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Noncondensible Gas, Mass, and Adverse Tilt Effects on the Start-up of Loop Heat Pipes

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ABSTRACT

In recent years, loop heat pipe (LHP) technology has transitioned from a developmental technology to one that is flight ready. The LHP is considered to be more robust than capillary pumped loops (CPL) because the LHP does not require any preconditioning of the system prior to application of the heat load, nor does its performance become unstable in the presence of two-phase fluid in the core of the evaporator. However, both devices have a lower limit on input power: below a certain power, the system may not start properly. The LHP becomes especially susceptible to these low power start-ups following diode operation, intentional shut-down, or very cold conditions. These limits are affected by the presence of adverse tilt, mass on the evaporator, and noncondensible gas in the working fluid. Based on analytical modeling correlated to start-up test data, this paper will describe how the minimum power required to start the loop is increased due to the presence of mass, noncondensible gas, and adverse tilt. The end-product is a methodology for predicting a "safe start" design envelope for a given system and loop design.

INTRODUCTION

The LHP requires a pressure difference across the wick in order to start. This pressure drop is related to the temperature differential across the wick in accordance with the Clausius-Clapeyron relation. With no flow in the system and no heat load on the evaporator, no pressure or temperature gradient will exist across the evaporator wick. As heat is applied to the evaporator wall, a temperature gradient will develop across the wick, thus allowing the system to start pumping. For very low heat loads into the evaporator, the conductive paths to the compensation chamber and through the metal wick become dominant and the temperature gradient required to develop a sufficient pressure gradient across the wick is elusive. This is evident in the characteristic LHP performance curve (loop temperature drop vs. heat load) in which the curve depicts an increase in the temperature drop at low powers. In the presence of adverse tilt (evaporator at a higher elevation than the condenser[1]) the required pressure/temperature gradient increases making startup difficult at low powers.

The primary purpose of this effort is to establish the "safe envelope" under which these devices can be started passively. This envelope will aid in design risk mitigation by insuring that if the system were to start under the worst case scenario (coming out of diode operation, or post-shut-down heating of the compensation chamber), the temperature of the electronics being controlled by the LHP will not exceed design limits.

Active measures can be employed to assist in the start-up. For example, a thermoelectric (Peltier) cooling device on the compensation chamber can be used to actively lower the evaporator core temperature thus allowing the LHP to start in a more reasonable amount of time, or with less initial power input. However, there are no guidelines for sizing and controlling this cooler. Establishing such guidelines is a secondary purpose of this investigation.

ANALYTICAL TOOLS

The application of analytical tools for modeling capillary two-phase transport devices has been very limited to date. Many developers and users of this technology have resorted to basic spreadsheet methods which although simple to use are extremely limited in their capabilities and are very design specific. These simplistic methods are not capable of assessing system-level integration issues nor the hydrodynamic transient event of start up.

[1] Adverse conditions are often difficult to avoid during ground operations such as thermal balance testing and possible launch pad operations.

SINDA/FLUINT, the NASA-standard heat transfer and fluid flow analyzer (and its graphical user interface Sinaps*Plus),* is the most complete general-purpose thermohydraulic analyzer available. In addition, it is the only code that features special tools for dealing with capillarity and space and launch environments making it applicable both to detailed start-up transients and to top-level integration studies. SINDA/FLUINT provides tools for modeling the thermodynamic and hydrodynamic behavior of primary and secondary wicks, bayonet heat transfer, wick back conduction, in addition to the effects of mass, gas, and adverse tilt on startup.

Recent enhancements to SINDA/FLUINT (reference 1) have significantly expanded its ability to model complex phenomena of interest to both two-phase technologists and users. The introduction of new modeling elements, *interfaces*, has greatly improved the modeling of quasi-stagnant nonequilibrium control volumes (e.g., compensation chambers and reservoirs) as well as the modeling of liquid-vapor interfaces including those within wicks. While previous capabilities allowed the tracking of mass flowing from control volume to control volume, *interfaces* describe how the boundary between these control volumes moves. The recently added capability of modeling the dissolution and evolution of noncondensible gases (NCG) aids in the tracking of these gases as bubbles and/or solutes and in the prediction of their effect on steady or transient loop performance.

BACKGROUND

Typically the evaporator of the LHP will be attached to some type of heat source. In a realistic application, this source will generally have mass associated with it. If this mass is significant, the effect will be to reduce the amount of heat into the evaporator available to develop the required temperature gradient across the wick. This will result in an increased likelihood of start-up problems.

To quantitatively illustrate this effect, a simple calculation can be made. Assume an evaporator of thermal capacitance (mass times specific heat) C_e is attached to a payload of capacitance C_p. If the LHP has not yet started and the evaporator is not yet removing energy, then the peak power deposited into the evaporator is independent of the bonding resistance:

$$Q_e = \frac{Q_p}{1 + \frac{C_p}{C_e}}$$

The payload normally greatly outweighs the LHP by a factor of 10 or more (after all, the mass of the thermal control system is typically unacceptable if it exceeds 6% of the system mass). In such a case, 30W of dissipation in the payload would be reduced to only 2.7W entering the LHP and hence available to create the temperature differential across the wick.

