium Dynamic Liquid Viscosity, Ohse, page 771, $T_{\text{melt}} < T(\text{K}) < 2000$, $\text{Pa} \cdot \text{sec}$

$$\text{NaMuLiquid(Kelvin)} \equiv \left(\frac{\text{Kelvin}}{\text{K}} \right)^{-0.3958} \cdot \exp \left(-6.4406 + \frac{556.835 \cdot \text{K}}{\text{Kelvin}} \right) \cdot \frac{\text{kg}}{\text{m} \cdot \text{sec}}$$

Sodium Dynamic Vapor Viscosity, Ohse, pg 822, assume $x_2=0$, $\text{kg/m} \cdot \text{sec}$

$$\text{NaMuVapor(Kelvin)} \equiv \left[213.1 + 0.165 \cdot \left(\frac{\text{Kelvin}}{\text{K}} - 1000 \right) \right] \cdot 10^{-7} \cdot \frac{\text{kg}}{\text{m} \cdot \text{sec}}$$

Sodium Vapor Density, kg/m^3

For now, assume an ideal gas: see Ohse, Section 6.6 later

$\text{NaZFactor} \equiv 1.0$

$$\text{NaVaporDensity(Kelvin)} \equiv \frac{\text{NaVaporPressure(Kelvin)}}{\text{NaZFactor} \cdot \frac{\text{GasConstant}}{\text{NaMolecWeight}} \cdot \text{Kelvin}}$$

Sodium Vapor Density from PGP program, kg/m^3

$$\text{NaVaporDensityPGP(Kelvin)} \equiv 1000 \cdot \frac{\text{kg}}{\text{m}^3} \cdot \exp \left(-36.89 + 0.04218 \cdot \frac{1}{\text{K}} \cdot \text{Kelvin} - 1.507 \cdot 10^{-5} \cdot \frac{1}{\text{K}^2} \cdot \text{Kelvin}^2 \right)$$

Sodium Vapor Density (LASL) from PGP program, kg/m³

$$\text{NaVaporDensityLASL(Kelvin)} \equiv 23 \cdot 1000 \cdot \frac{\text{kg}}{\text{m}^3} \cdot \left(\frac{3.83 \cdot 10^{10}}{8.314 \cdot 10^7 \cdot \frac{\text{Kelvin}}{\text{K}} \cdot \exp\left(\frac{12180 \cdot \text{K}}{\text{Kelvin}}\right)} \right)$$

Sodium Sound Speed, m/s

For now, assume an ideal gas

$$\text{NaSoundSpeed(Kelvin)} \equiv \sqrt{\gamma_{\text{Na}} \cdot \frac{\text{GasConstant}}{\text{NaMolecWeight}} \cdot \text{Kelvin}}$$

Latent Heat, J/kg

Ohse, page 351

At T=1154.8 K, H=100.486*10³ joule/gram-mole.

Using the Watson relation

Ref: R. C. Reid, J. M. Prausnitz, and T. K. Sherwood, *The Properties of Liquids and Gases*, Third Edition, page 210, McGraw Hill, N.Y., N.Y., 1977.

$$\text{NaLatentHeat(Kelvin)} \equiv 100.486 \cdot 10^3 \cdot \frac{\text{joule}}{\text{mole}} \cdot \frac{1000}{\text{NaMolecWeight}} \cdot \left(\frac{1 - \frac{\text{Kelvin}}{\text{NaCriticalTemperature}}}{1 - \frac{1154.8 \cdot \text{K}}{\text{NaCriticalTemperature}}} \right)^{0.38}$$

Latent Heat from PGP program, joule/kg

$$\text{NaLatentHeatPGP(Kelvin)} \equiv 4186 \cdot \frac{\text{joule}}{\text{kg}} \cdot \left(1215 - 0.2569 \cdot \frac{1}{\text{K}} \cdot \text{Kelvin} + 5.976 \cdot 10^{-6} \cdot \frac{1}{\text{K}^2} \cdot \text{Kelvin}^2 \right)$$

Latent Heat (LASL) from the PGP program, joule/kg

$$\text{NaLatentHeatLASL(Kelvin)} \equiv (1000) \cdot \frac{\text{joule}}{\text{kg}} \cdot \left[5226 + \frac{\text{Kelvin}}{\text{K}} \cdot \left[-1.474 \dots + \frac{\text{Kelvin}}{\text{K}} \cdot \left(0.0003292 - \frac{\text{Kelvin}}{\text{K}} \cdot 5.462 \cdot 10^{-8} \right) \right] \right]$$

Solid Materials Properties

Hydrogen Permeation Rates

Hydrogen Permeability References:

S. Bhattacharyya, E. J. Vesely, and V. L. Hill, "Determination of Hydrogen Permeability in Uncoated and Coated Superalloys: Interim Report," NASA Report NASA CR-165209, January 1981.

R. Fraunfelder, "Permeation of Hydrogen Through Tungsten and Molybdenum," Westinghouse Research Laboratories, Final Report, U.S. AEC Contract AT(30-1)-2833, June 23, 1967.

E. Fromm and E. Gebhardt, *Gase und Kohlenstoff in Metallen*, Springer-Verlag, Berlin/Heidelberg, 1976.

V. A. Maroni, "Control of Tritium Permeation Through Fusion Reactor Structural Materials," Proc., D.O.E. Environmental Control Symposium, Washington, D. C., November 28-30, 1978.

R. C. Svedburg, "Hydrogen Permeability Through Rhenium and an Mo-Re Alloy with and without Chemically Vapor Deposited Tungsten Coatings," *Thin Solid Films*, 72, pp. 385-392, 1980.

Molybdenum Data from Frauenfelder, 250 < C < 1700

$$Q_{pMoly} := 20.2 \cdot 10^3 \cdot \frac{\text{cal}}{\text{mole}}$$

$$Q_{pMoly} = 84.573 \cdot \frac{\text{kJ}}{\text{mole}}$$

$$\text{MolyH2Permeability}(T) := 7.1 \cdot 10^{-4} \cdot \exp\left(\frac{-Q_{pMoly}}{\text{GasConstantGramMole} \cdot T}\right) \cdot \frac{\text{liter}}{\text{cm} \cdot \text{sec} \cdot \text{torr}^{\frac{1}{2}}}$$

$$\text{MolyH2Permeability}(1400 \cdot \text{K}) = 4.9639 \cdot 10^{-7} \cdot \frac{\text{liter}}{\text{cm} \cdot \text{sec} \cdot \text{torr}^{\frac{1}{2}}}$$

$$\text{MolyH2Permeability}(1400 \cdot \text{K}) = 4.299 \cdot 10^{-9} \cdot \frac{\text{m}^3}{\text{m} \cdot \text{sec} \cdot \text{Pa}^{\frac{1}{2}}}$$