LHP start-up becomes even more problematic by the addition of adverse tilt. Figure 1 presents the results of a simple spreadsheet analysis demonstrating that a minimum power must be applied to the LHP to overcome gravity. In this case, the payload is assumed to have a maximum temperature limit of 40C, and the sink condition is –20C (representative of space applications). The condenser must provide subcooling sufficient to offset the reverse conduction through the wick created by the static pressure difference, which creates a minimum temperature drop across the wick. The plot shows the sensitivity to wick conductance. A nickel wick of “typical” dimensions has a conductance on the order of 50 W/K, while stainless steel wicks offer conductances on the order of 25 W/K.

The estimates In Figure 1 are low: the minimum power will increase with heat leaks into the liquid line, which have been neglected in this simplified calculation. The effect of increasing the sink temperature to 10C (more representative of ambient testing) is shown in Figure 2. In ambient testing, heat transfer between the environment and the liquid line and compensation chamber is notoriously difficult to prevent since the liquid moves so slowly at low powers. Thus, Figure 2 greatly underestimates the minimum power for ambient testing.

When noncondensible gas (NCG) is present in the working fluid, it will typically come out of solution in the compensation chamber. Due to the partial pressure of the gas in the compensation chamber, a larger temperature gradient will be required to obtain the same pressure differential across the wick. Thus the amount of time required for the system to start after the application of heat will increase slightly in the presence of gas. Combining the effects of gas with mass and adverse tilt on a LHP, the delay in start-up can become significant. If

Figure 1: Minimum Power Required to Overcome Wick Back Conduction and Adverse Tilt with Nominal Sink

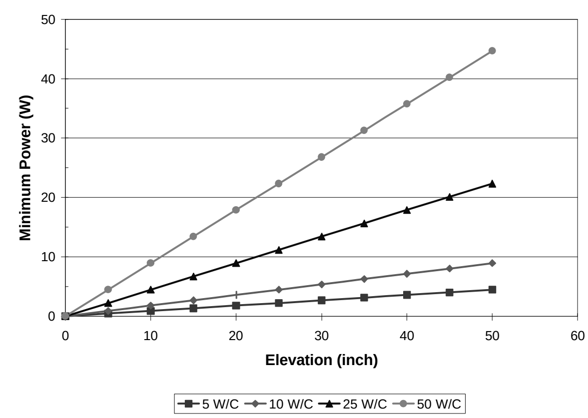
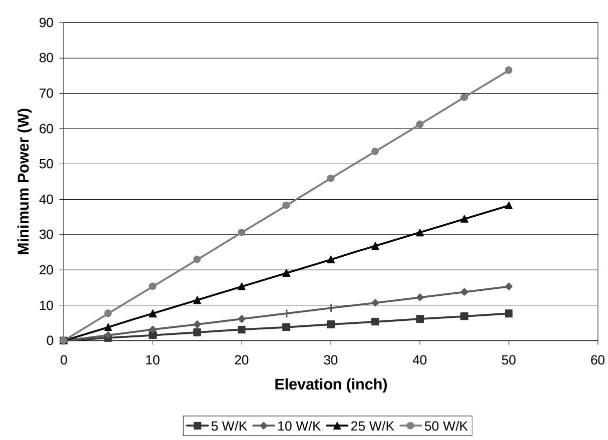


Figure 2: Minimum Power Required to Overcome Wick Back Conduction and Adverse Tilt with 10C Sink



the delay in start-up is significant, the rise in temperature at the heat source may result in the maximum design temperature being exceeded on sensitive electronics. In the case of zero gas, the minimum temperature difference required across the wick is zero. As gas is added, a finite temperature difference must be established before circulation can begin.

While each of these effects (NCG, tilt, mass) have been overcome independently in tests, the combined effects can be serious since they all tend to either reduce the power into the evaporator or increase the minimum temperature differential required to establish forward flow. For example, if Figure 2 predicts a minimum power input of 5W to overcome gravity, then over 50W would have to be applied to an attached plate whose mass is ten times that of the evaporator.

These effects are further complicated by uncertainties in phenomena such as incipient superheat and the location where vaporization first starts. It is possible, for example, for the vapor grooves to remain flooded with superheated liquid while heat flows into the evaporator core. If nucleation first occurs in the core rather than in the vapor grooves, the effect is temporary operation in the reverse direction and pressurization of the LHP, which can endanger the recovery of the device into normal (forward) operation.

STARTUP CONDITIONS

Not all conditions can be tested, and designers require tools for evaluating the relative merits of various system designs in the preliminary design stages. Therefore, *the primary purpose of this effort is to develop an analytic basis for predicting the success or failure of LHP start-up, and to have confidence in the analysis via comparisons with tests. This requires "testing for failure:"* purposely pushing the envelope of experimental conditions such that approximately half the start-ups do *not* succeed.

The phenomena of interest in this study (delay or failure in startup) occur more frequently when the vapor grooves of the evaporator are initially flooded with liquid. This is the condition under which superheating of the liquid prior to nucleation will occur. For the parallel test programs this was achieved through heating the compensation chamber prior to the application of the evaporator heat load. If the vapor grooves are not first flooded, the LHP will always start with no superheat prior to boiling, and is more likely (but not guaranteed) to succeed under adverse conditions of attached mass, NCG, and tilt.

The question then arises as to when such a condition (flooding of the vapor grooves) would occur in an actual system. Most often this condition would exist after a forced or diode shut down. Unlike the related CPL technology, LHPs cannot autonomously shut down below a minimum design temperature. If shut down of the system is required to prevent excessive heat loss from a system (such as a safe mode, cold case, or diurnal non-operation), the shut down must be forced on the system by heating the compensation chamber above the evaporator temperature. On the other hand, adverse starting conditions can happen in *any* LHP in which the condenser environment becomes warmer than the rest of the system, and the resulting intrinsic diode action floods the evaporator grooves.

MODEL DEVELOPMENT

As stated previously, the model was developed in Sinaps*Plus* and SINDA/FLUINT, requiring many of the new features available in Version 4.1. The thermal side of the system was represented by a series of nodes which represent the mass and the storage/release of energy, and conductors which describe how the energy is transported between nodes. A network of nodes and conductors presenting the evaporator and compensation chamber bodies, transport lines, condenser line, and environments, was created which is depicted in Figure 3.

The fluid was modeled as a closed loop system using a series of lumps and paths to model mass transport, evaporation and condensation. Since we were interested in the transient behavior of the system, tanks (control volumes which exchange energy with the thermal network) and tubes (lines with significant inertia) were used. In addition, advanced features such as capillary pumps, wicks (including capillary effects in the vapor grooves and liquid inertia in the wick), interfaces, non-equilibrium routines, and the dissolution and evolution of noncondensible gases were key to the model development. The fluid network is depicted in Figure 4.

Figure 3: SinapsPlus Depiction of Thermal Network

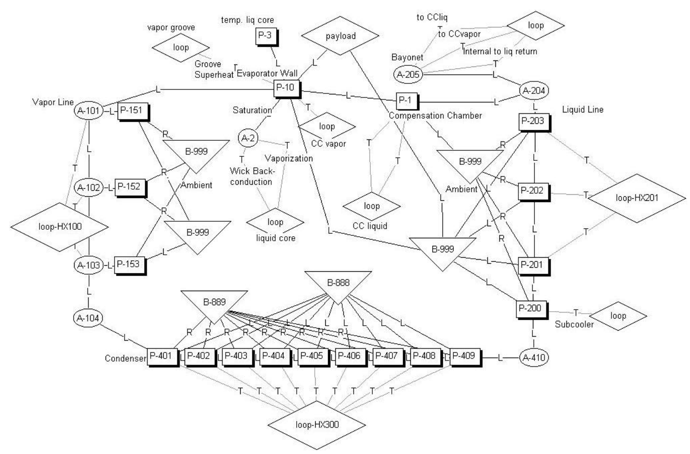
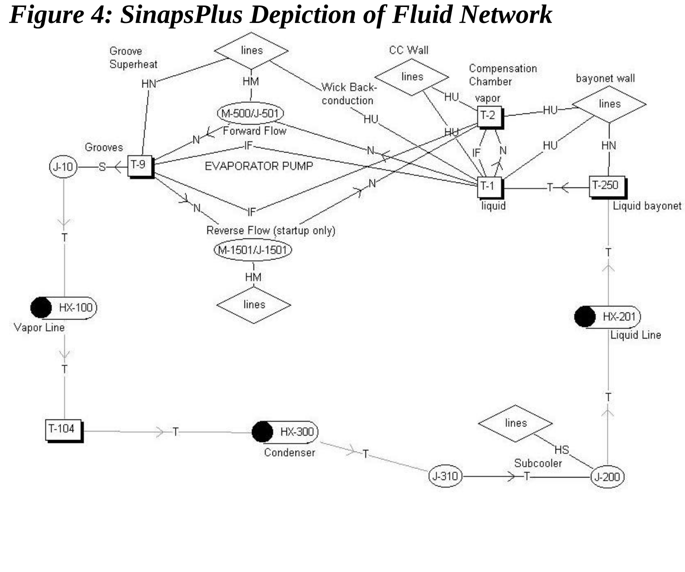


Figure 4: SinapsPlus Depiction of Fluid Network



The goal of the analyses was to determine the effect on start-up of significant evaporator mass, adverse tilt and noncondensible gas in the loop. This required a transient hydrodynamic simulation to which the modeling of the compensation chamber and the evaporator wick were paramount.

COMPENSATION CHAMBER ENERGY BALANCE

Key to modeling LHP startup is the energy balance on the compensation chamber. This energy balance is affected by the conduction through the evaporator wick, the wall conduction from the evaporator to the compensation chamber, the fluid/vapor to wall heat transfer within the compensation chamber and the environmental connections. In addition, some LHP designs may have a significant contribution by the heat transfer with the liquid bayonet. These parameters are all factors in establishing the temperature and pressure differential across the wick required for start-up.

Evaporator Wick Conduction

The back conduction from the evaporator wall through the wick to the core is critical to start-up and has been accounted for in the analyses. The effective conductivity through the wick is based on the Dunn and Reay correlation (reference 2) for sintered wicks.

$$K_{eff} = K_{wick} \times \left[\frac{2+g-2e(1-g)}{2+g+e(1-g)} \right]$$

Where: K_{wick} = thermal conductivity of the dry wick

K_{liq} = thermal conductivity of liquid

ε = wick porosity

$\gamma = K_{liq}/ K_{wick}$

This correlation is used as a first cut, although calibration to test data is required because of the extreme sensitivity of the results to this key unknown. Therefore, an effective wick material conductance well below that of the raw material was used in order to correlate to test data. The conduction through the wick is treated as an effective solid using the standard radial conductance equation:

$$G = \frac{2pK_{eff}L}{\ln\left(R_o / R_i \right)}$$

Where: L = thickness

R_o = outer radius

R_i = inner radius

For cylindrical wicks, the above term is corrected to account for the fact that the wick is wet and the temperature profile within the wick is not as simple as the above formula implies. Rather, the influx of slightly subcooled liquid (relative to the saturation condition at the OD of the wick) and the heat exchange of this fluid with the wick material causes a nonlinear profile. The correction for this heat exchange effect, which is normally rather small, is a function of the current flowrate (FR), specific heat (Cp), etc. as shown below.

$$G_{corrected} = \frac{FR_{liq} \times Cp_{liq}}{\left(R_o / R_i \right)^{\left(\frac{FR_{liq} \times Cp_{liq}}{G_{uncorrected}}\right)} - 1}$$

An analogous correction can be made for flat wicks.