Heat Pipe Dimensions

$$\text{EvaporatorOD} := 1.6 \cdot \text{cm}$$

$$\text{EvaporatorID} := 1.2 \cdot \text{cm}$$

$$\text{HeatPipeWallThickness} := \frac{\text{EvaporatorOD} - \text{EvaporatorID}}{2}$$

$$\text{HeatPipeWallThickness} = 0.2 \cdot \text{cm}$$

$$\text{CondenserOD} := 2.54 \cdot \text{cm}$$

$$\text{CondenserID} := \text{CondenserOD} - 2 \cdot \text{HeatPipeWallThickness}$$

$$\text{EvaporatorLength} := 49 \cdot \text{cm}$$

$$\text{CondenserLength} := 40 \cdot \text{cm}$$

$$\text{AdiabaticLength} := 120 \cdot \text{cm}$$

$$\text{HeatPipeLength} := \text{EvaporatorLength} + \text{AdiabaticLength} + \text{CondenserLength}$$

$$\text{HeatPipeLength} = 2.09 \cdot \text{m}$$

$$\text{HeatPipeEffectiveLength} := \frac{\text{EvaporatorLength}}{2} + \text{AdiabaticLength} + \frac{\text{CondenserLength}}{2}$$

$$\text{EvaporatorSurfaceArea} := \pi \cdot \text{EvaporatorOD} \cdot \text{EvaporatorLength}$$

$$\text{EvaporatorSurfaceArea} = 0.02463 \cdot \text{m}^2$$

$$\text{CondenserSurfaceArea} := \pi \cdot \text{CondenserOD} \cdot \text{CondenserLength}$$

$$\text{CondenserSurfaceArea} = 0.03192 \cdot \text{m}^2$$

$$\text{WickThickness} := 0.075 \cdot \text{cm}$$

Assume the vent tube extends the length of the condenser

$$N_{\text{WindowTubes}} := 20$$

$$\text{WindowOD} := 0.060 \cdot \text{in}$$

$$\text{WindowID} := 0.036 \cdot \text{in}$$

$$\text{WindowWallThickness} := \frac{\text{WindowOD} - \text{WindowID}}{2}$$

$$\text{WindowWallThickness} = 0.012 \cdot \text{in}$$

$$\text{WindowLength} := \text{CondenserLength}$$

$$\text{WindowSurfaceArea} := N_{\text{WindowTubes}} \cdot \pi \cdot \text{WindowOD} \cdot \text{WindowLength}$$

$$\text{WindowSurfaceArea} = 383.023 \cdot \text{cm}^2$$

$$\text{VaporCrossSectionalArea} := \frac{\pi}{4} \left[(\text{CondenserID} - 2 \cdot \text{WickThickness})^2 - N_{\text{WindowTubes}} \cdot \text{WindowOD}^2 \right]$$

$$\text{VaporCrossSectionalArea} = 2.745 \cdot \text{cm}^2$$

$$\text{TubeFraction} := 1 - \frac{\text{VaporCrossSectionalArea}}{\frac{\pi}{4} \cdot (\text{CondenserID} - 2 \cdot \text{WickThickness})^2} \quad \text{TubeFraction} = 11.73 \cdot \%$$

$$\text{TubeAreaFraction} := \frac{N_{\text{WindowTubes}} \cdot \pi \cdot \text{WindowOD}}{\pi \cdot \text{CondenserID}} \quad \text{TubeAreaFraction} = 142.43 \cdot \%$$

Operating Conditions

$$\text{H2SourcePressure} := 2 \cdot \text{MPa}$$

$$\text{H2SourcePressure} = 2 \cdot 10^6 \cdot \text{Pa}$$

$$\text{BurnTime} := 794 \cdot \text{s}$$

$$\text{VentHydrogenPressure} := \text{Zero} \cdot \text{Pa}$$

Pressure of hydrogen in the Vent is essentially zero.

$$N_{\text{TimeSteps}} := 1500$$

TimeBeforeNextBurn := 2.578·hr

$Z := 0, 1 \dots N\text{TimeSteps}$

$\Delta\text{Time} := \frac{\text{BurnTime} + \text{TimeBeforeNextBurn}}{N\text{TimeSteps}}$

$\Delta\text{Time} = 6.717 \cdot \text{s}$

$\text{Time}_Z := Z \cdot \Delta\text{Time}$

$\text{HydrogenSupplyPressure}_Z := \text{if}(\text{Time}_Z \leq \text{BurnTime}, \text{H2SourcePressure}, 0 \cdot \text{Pa})$

BurnTemp := 800·K

BetweenBurnTemp := 1400·K

$\text{EvaporatorTemperature}_Z := \text{if}(\text{Time}_Z \leq \text{BurnTime}, \text{BurnTemp}, \text{BetweenBurnTemp})$

Sodium Working Fluid

$\text{HeatPipeVaporPressure}_Z := \text{NaVaporPressure}(\text{EvaporatorTemperature}_Z)$

Assume that the hydrogen bubble in the heat pipe is 200 K colder than the sodium, based on Jacox's experimental data.

HydrogenBubbleSubcooling := 200· K

HydrogenBubbleTemperature := EvaporatorTemperature — HydrogenBubbleSubcooling

HydrogenBubblePressure := HeatPipeVaporPressure

Hydrogen Permeation Rates

Inflow Rate

First calculate the hydrogen permeation into the heat pipe. The hydrogen will be swept to the end of the condenser, so the effective hydrogen pressure for permeation *into* the heat pipe will be zero.

Permeation of hydrogen through a solid at high temperature is well documented and can be assumed to be a diffusion controlled process described by the following equation:

$$\text{MolyH2FluxSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}) := \frac{\text{MolyH2Permeability}(\text{Temperature})}{\text{HeatPipeWallThickness}} \cdot \left(\text{SourcePressure}^{\frac{1}{2}} - \text{SinkPressure}^{\frac{1}{2}} \right)$$

The hydrogen flow rate into the heat pipe is given by

$$\text{MolyH2FlowSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea}) := \text{MolyH2FluxSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}) \cdot \text{SurfaceArea}$$

The above equation for the hydrogen permeation gives the hydrogen flow rate in terms of liters/second at the standard conditions, which are 1 torr pressure and a temperature of 273 K. Use the perfect gas law to translate this to a mass flux.

$$\text{MolyH2MassFlow}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea}) := \frac{(1 \cdot \text{torr}) \cdot \text{MolyH2FlowSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea})}{\text{H2GasConstant} \cdot 273 \cdot \text{K}}$$

The heat pipe that we are looking at uses molybdenum as the wall material, so

$$\text{EvaporatorH2MassFlow}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea}) := \text{MolyH2MassFlow}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea})$$

The above equation is valid for both influx and outflux from the heat pipe. Similar Equations can be written for the vent tube flow rate.

$$\text{H2WindowPermeability}(\text{Temperature}) := \text{MolyH2Permeability}(\text{Temperature})$$

$$\text{H2WindowFluxSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}) := \frac{\text{H2WindowPermeability}(\text{Temperature})}{\text{WindowWallThickness}} \cdot \left(\text{SourcePressure}^{\frac{1}{2}} - \text{SinkPressure}^{\frac{1}{2}} \right)$$

The hydrogen flow rate is given by

$$\text{H2WindowFlowSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea}) := \text{H2WindowFluxSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}) \cdot \text{SurfaceArea}$$

The above equation for the hydrogen permeation gives the hydrogen flow rate in terms of liters/second at the standard conditions, which are 1 torr pressure and a temperature of 273 K. Use the perfect gas law to translate this to a mass flux.

$$\text{H2WindowMassFlow}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea}) := \frac{(1 \cdot \text{torr}) \cdot \text{H2WindowFlowSTP}(\text{Temperature}, \text{SourcePressure}, \text{SinkPressure}, \text{SurfaceArea})}{\text{H2GasConstant} \cdot 273 \cdot \text{K}}$$

Condenser Vent Rate

The hydrogen in the condenser is at the heat pipe vapor pressure, with a known temperature. First, calculate the portion of the condenser that is blanketed by hydrogen as a function of hydrogen mass in the heat pipe.

The hydrogen volume is:

$$\text{H2BubbleVolume}(\text{HydrogenMass}) := \frac{\text{HydrogenMass} \cdot \text{H2GasConstant} \cdot \text{HydrogenBubbleTemperature}}{\text{HydrogenBubblePressure}}$$

Case 0: No Vent Tube, Moly Heat Pipe

First, calculate the hydrogen pressure in the system with no vent tubes.