Compensation Chamber Heat Transfer Coefficients

As previously stated, the fluid-to-wall and liquid-to-vapor heat transfer within the compensation chamber is critical to the energy balance. Unfortunately, these parameters are not easily quantifiable. For example, the liquid-to-wall heat transfer coefficients are calculated based on solid conduction or pool boiling correlations.

For pool boiling, the relation developed by Rohsenow (reference 3) through correlated experimental data was used. The estimate based on solid conduction (KA/x) is determined by the following equation:

$$h = F_{liq} \times K_{liq} \times \frac{\left(A \times (1 - \frac{V_{vap}}{V_{liq} + V_{vap}}) \right)^2}{V_{liq}}$$

Where: F = estimated stirring factor (effectively, a Nusselt number that is parameterized)

A = area

V = volume (liquid or vapor)

In the above equation, the characteristic conduction path length is based on the ratio of to surface area of the appropriate phase. In addition, the surface area is estimated based on the void fraction of the fluid inside the compensation chamber.

The model in this study was developed to compare with ground testing of a horizontal compensation chamber. This model could easily be modified to better predict zero-g effects. For example, in zero-g there would be no direct contact between the wall and the vapor.

Vapor Groove Superheat

A known characteristic of LHPs is the ability to occasionally start in the reverse direction. This was taken into account in the analyses by permitting the vapor grooves to be initially flooded. If the degree of superheat required to start boiling in the vapor grooves happens to exceed the superheat requirements of the core, then vaporization takes place first in the core and the system will start with backwards flow. Depending on the magnitude of the superheat required to initiate boiling in the grooves, the LHP may or may not revert to forward flow: the temperature differential required to superheat the grooves and start forward flow may never be developed. If this is the case the start-up fails: the system pressurizes (the reverse flow rates are too small to matter) and the payload overheats.

Two unknowns are assigned to this process: the degree of superheat required to initiate boiling in grooves and the superheat to initiate core boiling. Since these two values are stochastic and can be affected by previous events (hysteresis), their values are left for the user to define. By running a series of parametrics on superheat values, we can learn how they can effect start-up and at what values they may lead to start-up failures.

Noncondensible Gas

The model was developed to account for the presence of both nitrogen and hydrogen noncondensible gases either alone or in combination. The dissolution of gas could be modeled, but it has thus far been neglected since the most of the liquid volume is in the compensation chamber, which is not highly subcooled and therefore cannot contain much dissolved gas.

The presence of gas in the compensation chamber will effectively increase the temperature differential across the wick required for startup. The Clausius-Claperyron relation (reference 4) below provides the relationship between the pressure and temperature gradients across the wick for startup.

$$\frac{dT}{dp} = \frac{T_{vap}\ v_{fg}}{h_{fg}}$$

Where: T = temperature (vapor, liquid)

p = pressure

v_{fg} = delta (liquid, vapor) specific volume

h_{fg} = latent heat of evaporation

When the partial pressure of the gas is added into the pressure gradient in this relation, it effectively increases the temperature gradient required for startup, hence delaying startup. As noted above, the effects of mass or adverse tilt compound this problem and the "delay" may become permanent: the device may not start.

In modeling the increased temperature requirement of a system with gas, the required temperature gradient is calculated based on the amount of gas and the temperature of the fluid. Startup is then delayed until this temperature differential across the wick is achieved. Once startup has been allowed, FLUINT calculates the thermodynamic impact of NCGs automatically.

Adverse Tilt

Similar to the presence of gas, the effect of adverse tilt on the system is an increased temperature gradient required for startup of the LHP. In the case of tilt, it can be seen from the following equation that dT is directly proportional to the height (h).

$$dT \approx \frac{r\ g\ h}{dp/dT}$$

Where: ρ = density

g = acceleration due to gravity

h = height

In addition to the detailed start-up simulations, estimates were made with respect to the relative importance of gas and adverse tilt. The conclusion was that neither effect can be neglected: adverse tilts of 3m cause about the same order of magnitude problem as would end-of-life gas generation (estimated based on a heat pipe rule-of-thumb of 2.0e-7 gmol per gram of charge per year of operation).

Evaporator Mass

The effect of mass on the evaporator during the transient startup of the LHP is to decrease the amount of heat that flows into the evaporator. Assuming a constant heat load and a constant specific heat, it can be seen from the transient conduction equation that any increase in mass will require and increase in time to achieve a given temperature difference:

$$T - T_{init} = \frac{Q\ t}{MC_p}$$

Where: Q = applied heat load

t = time

M = mass

Cp = specific heat of mass

A device might start reliably (even with a given amount of NCG and adverse tilt) with a certain power input (say 10W). However, attaching enough mass to the evaporator will decrease the effective power, and at some point start-up reliability will be lost. Analytically, an LHP would seem to always start (at any mass) as long as gas is not present, no adverse tilt is applied, and the vapor grooves clear immediately (zero incipient superheat). However, if any of those three conditions (gas, tilt, incipient superheat) exist, then there appears to exist a finite amount of mass past which start-up will *not* occur.