Define the initial conditions:

1. No hydrogen in the pipe
2. Zero bubble length
3. Initial Permeation Rate into evap
4. No Permeation out of heat pipe
5. No permeation out of vent pipe

$$\begin{bmatrix} \text{HydrogenBubbleMass}_0 \\ \text{HydrogenBubbleVolume}_0 \\ \text{HydrogenBubbleLength}_0 \\ \text{EvapH2InfluxRate}_0 \\ \text{CondH2EffluxRate}_0 \\ \text{VentPipeEffluxRate}_0 \\ \text{HydrogenPermeationRate}_0 \end{bmatrix} := \begin{bmatrix} \text{Zero} \cdot \text{kg} \\ \text{Zero} \cdot \text{m}^3 \\ \text{Zero} \cdot \text{m} \\ \text{MolyH2MassFlow}(\text{EvaporatorTemperature}_1, \text{HydrogenSupplyPressure}_0, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \\ 0 \cdot \frac{\text{kg}}{\text{sec}} \\ 0 \cdot \frac{\text{kg}}{\text{sec}} \\ \text{MolyH2MassFlow}(\text{EvaporatorTemperature}_1, \text{HydrogenSupplyPressure}_0, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \end{bmatrix}$$

$Z := 1, 2 \dots \text{NTimeSteps}$

$$\begin{aligned}
& \left[\begin{array}{l} \text{HydrogenBubbleMass}_Z \\ \text{HydrogenBubbleVolume}_Z \\ \text{HydrogenBubbleLength}_Z \\ \text{EvapH2InfluxRate}_Z \\ \text{CondH2EffluxRate}_Z \\ \text{VentPipeEffluxRate}_Z \\ \text{HydrogenPermeationRate}_Z \end{array} \right] := \left[\begin{array}{l} \frac{\text{HydrogenBubbleMass}_{Z-1} + \text{HydrogenPermeationRate}_{Z-1} \cdot \Delta\text{Time}}{\text{HydrogenBubblePressure}_Z} \\ \frac{(\text{HydrogenBubbleMass}_{Z-1} + \text{HydrogenPermeationRate}_{Z-1} \cdot \Delta\text{Time}) \cdot \text{H2GasConstant} \cdot \text{HydrogenBubbleTemperature}_Z}{\text{HydrogenBubblePressure}_Z} \\ \text{HydrogenBubblePressure}_Z \cdot \left(\text{VaporCrossSectionalArea} + N \cdot \text{WindowTubes} \cdot \frac{\pi}{4} \cdot \text{WindowOD}^2 \right) \\ \text{EvaporatorH2MassFlow}(\text{EvaporatorTemperature}_Z, \text{HydrogenSupplyPressure}_{Z-1}, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \\ \text{MolyH2MassFlow}(\text{HydrogenBubbleTemperature}_Z, \text{HydrogenBubblePressure}_Z, 0 \cdot \text{Pa}, \pi \cdot \text{CondenserOD} \cdot \text{HydrogenBubbleLength}_{Z-1}) \\ \text{Zero} \cdot \frac{\text{kg}}{\text{sec}} \\ \text{MolyH2MassFlow}(\text{EvaporatorTemperature}_Z, \text{HydrogenSupplyPressure}_{Z-1}, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \dots \\ + \text{MolyH2MassFlow}(\text{HydrogenBubbleTemperature}_Z, \text{HydrogenBubblePressure}_Z, 0 \cdot \text{Pa}, \pi \cdot \text{CondenserOD} \cdot \text{HydrogenBubbleLength}_{Z-1}) \dots \\ + \text{Zero} \cdot \frac{\text{kg}}{\text{sec}} \end{array} \right]
\end{aligned}$$

$$\text{Time}_{118} = 792.551 \cdot \text{sec}$$

$$\text{HydrogenBubbleMass}_{118} = 3.021 \cdot 10^{-8} \cdot \text{kg}$$

$$\text{HydrogenBubbleLength}_{118} = 25.336 \cdot \text{cm}$$

$$\text{HydrogenBubbleLength}_{1500} = 0.067 \cdot \text{cm}$$

$$\text{HydrogenPermeationRate}_{118} = 3.811 \cdot 10^{-11} \cdot \frac{\text{kg}}{\text{s}}$$

$$\text{HydrogenBubbleLength}_{119} = 0.077 \cdot \text{cm}$$

Molybdenum Heat Pipe with No Vent Tubes

HydrogenBubbleSubcooling = 200

Case 1: Moly Vent Tubes and Moly Heat Pipe

Repeat the above calculations with moly vent tubes.

$$\begin{bmatrix}
 \text{HydrogenBubbleMass}_0 \\
 \text{HydrogenBubbleVolume}_0 \\
 \text{HydrogenBubbleLength}_0 \\
 \text{EvapH2InfluxRate}_0 \\
 \text{CondH2EffluxRate}_0 \\
 \text{VentPipeEffluxRate}_0 \\
 \text{HydrogenPermeationRate}_0
 \end{bmatrix}
 :=
 \begin{bmatrix}
 \text{Zero} \cdot \text{kg} \\
 \text{Zero} \cdot \text{m}^3 \\
 \text{Zero} \cdot \text{m} \\
 \text{MolyH2MassFlow}(\text{EvaporatorTemperature}_1, \text{HydrogenSupplyPressure}_0, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \\
 0 \cdot \frac{\text{kg}}{\text{sec}} \\
 0 \cdot \frac{\text{kg}}{\text{sec}} \\
 \text{MolyH2MassFlow}(\text{EvaporatorTemperature}_1, \text{HydrogenSupplyPressure}_0, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea})
 \end{bmatrix}$$

Z := 1, 2.. NTimeSteps

$$\begin{aligned}
 & \left[\begin{array}{l} \text{HydrogenBubbleMass}_Z \\ \text{HydrogenBubbleVolume}_Z \\ \text{HydrogenBubbleLength}_Z \\ \text{EvapH2InfluxRate}_Z \\ \text{CondH2EffluxRate}_Z \\ \text{VentPipeEffluxRate}_Z \\ \text{HydrogenPermeationRate}_Z \end{array} \right] := \left[\begin{array}{l} \frac{\text{HydrogenBubbleMass}_{Z-1} + \text{HydrogenPermeationRate}_{Z-1} \cdot \Delta\text{Time}}{\text{HydrogenBubblePressure}_Z} \\ \frac{(\text{HydrogenBubbleMass}_{Z-1} + \text{HydrogenPermeationRate}_{Z-1} \cdot \Delta\text{Time}) \cdot \text{H2GasConstant} \cdot \text{HydrogenBubbleTemperature}_Z}{\text{HydrogenBubblePressure}_Z \cdot \text{VaporCrossSectionalArea}} \\ \text{EvaporatorH2MassFlow}(\text{EvaporatorTemperature}_Z, \text{HydrogenSupplyPressure}_{Z-1}, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \\ \text{MolyH2MassFlow}(\text{HydrogenBubbleTemperature}_Z, \text{HydrogenBubblePressure}_Z, 0 \cdot \text{Pa}, \pi \cdot \text{CondenserOD} \cdot \text{HydrogenBubbleLength}_{Z-1}) \\ \text{Zero} \cdot \frac{\text{kg}}{\text{sec}} \\ \text{MolyH2MassFlow}(\text{EvaporatorTemperature}_Z, \text{HydrogenSupplyPressure}_{Z-1}, 0 \cdot \text{Pa}, \text{EvaporatorSurfaceArea}) \dots \\ + \text{MolyH2MassFlow}(\text{HydrogenBubbleTemperature}_Z, \text{HydrogenBubblePressure}_Z, 0 \cdot \text{Pa}, \pi \cdot \text{CondenserOD} \cdot \text{HydrogenBubbleLength}_{Z-1}) \dots \\ + \text{Zero} \cdot \frac{\text{kg}}{\text{sec}} \end{array} \right]
 \end{aligned}$$

$$\text{Time}_{118} = 792.551 \cdot \text{sec}$$

$$\text{HydrogenBubbleMass}_{118} = 3.021 \cdot 10^{-8} \cdot \text{kg}$$

$$\text{HydrogenBubbleLength}_{118} = 28.703 \cdot \text{cm}$$

$$\text{HydrogenBubbleLength}_{1500} = 0.065 \cdot \text{cm}$$

$$\text{HydrogenPermeationRate}_{118} = 3.811 \cdot 10^{-11} \cdot \frac{\text{kg}}{\text{s}}$$

APPENDIX B

PARTS DRAWINGS FOR PROOF-OF-CONCEPT HEAT PIPE

DETAIL B

DETAIL A

DETAIL D

DETAIL C

REVISIONS <table border="1"> <thead> <tr> <th>LTR.</th> <th>DESCRIPTION</th> <th>DATE</th> <th>APPROVED</th> </tr> </thead> <tbody> <tr><td> </td><td> </td><td> </td><td> </td></tr> <tr><td> </td><td> </td><td> </td><td> </td></tr> <tr><td> </td><td> </td><td> </td><td> </td></tr> <tr><td> </td><td> </td><td> </td><td> </td></tr> <tr><td> </td><td> </td><td> </td><td> </td></tr> </tbody> </table>				LTR.	DESCRIPTION	DATE	APPROVED																					UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO. DRAWN EHD 2/1/95 CHECKED APPROVED MJ Rosta 12/3/95 APPROVED CUSTOMER		THERMACORE, INC. HEAT TRANSFER SPECIALISTS TITLE HyPer VCHP - ASSEMBLY SIZE B DWG. NO. B-11-1272-1 SCALE RELEASE DATE SHEET		
				LTR.	DESCRIPTION	DATE	APPROVED																												
TREATMENT FINISH SIMILAR TO ACT. WT. CALC. WT.			THERMACORE, INC. HEAT TRANSFER SPECIALISTS TITLE HyPer VCHP - ASSEMBLY SIZE B DWG. NO. B-11-1272-1 SCALE RELEASE DATE SHEET																																

REV A 3\13\94 REMOVED SHOULDER ONE END ONLY

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS		
			DRAWN EHD				
TREATMENT			CHECKED		TITLE ENVELOPE		
			APPROVED <i>MJ</i>				
FINISH			APPROVED		SIZE A	DWG. NO. A-11-1272-2	REV A
			CUSTOMER		SCALE	RELEASE DATE	SHEET
SIMILAR TO	ACT. WT.	CALC. WT.					

MATERIAL:
 NICKEL 200

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS			
			DRAWN	EHD	12/3/95			
TREATMENT			CHECKED			TITLE EVAPORATOR END CAP		
			APPROVED	<i>M J Hurd</i>	12/3/95			
FINISH			APPROVED			SIZE		DWG. NO.
						A		A-11-1272-3
SIMILAR TO	ACT. WT.	CALC. WT.	CUSTOMER			SCALE	RELEASE DATE	SHEET

MATERIAL:
NICKEL 200

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS		
			DRAWN	EHD			
TREATMENT			CHECKED		TITLE CONDENSER END CAP		
			APPROVED <i>MJ</i>				
FINISH			APPROVED		SIZE	DWG. NO.	
					A	A-11-1272-4	
SIMILAR TO	ACT. WT.	CALC. WT.	CUSTOMER		SCALE	RELEASE DATE	SHEET

MATERIAL:
NICKEL 200

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES		CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS	
DECIMALS .XX=+.015 -.015 +30' -30'		DRAWN EHD		TITLE VAPOR PLATE	
ANGULAR .XXX=+.005 -.005 DO NOT SCALE DRAWING		CHECKED		SIZE A	
TREATMENT		APPROVED M J Nowak		DWG. NO. A-11-1272-5	
FINISH		APPROVED		SCALE	
SIMILAR TO		CUSTOMER		RELEASE DATE	
ACT. WT.		CALC. WT.		SHEET	

.5" O.D. x .040" WALL
NICKEL 200 TUBING

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS		
			DRAWN EHD	DATE 2/3/95	TITLE AXIAL VAPOR TUBE		
TREATMENT			CHECKED				
			APPROVED <i>MJ</i>	12/3/95	SIZE A		
FINISH			APPROVED				
			CUSTOMER		DWG. NO. A-11-1272-6		
SIMILAR TO	ACT. WT.	CALC. WT.			SCALE	RELEASE DATE	SHEET

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS		
			DRAWN	EHD	DATE	12/3/95	
TREATMENT			CHECKED		TITLE THERMOCOUPLE WELL		
			APPROVED	<i>MJ</i>	DATE	12/3/95	
FINISH			APPROVED		SIZE	A	
			CUSTOMER		DWG. NO.	A-11-1272-7	
SIMILAR TO	ACT. WT.	CALC. WT.			SCALE	RELEASE DATE	SHEET

DIM "L"	QUAN.
4.0"	11
4.625"	1
2.0"	1

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS		
			DRAWN EHD	DATE 2/3/95			
TREATMENT			CHECKED		TITLE VENT/THERMOCOUPLE TUBE		
			APPROVED <i>MJ Hord</i>	DATE 12/3/95			
FINISH			APPROVED		SIZE A	DWG. NO. A-11-1272-8	
			CUSTOMER		SCALE	RELEASE DATE	
SIMILAR TO	ACT. WT.	CALC. WT.			SHEET		

MATERIAL:
1.0" O.D. x .065
NICKEL 200 TUBING

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS		
			DRAWN EHD				
TREATMENT			CHECKED		TITLE Hy-CHAMBER WALL		
FINISH			APPROVED <i>MJ</i>				
SIMILAR TO			APPROVED		SIZE A	DWG. NO. A-11-1272-9	
ACT. WT.		CALC. WT.	CUSTOMER		SCALE	RELEASE DATE	SHEET

MATERIAL:
NICKEL 200

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES DECIMALS ANGULAR .XX=+.015 -.015 +30' -30' .XXX=+.005 -.005 DO NOT SCALE DRAWING			CONTRACT NO.		THERMACORE, INC. HEAT TRANSFER SPECIALISTS			
			DRAWN	EHD				1/3/95
TREATMENT			CHECKED			TITLE Hy-CHAMBER CAP		
			APPROVED	<i>M J Hurd</i>	12/3/95			
FINISH			APPROVED			SIZE	DWG. NO.	
			CUSTOMER			A	A-11-1272-15	
SIMILAR TO	ACT. WT.	CALC. WT.				SCALE	RELEASE DATE	SHEET

APPENDIX C: TEST PROCEDURE FOR HYDROGEN TOLERANT HEAT PIPE

Revision 5-15-95

1.0 OBJECTIVE

The objective of this test procedure is to outline the steps necessary to operate the hydrogen tolerant heat pipe and determine its effectiveness in enhancing hydrogen permeation through the heat pipe walls. Permeation of hydrogen out of the heat pipe in a time comparable to that predicted by the model will confirm the hydrogen window approach to hydrogen permeation enhancement. This work is being carried out under contract to the United States Department of Energy, contract number DE-FG01-94NE32182.

2.0 TEST CONDITIONS

The operational test conditions and a description of the test apparatus are documented in this section.

2.1 Test Apparatus

The test apparatus includes a heat pipe, a hydrogen injecting system, a hydrogen collecting system, and instrumentation. Each of these systems are described below.

2.1.1 Hydrogen Tolerant Heat Pipe

The heat pipe is designed to run in a gravity aided orientation. The heat pipe features an annular pocket in the condenser where vapor flows from the evaporator down a concentric tube and then folds back down a concentric annulus. With the condenser endcap insulated, non-condensable gases will be swept into this pocket, however, condensed working fluid will be capable of flowing through the pocket dividing plate and back to the evaporator.

Since the heat pipe wall in the vicinity of the gas pocket will be heated by conduction from both ends, as opposed to from one end as in a standard straight heat pipe, and by radiation and conduction from the inner tube wall, the hydrogen tolerant heat pipe condenser wall temperature will be higher than in a straight heat pipe. Because hydrogen permeability is strongly temperature dependent, this increased wall temperature will increase the permeation rate of hydrogen out of the heat pipe.