CORRELATION OF ANALYTICAL METHODS

Parallel to the development of these analytical methods, both Dynatherm (reference 5) and Swales Aerospace conducted independent test programs to evaluate the effects of mass, tilt, and noncondensible gas on the startup of LHPs. Both contractors freely provided data for use in the development and correlation of the analytical models and methods discussed herein.

The correlation effort for these models was comprised of two phases. First a top level correlation of the system level parameters was performed. These parameters, (such as losses to the environment, wick back-conduction, interface conductances at the condenser and the evaporator mass) were correlated to steady test results at various power levels using the automated data correlation techniques in SINDA/FLUINT.

The (ongoing) second phase of the correlation was more difficult since the remaining parameters had high uncertainties (such as internal film convection coefficients) or where stochastic (such as incipient superheat levels): they cannot be as easily measured or quantified through test. Rather, the parameters were qualitatively correlated based on trends of the transient behavior during LHP start-up. If a particular test proved successful, for example, the conditions under which this success could be duplicated in simulations were explored.

DISCUSSION OF RESULTS

The models were successful in predicting various, rather complicated, LHP startup phenomena. Although this study is only interested in LHP start from a flooded groove condition, the models were also verified to properly simulate startup from a non-flooded condition. Only flooded vapor groove start-ups will be discussed herein.

A typical predicted LHP startup with a 10 watt load on the evaporator, a 10°C condenser sink and ambient environment is shown in Figure 5. This particular case had no mass attached to the evaporator, no adverse tilt (horizontal) and no gas added to the working fluid. Since the amount of vapor groove superheat (temperature rise of the evaporator above the compensation chamber) is stochastic, the value was randomly set to 2.4°C for this case and subsequent cases discussed herein. Average super heat values vary between loop designs. In the two test programs supporting this effort, an average superheat value 2.4°C was observed on one LHP while 6.5°C was observed on the other. The selection of the lower superheat was simply for the purposes of expediting the computational time for the analyses.

EVAPORATOR MASS

Figures 6 show the predicted effect on startup of increasing the evaporator mass to 24 lbs.. It can be seen from this plot that the time required to develop the temperature differential required for startup (the 2.4 °C superheat) took three times longer respectively than the zero mass case (72 minutes versus 22 minutes). In this particular case due to the compensation chamber dropping in temperature, the payload showed no significant temperature rise attributable to the additional mass.

ADVERSE ELEVATION

When adverse tilt is added to the system not only does the loop need to provide additional subcooling to offset the static pressure head but, as discussed earlier, the minimum power to start the loop will increase significantly. This is evident by the predicted 10 watt startup failure depicted in Figure 7 with no mass or gas but with 25 inches of adverse tilt. After approximately 22 minutes, the loop attempted to start (note the temporary increase in the vapor line temperature) but failed due to the dominance of wick back conduction. In this particular case, of the 10 watts applied to the evaporator only 6.8 watts went into the evaporator body (due to convective losses) of which 31% when into wick back conduction.

For comparison, the zero tilt case shown in Figure 5 had only 5% of the evaporator heat consumed by wick back conduction. Reference back to Figure 2 for 50W/K wick conductance shows that 10 watts is insufficient in offsetting the wick back conduction and subcooling associated with 25 inches of adverse tilt.