In addition to the wall temperature enhancement, the heat pipe features twelve hydrogen permeation "windows" in the annular gap. These windows are thin-walled small diameter tubes which are closed on one end and protrude through the condenser endcap (with the closed end inside the heat pipe). These windows provide additional surface area for hydrogen permeation; the thinness of the tube walls also provides a permeation enhancement.

The evaporator is heated with an electric resistance heater which is brazed onto the outside wall of the heat pipe. The condenser is cooled by radiation to the inside of the metal bell jar. The entire heat pipe is surrounded by five wraps of stainless foil to reduce radiative cooling except in the region of the annular pocket, which is bare. This will insure that the coolest part of the heat pipe is in the annular pocket region and that the gas collects next to the hydrogen windows.

2.1.2 Hydrogen Injecting System

The heat pipe is connected to a system of valves and small diameter tubing which will be used to inject a known mass of hydrogen into the heat pipe at $t = 0$. A schematic of this system is shown in Figure 1.

The system starts with two tanks of compressed gas: one is argon, which will not be used in the present experiment, and the other is the high purity hydrogen which will be injected into the heat pipe. The valve to the hydrogen line is designated as valve #2. The valve to the argon line should never be opened and is not given a designation.

Connected to the main tubing line next is a vacuum turbo pump. This pump is used to evacuate the injection line prior to charging it with the hydrogen which is later introduced into the heat pipe. The vacuum pump is connected to the main line by valve #1.

A known volume has been placed in the line next, and it has valves immediately at both ends. These are valves #3 and #4. The known volume is 10.6 cm^3 including the volume in the valve bodies on the volume side of the valve.

The line passes into the bell jar and then there is a final high temperature valve, valve #5, before passing into the heat pipe through the evaporator endcap. The hydrogen injection line extends approximately 6" into the evaporator vapor space.

2.1.3 Hydrogen Collecting System

The hydrogen which is injected into the heat pipe is expected to leave the heat pipe via two paths: permeation through the condenser walls, and permeation through the small diameter hydrogen windows. Hydrogen which escapes through the wall of the condenser will go directly into the bell jar, which is pumped independently by a separate vacuum pump. This hydrogen will not be sensed either quantitatively or qualitatively. Hydrogen which escapes through the hydrogen windows* will be collected in the "cup" on the end of the condenser endcap. This cup is connected by a line to a large sample volume with a baratron pressure transducer. The baratron will be able to sense the rise in pressure with time due to the accumulation of hydrogen which permeates out through the hydrogen windows. The large sample volume is connected through valve #6 to a residual gas analyzer (RGA) which is capable of qualitatively detecting the presence of hydrogen gas. The combined indications of pressure rise vs. time in the sample volume and the output of the RGA will be used to determine if hydrogen is continuing to permeate through the windows. No accumulation of hydrogen with time will be one indication of no hydrogen permeation.

* One of the twelve hydrogen windows also acts as a thermocouple well. Hydrogen which escapes through this well is not routed through the RGA; it escapes into the bell jar.

2.1.4 Instrumentation

The heat pipe and the auxiliary valves and tubing are instrumented with 24 thermocouples and 2 pressure transducers. Three of these are located in thermocouple wells inside the heat pipe, and 10 are located on the surface of the condenser wall. The temperature profile indicated by the thermocouples on the condenser wall is expected to give a second indication of the presence of hydrogen; a uniform (or baseline) temperature profile is expected to indicate a hydrogen-free heat pipe. The two pressure transducers will be used for measuring the pressure of hydrogen input and output. The pressure of the sodium vapor inside the heat pipe will not be measured.

Due to the long transient nature of the operation of this heat pipe, the output from all the thermocouples and pressure transducers will be recorded automatically by a data acquisition system

for later analysis. It is anticipated that once the hydrogen is injected into the heat pipe, very little intervention will be required for the normal operation of the heat pipe.

2.2 Test Conditions

- The heat pipe will be operated at a constant temperature of nominally 800°C. If time permits, additional testing may be done at lower temperatures (750°C, 700°C).
- The mass of hydrogen to be injected into the heat pipe will be calculated based on the known volume between valves 3 and 4, accounting for gas which remains in the volume and in the line between valves 4 and 5.
- The pressure in the large sample volume will be maintained at 10^{-5} torr or less. Periodic purging of this volume will be done if necessary to maintain a low enough pressure.

3.0 TEST PROCEDURE

The test procedure is outlined in the steps listed below.

- 3.1 Assemble the test apparatus with the charged heat pipe as shown in Figure 1. Make sure all 6 valves are completely closed. Place the bell jar over the apparatus and evacuate the vacuum system to approximately 10^{-5} torr.
- 3.2 Open valves 1, 3, 4, and 6, and bakeout the output lines and the line between valves 4 and 5. Close all valves except 1 when bakeout is complete.
- 3.3 Start the heat pipe. Increase power until the vapor temperature is 800°C. When the heat pipe reaches steady state record temperature readings for all of the thermocouples, and record the output of the RGA. These will serve as baselines for performance with no hydrogen present.
- 3.4 Charge the hydrogen injection system.
  - a) If necessary, evacuate the hydrogen filling system by opening valves 4, 3, and 1.
  - b) Close valve 4 and then valve 1.

- c) Charge the line with hydrogen by opening valve 2.
- d) When transducer indicates desired pressure (bleed excess through valve 1 if required), close valve 3 and then close valve 2.
- e) Record temperature of known volume so that mass can be calculated.

3.5 Inject hydrogen into heat pipe.

- a) With heat pipe at operating temperature, open valve 4, then open valve 5.
- b) Close valve 5 immediately after hydrogen enters heat pipe to prevent sodium from backstreaming into charging apparatus.
- c) Close valve 4.

3.6 Monitor heat pipe operation with hydrogen in condenser.

- a) Record thermocouple temperatures vs. time.
- b) Record RGA output vs. time.

3.7 Shut down heat pipe

- a) When thermocouple and RGA output suggest that there is no hydrogen left in the heat pipe, slowly decrease the power and watch to see if a cold end develops. If one does not then there is no more hydrogen.
- b) Turn off power.
- c) Check to be sure all valves are closed tightly.
- d) Turn off data acquisition, vacuum pumps, RGA, etc.