Figure 5: Baseline Startup with 10 Watt Load, 10°C Sink
No mass, no tilt, no gas

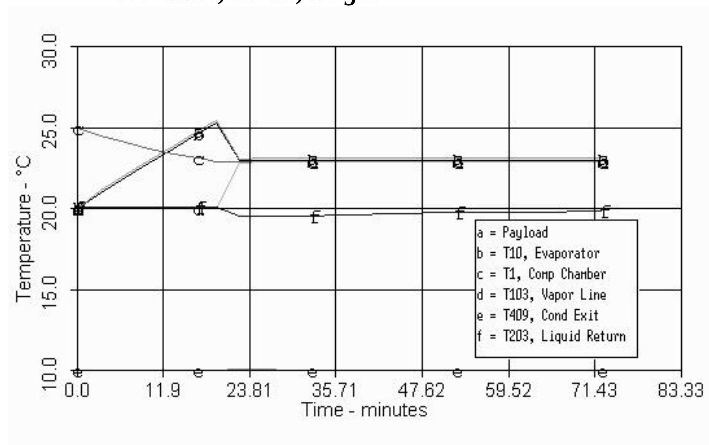


Figure 6: Startup with 10 Watt Load
24 lbs mass, no tilt, no gas

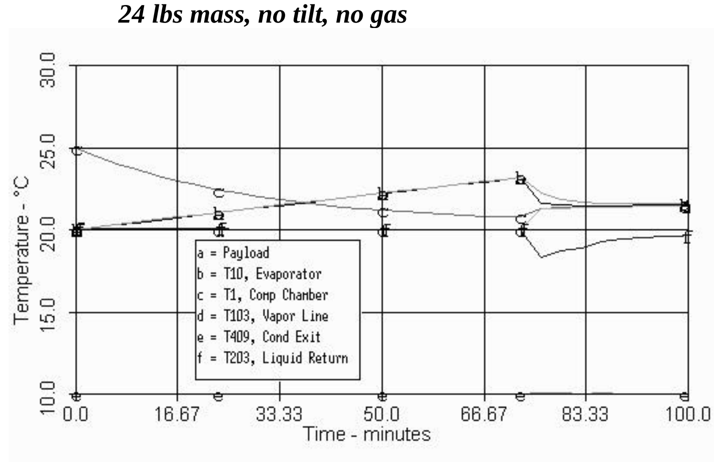
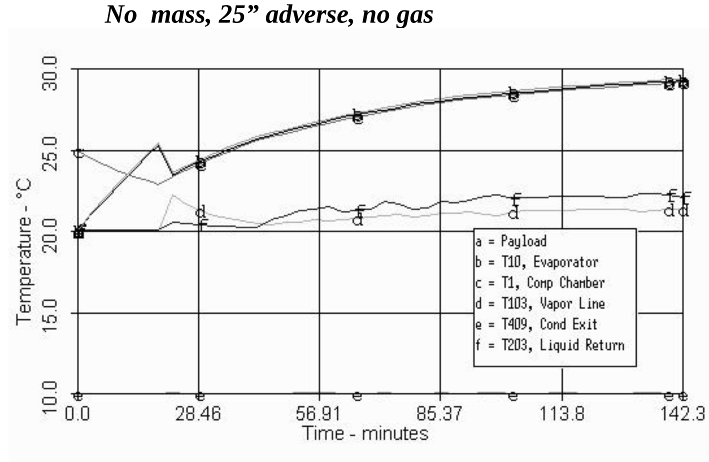


Figure 7: Startup with 10 Watt Load
No mass, 25" adverse, no gas



Additional analyses showed that a minimum of 50 watts is required to start the LHP in this adverse condition. Figure 8 shows the predicted transient response of the successful tilt startup with a 50 watt load while Figure 9 shows the heat flow summary from the analysis. The large amount of subcooling (20°C) required to overcome the adverse tilt is quite evident when comparing back to the no mass or gas, zero tilt case shown in Figure 5 which had roughly 3°C of subcooling at the evaporator. The increased subcooling requirement related to the adverse tilt bumps the operational temperature of the loop up to 40°C.

NONCONDENSIBLE GAS

For this system, 3.8×10^{-3} gm-moles was calculated for the expected end-of-life noncondensible gas generation. Hydrogen was defined as the gas in the model. Applying this amount of gas to the model with 24 lbs. of mass and no adverse tilt, the predicted system response showed a failed startup at 10 watts as evident in Figure 10. Note that the time required to achieve the required temperature differential across the wick increased from 72 minutes (with 24 lbs. mass) to 108 minutes, a factor of 50% (combined mass and gas effects resulted in a 300% increase in time). Increasing the power to 100 watts did result in a successful startup.

WICK BACK CONDUCTION

Heat flowing backwards through the wick is normally the dominant source of problems for start-up, whether it causes boiling in the core rather than the grooves, or just raises the threshold for sustained operation in the presence of adverse tilt or NCG. A lower conductance wick not only performs better at steady state, it is also more likely to start successfully. These facts are well know to LHP manufacturers such as Swales and Dynatherm, who continue to seek low conductivity materials with adequate properties. Nickel, which otherwise is a very suitable wick material, unfortunately has a rather high thermal conductivity. All testing reported in this paper was performed using nickel wicks in ammonia LHPs.

Normally good success has resulted from using fixed values (correlated from test data or measured in separate tests) for wet wick conductance. Unfortunately, this has not been the case in this program: some tests could not be replicated analytically without seemingly drastic changes to the wet wick conductivity. Figure 11 depicts the test profile of such a case with a 25 watt heat load, 24lbs. mass, and 34” of adverse tilt. In this test the system starts and continues operation for 1.5 hours on less than 3°C of subcooling at the evaporator. Theoretically, this system should not have started based on the minimum subcooling requirement to offset the adverse tilt. At the time of this writing, the cause of this discrepancy is unknown. Suspects include a missing phenomena in the prediction of wick back conduction including hysteresis[2], a failure to truly achieve steady state conditions in certain tests, or a problem in the thermal connection (either analytically or experimentally) between the compensation chamber and the ambient and/or between the vapor line and the ambient.

[2] In a paper to be presented this summer (Investigation of the Temperature Hysteresis Phenomenon of a Loop Heat Pipe) at the National Heat Transfer Conference, Kaya and Ku have investigated changes to the reverse conductance in LHPs, and have attributed it to voids in the secondary wick. This phenomena seems unlikely to be the explanation in this particular case since the reverse conduction appears to be occasionally much *lower* than what “solid” conduction predicts, whereas the hysteresis effect noted by Kaya and Ku relates to an *augmentation* of the reverse conduction.

Figure 8: Startup with 50 Watt Load
No mass, 25” adverse, no gas

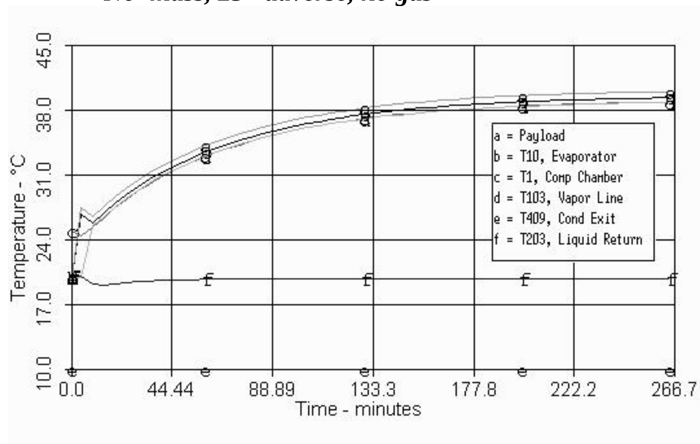


Figure 9: Startup with 50 Watt Load, Heat Flow Summary
No mass, 25” adverse, no gas

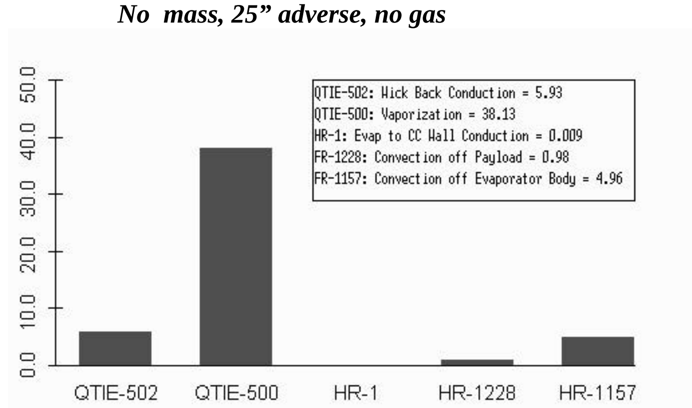
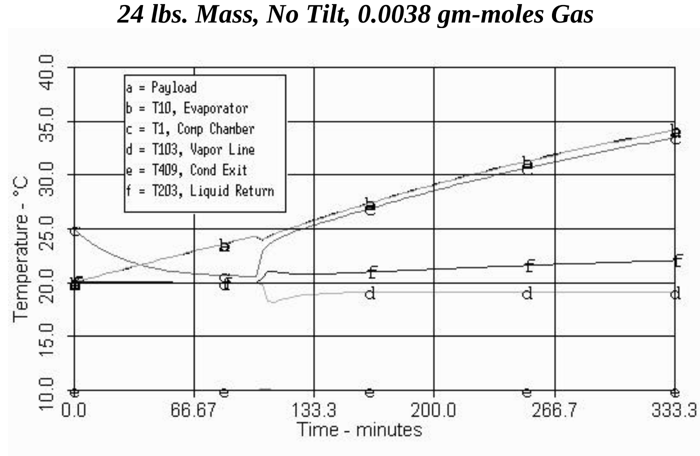


Figure 10: Startup with 10W Load, Mass and Gas
24 lbs. Mass, No Tilt, 0.0038 gm-moles Gas



Although more investigation is required, it appears that the persistence of superheated liquid within the core of the evaporator is a likely explanation for these unusual test results. If the core liquid never boils and no bubbles enter it, then energy back-conducting through the wick must not only flow radially, it must also flow axially through the superheated liquid, helping to explain the order of magnitude reduction in back-conduction evidenced in these tests. This explanation is thus far consistent with the unique (preheated compensation chamber) start-up methods applied, and with tilting and heating applied to the evaporator unit in attempts to determine the nature of this phenomena.

However, tests experiencing this behavior are being excluded from consideration in part because of the lack of a full explanation and in part because they experience little if any start-up problems because of the low back-conduction.

INCIPIENT SUPERHEAT ISSUES

The stochastic nature of superheat made exact comparisons difficult: some test runs have succeeded (Figure 11) while repeated tests under identical conditions have failed (Figure 12). Both these tests had a heat load of 25 watts, 24 lbs. evaporator mass, 34 inches of adverse tilt, no gas, and 10°C condenser sink. In the case of reverse flow as depicted in Figure 12, the models were able to duplicate the results by defining the core superheat level lower than the vapor groove superheat. In this particular test, stable forward flow was not established until the heat load was increased to 100 watts. The “chugging” evident at 50W by the wide oscillation in the vapor line temperature was most likely a result of sufficient condensation in the vapor line and insufficient flow to maintain the vapor front in the condenser.

Eventually the safe envelope of start-up must be predicted with adequate but not excessive conservatism, and this will require statistical studies along with empirical. For now, the degree of superheat was taken from test data and applied to the corresponding analytic model to remove this complication.

The “typical” degree of superheat experienced varies from unit to unit based not only on design but also manufacturing, cleaning, and charging procedures. One of the two units tested, for example, typically experienced 2 to 3°C superheat, while another experienced about 10°C.[3] It is not clear how the degree of superheat can be predicted for any one unit in advance, although superheats higher than 12°C have not been witnessed (to the author’s knowledge) in ammonia systems.

Unfortunately, one cannot say definitively that assuming a high degree of superheat (such as 12°C) is always conservative: as long as boiling initiates in the grooves and not in the core, superheat is not necessarily bad when there is mass attached to the evaporator. In such cases, superheat transiently raises the amount of power input into the evaporator, in some cases several times that applied to the attached mass.