11\1272\ 1272 PG 4 .JPG

Figure 1: Heat Pipe Experimental Setup

APPENDIX D

NUMERICAL MODEL COMPUTER CODE LISTING

```
' Version 3.1 of the nickel-sodium heat pipe model tracks the mass of hydrogen
' in a plain straight heat pipe with vent windows and with dissolution of
' hydrogen in the liquid sodium.
```

```
DEFDBL a-z, A-Z
```

```
DIM gGravity, gBoltzmannConstant, gStefanBoltzmannConstant, gPi
DIM gAvogadro, gUnivGasConstant, gAtmosphericPressure, gT_C_to_K
```

```
'Hydrogen Properties
```

```
DIM gH2MolecWeight, gGammaH2, gH2GasConstant, gH2MolecMass
DIM gH2CritTemp, gH2CritPressure, gH2CritVolume, gH2CritZ
DIM gH2CollisionCrossSection
```

```
'Sodium Properties
```

```
DIM gNaMolecWeight, gNaGasConstant, gGammaNa, gNaMeltingTemp
DIM gNaCritTemp, gNaCritPressure, gNaCritDensity, gNaCritZ
```

```
DIM gEvaporatorOD, gEvaporatorID, gEvapWallThickness, gWickThickness
DIM gCondenserOD, gCondenserID, gCondWallThickness
DIM gPocketWallThickness, gPocketWickThickness
DIM gEvaporatorLength, gAdiabaticLength, gCondenserLength, gPocketLength
DIM gEvaporatorSurfaceArea, gCondenserSurfaceArea, gVaporCrossSectionalArea
DIM gH2SourcePressure, gH2VentPressure, gBurnTime, gTimeBetweenBurns
DIM gBurnTemp, gBetweenBurnTemp, gH2BubbleSubcooling, gEpsilon, gPorosity
DIM gNWindows, gWindowDiameter, gWindowThickness, gInnerTubeDiameter
```

```
DIM gNTimeSteps, gNNodes
```

```
DIM gH2BubbleMass, gH2BubbleVolume, gH2BubbleLength, gEvapInfluxRate
DIM gCondEffluxRate, gVentPipeEffluxRate, gHydrogenPermeationRate
DIM gAmbientTemperature, gInitialH2Mass
```

```
DIM gThermalConductivity
```

```
DIM gTemperature(100), gTemperatureStar(100)
DIM gP(100), gQ(100)
```

```
END GLOBALS
```

```
DEF OPEN "TEXT"
```

```
OPEN "O", 1, "NiHPout1-3.2.1-750"
OPEN "O", 2, "NiHPout2-3.2.1-750"
```

```
gPi = ATN(1)<<2
```

```
gGravity = 9.807 'm/s^2
gBoltzmannConstant = 1.3805E-23 'J/K
gStefanBoltzmannConstant = 5.6696E-8 'W/m^2*K^4
gAvogadro = 6.0225E26 'molecules/kgmole
gUnivGasConstant = 8314.4 'J/kgmole*K
gAtmosphericPressure = 101325 'Pa
gT_C_to_K = 273.15 'conversion from celsius to kelvin
```

```
'Hydrogen Properties
```

```
gH2MolecWeight = 2.016 'kg/kgmole
gGammaH2 = 1.405 'ratio of specific heats
gH2GasConstant = gUnivGasConstant/gH2MolecWeight 'J/kg*K
```

```

gH2MolecMass = gH2MolecWeight/gAvogadro'kg/molecule

gH2CritTemp = 33.2 'K
gH2CritPressure = 12.8*gAtmosphericPressure'Pa
gH2CritVolume = 0.065 'm^3/kgmole
gH2CritZ = 0.305

gH2CollisionCrossSection = 1.5E-19 'm^2

'Sodium Properties
gNaMolecWeight = 22.9898 'kg/kgmole
gNaGasConstant = gUnivGasConstant/gNaMolecWeight
gGammaNa = 1.6666667
gNaMeltingTemp = 370.90 'K

gNaCritTemp = 2507 'K
gNaCritPressure = 26.16E6 'Pa
gNaCritDensity = 209.7 'kg/m^3
gNaCritZ = gNaCritPressure/(gNaCritDensity*gNaGasConstant*gNaCritTemp)

gEvaporatorOD = 0.0254 'm
gEvaporatorID = 0.87*0.0254 'm
gEvapWallThickness = (gEvaporatorOD-gEvaporatorID)/2
gWickThickness = 0.080*0.0254 'm

gPocketWallThickness = (0.065-0.010)*0.0254
gPocketWickThickness = (0.080+0.010)*0.0254

gCondenserOD = gEvaporatorOD
gCondWallThickness = gPocketWallThickness
gCondenserID = gCondenserOD - 2*gCondWallThickness

gEvaporatorLength = 8*0.0254 'm
gAdiabaticLength = 0 'm
gCondenserLength = 16*0.0254 'm
gPocketLength = 4*0.0254 'm

gNwindows = 12
gWindowDiameter = 0.062*0.0254 'm
gWindowThickness = 0.010*0.0254 'm
gInnerTubeDiameter = 0.5*0.0254 'm

gEvaporatorSurfaceArea = gPi*gEvaporatorOD*gEvaporatorLength
gCondenserSurfaceArea = gPi*gCondenserOD*gCondenserLength

gVaporCrossSectionalArea = gPi/4*(gCondenserID-2*gWickThickness)^2

gH2SourcePressure = 0 'Pa
gH2VentPressure = 0
gBurnTime = 0 'sec
gNTimeSteps = 1
gTimeBeforeNextBurn = 2*3600 'sec
gDeltaTime = (gBurnTime+gTimeBeforeNextBurn)/gNTimeSteps

gBurnTemp = 1023 'K
gBetweenBurnTemp = 1023 'K
gAmbientTemperature = 300 'K
gH2BubbleSubcooling = 0

gThermalConductivity = 50 'W/mK
gNNodes = 50
gEpsilon = 0.3 'emissivity

```

gPorosity = 0.4

'----- functions -----'

' function initialize draws the results window on the screen

LOCAL FN initialize

WINDOW 1, "1272 Nickel Heat Pipe Model", (0,0)-(SYSTEM(_scrnWidth),SYSTEM(_scrnHeight)-40), _docZoom

END FN

LOCAL FN H2SoundSpeed(AbsoluteTemperature)

END FN = SQR(gGammaH2*gUnivGasConstant/gH2MolecWeight*AbsoluteTemperature)

LOCAL FN NaVaporPressure(AbsoluteTemperature)

END FN = (1/AbsoluteTemperature)^0.3869*EXP(11.2916-12532.694/AbsoluteTemperature)*1E6

LOCAL FN NaVaporPressF(AbsoluteTemperature)

END FN = gAtmosphericPressure*(1/AbsoluteTemperature)^0.5*10^(6.354-5567/AbsoluteTemperature)

LOCAL FN NaSurfaceTension(AbsoluteTemperature)

END FN = 0.1979 - 1.01E-4*(AbsoluteTemperature - gNaMeltingTemp)

LOCAL FN NaSurfaceTenF(AbsoluteTemperature)

END FN = 0.2067 - 1E-4*(AbsoluteTemperature - gT_C_to_K)

LOCAL FN NaLiquidDensity(AbsoluteTemperature)

DensityTemp = 0

DensityTemp = DensityTemp + 0.89660679

DensityTemp = DensityTemp + 0.51613430*(AbsoluteTemperature/1000)

DensityTemp = DensityTemp - 1.8297218*(AbsoluteTemperature/1000)^2

DensityTemp = DensityTemp + 2.2016247*(AbsoluteTemperature/1000)^3

DensityTemp = DensityTemp - 1.3975634*(AbsoluteTemperature/1000)^4

DensityTemp = DensityTemp + 0.44866894*(AbsoluteTemperature/1000)^5

DensityTemp = DensityTemp - 0.057963628*(AbsoluteTemperature/1000)^6

END FN = 1000*DensityTemp

LOCAL FN NaLiquidDensFoust(AbsoluteTemperature)

DensityTemp = 0

DensityTemp = DensityTemp + 0.9501

DensityTemp = DensityTemp - 2.2976E-4*(AbsoluteTemperature-gT_C_to_K)

DensityTemp = DensityTemp + 1.46E-8*(AbsoluteTemperature-gT_C_to_K)^2

DensityTemp = DensityTemp + 5.638E-12*(AbsoluteTemperature-gT_C_to_K)^3

END FN = 1000*DensityTemp

LOCAL FN NaThermalCond(AbsoluteTemperature)

END FN = 100*(0.918 - 4.9E-4*(AbsoluteTemperature-gT_C_to_K))

LOCAL FN NaMuLiquid(AbsoluteTemperature)

END FN = (AbsoluteTemperature)^-0.3958*EXP(-6.4406+556.835/AbsoluteTemperature)

LOCAL FN NaMuVapor(AbsoluteTemperature)

END FN = 1E-7*(213.1+0.165*(AbsoluteTemperature-1000))

LOCAL FN NaVaporDensity(AbsoluteTemperature)

END FN = FN NaVaporPressure(AbsoluteTemperature)/(gNaGasConstant*AbsoluteTemperature)

LOCAL FN NaSoundSpeed(AbsoluteTemperature)

END FN = SQR(gGammaNa*gNaGasConstant*AbsoluteTemperature)

LOCAL FN NaLatentHeat(AbsoluteTemperature)