It has been observed analytically that a loop may initially start due to the high sensible heat associated with incipient superheat, but well into operation the system can fail due to dominance of back conduction after all the sensible heat has been removed from the system[4]. Figure 13 shows the predicted transient heat load available for vaporization for three similar cases with and without evaporator mass. In these plots the heat applied

Figure 11: Unreplicated Successful LHP Startup (Test Data)

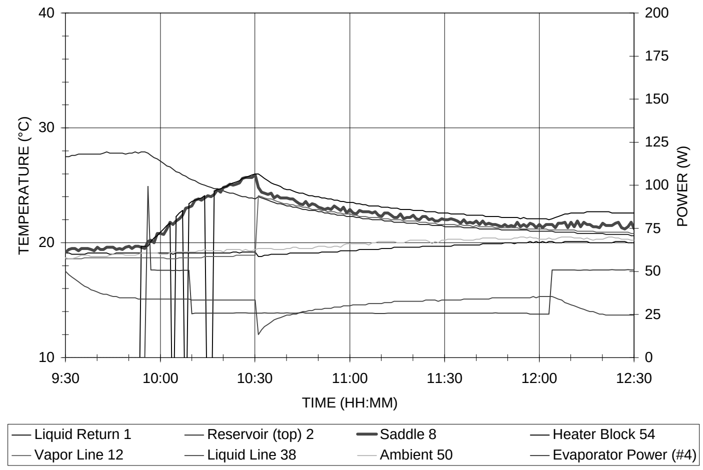


Figure 12: Reverse Flow at LHP Startup (Test Data)

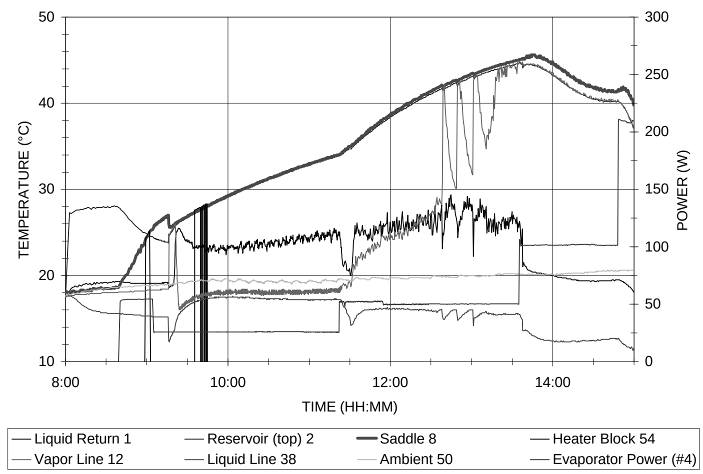
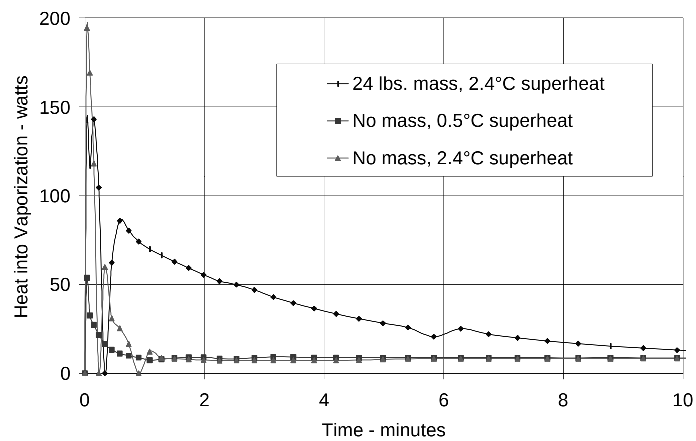


Figure 13: Effects of Superheat and Sensible Heating



to the evaporator was 10 watts and the level of superheat specified for boiling was 0.5°C (one of the no mass cases) and 2.4°C (no mass and mass case). The 2.4°C superheat plots are extracted from the results of the runs previously depicted in Figures 5 and 6. Although the peak power is greatest for the no mass case with superheat, the transient load for both no mass cases drops below 10 watts within one minute of startup. Whereas in the case with mass and superheat a power level well above 10 watts is maintained for over 10 minutes after the flow is initiated.

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[3] As a reminder, superheat is typically only experienced in LHPs when the compensation chamber has been heated, or when coming out of the diode mode.

[4] From the system integration perspective, startup is defined when the vapor front has *stabilized* in the condenser and long term operation can be maintained. In this paper, startup is defined as initiating forward flow in the evaporator.

In other words, the unit is “kick-started” into a high power mode, with power decreasing thereafter. This causes the temperature of the attached mass to rise (prior to boiling), then to drop suddenly after boiling begins, and then to rise again slowly as the power drops along with the overall loop conductance. In other words, without superheat the input power on the LHP must grow from zero to the final value, whereas with superheat the power starts high and drops to the final value thereby avoiding the dangerous low power region of operation.

CONCLUSION

An analytic methodology has been developed for predicting the safe envelope of start-up for LHPs, and it is currently being correlated with an extensive test program. Once this effort is complete, design guidelines are expected to be produced including perhaps sizing rules for mitigation approaches such as Peltier elements.

As of the deadline for this paper, neither the test program nor the analytic comparisons and correlations had been completed. As such, they (along with important conclusions such as design guidelines) must be presented in a later paper.

However, the combined test and analytic program has been highly successful in showing the relative importance of the phenomena involved, in interpreting test data, and in dynamically adjusting the test matrix as needed to locate the envelope of safe operation. Testing with large masses is time consuming and it can be difficult even to decide whether a particular start-up was successful or not. Concurrent analysis helps to decide how long a test must be run before steady state has truly been achieved.

Testing in ambient conditions (to avoid expensive vacuum testing) has a large effect on the results. Although the concurrent analysis similarly takes into account interactions with the environment, future work should perhaps consider a limited vacuum test program as a further check on the analysis.

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CONTACT

For more information on the modeling tools (SINDA/FLUINT and Sinaps*Plus*) or the development of analytical methods for modeling two-phase heat transport loops please contact Cullimore & Ring Technologies through their webpage at www.crtech.com or via email to info@crtech.com.

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