END FN = 100.486E6*((1-AbsoluteTemperature/gNaCritTemp)/(1-1154.8/gNaCritTemp))^0.38

```

LOCAL FN NiH2Permeability(AbsoluteTemperature)
  QpNickel = 54.81E6          'J/kgmole

END FN = 5.716E-6*EXP(-QpNickel/(gUnivGasConstant*AbsoluteTemperature))'m^3/m*s*Pa^0.5

LOCAL FN EvapH2MassFlow(Temperature, SourcePressure, SinkPressure, SurfaceArea)
  Temporary = 133.32237*SurfaceArea*FN NiH2Permeability(Temperature)
  Temporary = Temporary*(SourcePressure^0.5-SinkPressure^0.5)
  Temporary = Temporary/(gEvapWallThickness*gH2GasConstant*273)
END FN = Temporary

LOCAL FN CondH2MassFlow(Temperature, SourcePressure, SinkPressure, SurfaceArea)
  Temporary = 133.32237*SurfaceArea*FN NiH2Permeability(Temperature)*(SourcePressure^0.5-SinkPressure^0.5)
  Temporary = Temporary/(gCondWallThickness*gH2GasConstant*273)
END FN = Temporary

LOCAL FN VentH2MassFlow(Temperature, SourcePressure, SinkPressure, SurfaceArea)
  Temporary = 133.32237*SurfaceArea*FN NiH2Permeability(Temperature)*(SourcePressure^0.5-SinkPressure^0.5)
  Temporary = Temporary/(gWindowThickness*gH2GasConstant*273)
END FN = Temporary

printflag = 0
FN initialize
PRINT "starting."

'--- define initial state -----
gInitialH2Mass = 0.5e-6
gH2BubbleMass = gInitialH2Mass
gH2BubbleVolume = 0
gH2BubbleLength = 0

gEvapInfluxRate = FN EvapH2MassFlow( gBurnTemp, gH2SourcePressure, 0, gEvaporatorSurfaceArea)
'PRINT "gBurnTemp = "; gBurnTemp
'PRINT "gH2SourcePressure = "; gH2SourcePressure
'PRINT "gEvaporatorSurfaceArea = "; gEvaporatorSurfaceArea
'PRINT "condenser od,id = "; gCondenserOD, gCondenserID
'PRINT USING "%.#####"; (gCondenserOD-gCondenserID)/2
'PRINT "gCondWallThickness = "; gCondWallThickness
'PRINT "gH2GasConstant ="; gH2GasConstant
gCondEffluxRate = 0
gVentPipeEffluxRate = 0
gHydrogenPermeationRate = gEvapInfluxRate

'-- march through time -----
time = 0          'start marching from t = 0
gInitialH2Mass = gInitialH2Mass/2      'this calc is for only half of pocket
gH2BubbleMass = gInitialH2Mass
WHILE gH2BubbleMass/gInitialH2Mass > 0.0001'condition for stopping march through time

  '-- set heat pipe temperature/pressure based on time into mission
  LONG IF time < gBurnTime
    EvaporatorTemperature = gBurnTemp
    TotalPressure = FN NaVaporPressure(EvaporatorTemperature)
    EvapH2InPressure = gH2SourcePressure
  XELSE
    EvaporatorTemperature = gBetweenBurnTemp
    TotalPressure = FN NaVaporPressure(EvaporatorTemperature)

```

```

EvapH2InPressure = 0
END IF
gTemperature(0) = EvaporatorTemperature'root of cold finger fin will be at EvapTemp
gTemperatureStar(0) = EvaporatorTemperature'root of cold finger fin will be at EvapTemp

'-- for the mass of the bubble at a given time, determine its size
lengthcount% = 0          'counter for length do loop
DO
    lengthcount% = lengthcount% + 1    'increment do loop counter

'-- if first pass through do loop, then guess bubble length/temperature distribution
LONG IF lengthcount% = 1
    LONG IF ABS(time) < 1e-5          't=0 special case
        guessLow = 0                  'lowest possible length
        guessHigh = 10*gCondenserLength'guess something much higher than possible
        guessedLength = (guessLow + guessHigh)/2'guess for bisection search
        FOR counter% = 1 TO gNNodes%    'assume all nodes at Evap. Temp.
            gTemperatureStar(counter%) = EvaporatorTemperature
        NEXT
    XELSE
        'for t>0 use previous length as high guess (since it only gets smaller)
        guessLow = 0
        guessHigh = gH2BubbleLength
        FOR counter% = 1 TO gNNodes%    'use previous temp. dist. as guess for new T profile
            gTemperatureStar(counter%) = gTemperature(counter%)
        NEXT
    END IF

'-- if not first pass through loop, then use bisection to get new length guess
XELSE
    LONG IF CalcMassH2 < gH2BubbleMass'guess was too short
        guessLow = guessedLength
        guessedLength = (guessLow+guessHigh)/2
    XELSE
        'guess was too high
        guessHigh = guessedLength
        guessedLength = (guessLow+guessHigh)/2
    END IF
    'use the old T dist. as the guess for the new guessed length
    FOR counter% = 1 TO gNNodes%
        gTemperatureStar(counter%) = gTemperature(counter%)
    NEXT

END IF

DeltaX = guessedLength/gNNodes%    'divide the guessed length into nodes

' PRINT "lengthcount% = "; lengthcount%, "guessedLength = "; guessedLength

'----- for the guessed length, solve for the temperature distribution using the
'----- method outlined in Patankar(1980). assume a 1D radiating fin.
tempcount% = 0          'temperature do loop counter
DO
    tempcount% = tempcount% + 1    'increment counter

'-- calculate all the coefficients needed for the linearized equations
FOR count% = 0 TO gNNodes%-1    'Nodes 0 through N-1 are handled the same
    Spi = -4*gEpsilon*gStefanBoltzmannConstant*gCondenserOD*gTemperatureStar(count%)^3

```



```

Spi = Spi/(gCondWallThickness+gWickThickness)
Spi = Spi/(gCondenserOD-(gCondWallThickness+gWickThickness))

Sci1 = 3*gEpsilon*gStefanBoltzmannConstant*gCondenserOD*gTemperatureStar(count%)^4
Sci1 = Sci1/(gCondWallThickness+gWickThickness)
Sci1 = Sci1/(gCondenserOD-(gCondWallThickness+gWickThickness))
Sci2 = gEpsilon*gStefanBoltzmannConstant*gCondenserOD*gAmbientTemperature^4
Sci2 = Sci2/(gCondWallThickness+gWickThickness)
Sci2 = Sci2/(gCondenserOD-(gCondWallThickness+gWickThickness))
Sci = Sci1 + Sci2

ai = 2*gThermalConductivity/DeltaX-DeltaX*Spi
bi = gThermalConductivity/DeltaX
ci = gThermalConductivity/DeltaX
di = Sci*DeltaX

LONG IF count% = 0
  gP(count%) = 0
  gQ(count%) = gTemperature(0)
XELSE
  gP(count%) = bi/(ai-ci*gP(count%-1))
  gQ(count%) = (di+ci*gQ(count%-1))/(ai-ci*gP(count%-1))
END IF
NEXT

'-- Node N is a special case
Spi = -4*gEpsilon*gStefanBoltzmannConstant*gCondenserOD*gTemperatureStar(gNNodes%)^3
Spi = Spi/(gCondWallThickness+gWickThickness)
Spi = Spi/(gCondenserOD-(gCondWallThickness+gWickThickness))

Sci1 = 3*gEpsilon*gStefanBoltzmannConstant*gCondenserOD*gTemperatureStar(gNNodes%)^4
Sci1 = Sci1/(gCondWallThickness+gWickThickness)
Sci1 = Sci1/(gCondenserOD-(gCondWallThickness+gWickThickness))
Sci2 = gEpsilon*gStefanBoltzmannConstant*gCondenserOD*gAmbientTemperature^4
Sci2 = Sci2/(gCondWallThickness+gWickThickness)
Sci2 = Sci2/(gCondenserOD-(gCondWallThickness+gWickThickness))
Sci = Sci1 + Sci2

ai = gThermalConductivity/DeltaX-DeltaX/2*Spi
bi = 0
ci = gThermalConductivity/DeltaX
di = Sci*DeltaX/2

gP(gNNodes%) = 0
gQ(gNNodes%) = (di+ci*gQ(gNNodes%-1))/(ai-ci*gP(gNNodes%-1))

'----- with the coefficients known, back calculate temperatures
gTemperature(gNNodes%) = gQ(gNNodes%)
FOR count% = gNNodes%-1 TO 1 STEP -1
  gTemperature(count%) = gP(count%)*gTemperature(count%+1) + gQ(count%)
NEXT

'-- calculate RMS difference between this iteration and previous iteration
sum = 0
FOR count% = 1 TO gNNodes%
  sum = sum + (gTemperature(count%) - gTemperatureStar(count%))^2
NEXT

```

```

RMSdiff = SQR(sum/gNNodes%)

'-- move all new temps to old temps array
FOR count% = 1 TO gNNodes%
    gTemperatureStar(count%) = gTemperature(count%)
NEXT

'PRINT tempcount%, RMSdiff
TRON x

UNTIL RMSdiff < 0.001          'terminate loop when iteration changes are small enough
'-- when the preceding UNTIL is satisfied, the temperature profile has been found for the guessed length

'-- calculate the hydrogen pressure distribution, and the mass
'-- of hydrogen in the guessed bubble length
tempMassH2 = 0                'initialize a summing variable
FOR counter% = 1 TO gNNodes%   'march through the nodes
    '--- the hydrogen pressure is the difference between the vapor pressure in the evaporator
    '--- and the local wall/sodium temperature
    ph2 = (FN NaVaporPressure(EvaporatorTemperature) - FN NaVaporPressure(gTemperature(counter%)))

    LONG IF counter% = gNNodes% 'end element volume is only half as big
        volh2 = gPi/4*DeltaX/2*((gCondenserID-2*gWickThickness)^2-gInnerTubeDiameter^2)-gNwindows*gWindowDiameter^2)
        '--- calculate mass in solution
        massNa = FN NaLiquidDensity(gTemperature(counter%))*gPi*gWickThickness*(gCondenserID-gWickThickness)*DeltaX/2*gPorosity
        massH2Solution = massNa*2.017/22.990*SQR(ph2*EXP(1007/gTemperature(counter%)))/8.2914e9)
    ELSE
        'volume of a standard element
        volh2 = gPi/4*DeltaX*((gCondenserID-2*gWickThickness)^2-gNwindows*gWindowDiameter^2)
        '--- calculate mass in solution
        massNa = FN NaLiquidDensity(gTemperature(counter%))*gPi*gWickThickness*(gCondenserID-gWickThickness)*DeltaX/2*gPorosity
        massH2Solution = massNa*2.017/22.990*SQR(ph2*EXP(1007/gTemperature(counter%)))/8.2914e9)
    END IF
    '---use ideal gas law to compute mass in the element
    massh2 = ph2*volh2/(gH2GasConstant*gTemperature(counter%))

    tempMassH2 = tempMassH2 + massh2 + massH2Solution 'add the element H2 mass to the total
NEXT

CalcMassH2 = tempMassH2        'put final total into CalcMassH2

'--- check the calculated mass for out of bounds condition.
LONG IF CalcMassH2 < 0
    'these statements should never be executed
    PRINT "Calculated mass less than zero encountered."
    DO                          'pause until any key is hit
        UNTIL LEN(INKEY$)
    END IF

```

```

'PRINT "gh2bubblemass = "; gH2BubbleMass
'PRINT "calcmassh2 = "; CalcMassH2
'PRINT "CalcMassH2/gH2BubbleMass = "; CalcMassH2/gH2BubbleMass

'----- stop loop when calculated mass is "close enough" to gH2BubbleMass
UNTIL ABS((CalcMassH2-gH2BubbleMass)/gH2BubbleMass) < 1E-4

gH2BubbleLength = guessedLength      'the length is the guessed length

PRINT time, gH2BubbleMass, gH2BubbleLength, gH2BubbleMass/gInitialH2Mass
PRINT #2, "time: "; time, gH2BubbleMass, gH2BubbleLength
PRINT #1, "time: "; time, gH2BubbleMass, gH2BubbleLength

'---- calculate the hydrogen influx rate (this is always zero in the present model)
gEvapInfluxRate = FN EvapH2MassFlow(EvaporatorTemperature, EvapH2InPressure, 0, gEvaporatorSurfaceArea)

'---- calculate the hydrogen efflux rate
tempEffluxRate = 0                    'zero the summing variable
FOR counter% = 1 TO gNNodes%-1        'nodes 1 through N-1 are identical
    ph2 = (FN NaVaporPressure(EvaporatorTemperature) - FN NaVaporPressure(gTemperature(counter%)))
    tempEffluxRate = tempEffluxRate + FN CondH2MassFlow(gTemperature(counter%), ph2, 0, gPi*gCondenserOD*Delt
    aX)
    tempEffluxRate = tempEffluxRate + FN EvapH2MassFlow(gTemperature(counter%), ph2, 0, gNwindows*gPi*gWindow
    Diameter*DeltaX)
    PRINT #2, counter%, gTemperature(counter%), tempEffluxRate
NEXT
'node N is a special case
ph2 = (FN NaVaporPressure(EvaporatorTemperature) - FN NaVaporPressure(gTemperature(gNNodes%)))
tempEffluxRate = tempEffluxRate + FN CondH2MassFlow(gTemperature(counter%), ph2, 0, gPi*gCondenserOD*DeltaX
/2+gPi/4*gCondenserOD^2)
tempEffluxRate = tempEffluxRate + FN EvapH2MassFlow(gTemperature(counter%), ph2, 0, gNwindows*gPi*gWindowDi
ameter*DeltaX/2)
PRINT #2, gNNodes%, gTemperature(gNNodes%), tempEffluxRate
gCondEffluxRate = tempEffluxRate      'put final total in global variable

'----- there are no vent pipes yet accounted for
'gVentPipeEffluxRate = 0

'----- calculate the net hydrogen influx rate (outflow will be negative)
gHydrogenPermeationRate = gEvapInfluxRate - gCondEffluxRate

'LONG IF time > 794 AND printflag = 0
'PRINT "max length = "; gH2BubbleLength
'printflag = 1
'END IF

'----- calculate the timestep, time, and bubble mass at the next time step
time = time + gDeltaTime              'new time
oldH2Mass = gH2BubbleMass             'hold old mass in temporary holder
gH2BubbleMass = oldH2Mass + gHydrogenPermeationRate*gDeltaTime

'-- limit the maximum change in mass to 5% of old value

```

```

LONG IF gH2BubbleMass < 0 OR gH2BubbleMass/oldH2Mass < 0.95
'PRINT "Bubble mass decrement too large."
gH2BubbleMass = gH2BubbleMass - gHydrogenPermeationRate*gDeltaTime
time = time - gDeltaTime
DO
'decrease timestep size until mass change is small enough
gDeltaTime = gDeltaTime*0.95
gH2BubbleMass = oldH2Mass + gHydrogenPermeationRate*gDeltaTime
'PRINT "Atime, mass = "; gDeltaTime, gH2BubbleMass
UNTIL gH2BubbleMass > 0 AND gH2BubbleMass/oldH2Mass > 0.95
time = time + gDeltaTime      'increment time using new stepsize
'PRINT "time, massratio = "; time, gH2BubbleMass/oldH2Mass
END IF
'PRINT "gH2BubbleMass = "; gH2BubbleMass

WEND                                'go back to top of loop and step through time

PRINT "done."

DO
'-- wait for a cmd-period before quitting
HANDLEEVENTS
UNTIL cmdPeriod

```