Loop Heat Pipe Radiator Trade Study for the 300-550 K Temperature Range

William G. Anderson[1] and Walter Bienert[2]

[1]Advanced Cooling Technologies, 1046 New Holland Ave., Lancaster, PA 17601
[2]Consultant, 401 Valley Court Road, Lutherville, MD 21093
717-295-6059, Bill.Anderson@1-ACT.com

Abstract. A loop heat pipe (LHP) radiator trade study has been completed for radiators in the 300-550 K temperature range. Initially, a thorough component level study was completed to determine LHP operating properties over the temperature range, particularly at the high end where the Merit Number starts to fall off. Water was found to be the optimum fluid for the high end, while ammonia was a better working fluid for temperatures below 350 K. A trade study was then conducted that varied condenser O.D., temperature, panel area, fin width, fin thickness, and fin thermal conductivity. A comparison with comparable heat pipe radiators shows that the specific power of LHP radiators can be as much as 50 percent higher at temperatures above 500 K. This is offset by the fact that LHP radiator technology is much less mature.

INTRODUCTION

NASA is interested in Brayton cycle converters for Nuclear electric propulsion for missions including the proposed Jupiter Icy Moon Orbital (JIMO) mission (Mason, 2003, Siamidis et al., 2004). A radiator is required to dissipate the waste heat generated during the thermal-to-electric conversion process. The radiated power is on the order of 200 to 800 kW_{th}, with radiator temperatures in the 300 to 550 K range. Mason (2003) discusses the overall system concept. Siamidis et al. (2004) describe a typical radiator design for a Brayton system. A pumped sodium-potassium (NaK) secondary loop is used to transfer waste heat from the power converters to the heat pipe radiator. Heat is transferred from the NaK to the heat pipes by inserting the evaporator sections into the NaK duct channel. The radiator panel consists of a series of heat pipes located between two high conductivity fins. The heat pipes transfer the heat to the fins, which radiate the waste heat to space.

A similar radiator could be fabricated with Loop Heat Pipes replacing the heat pipes. The advantages of LHPs include self-priming, and that they can be ground tested in any orientation. LHP spacecraft radiators are being considered at lower temperatures, with propylene or ammonia as the working fluid (Baker et al., 2004). While water LHPs have been proposed for spacecraft radiators (Baumann and Rawal, 2001), water LHPs still require significant development.

This trade study examined LHP radiators over the 300 to 550 K temperature range. This trade study differs from most previous radiator trade studies in that it starts with a detailed design of the LHP. The limits from this design are fed into the trade study, and solutions that are outside of the LHP limits are rejected. A companion paper (Anderson and Stern, 2005) examines similar heat pipe radiators.

RADIATOR PANEL DESIGN

The overall radiator panel layout is shown in Figure 1. The panel has the following: (1) A series of titanium/water LHPs to transfer heat from the secondary fluid to the radiator panel, (2) High conductivity graphite foam saddles to form an interface between the circular heat pipe and the flat fin, (3) High conductivity Graphite Fiber Reinforced Composites (GFRC) fins, and (4) Aluminum honeycomb to provide stiffness to the structure.

The LHP configuration assumes that a series of round LHP condensers (with integral saddles) are embedded in the radiator panel to distribute heat. The thermally active part of the radiator panel uses high-temperature-capable Graphite Fiber Reinforced Composites (GFRC's). This is a polymer matrix material, which we feel represents a technologically better developed alternative to carbon-carbon panels. Carbon-carbon has not been commonly used as a self-supporting structure in lightweight spacecraft panels because in thin cross-sections it is rather brittle, but GFRC facesheets with aluminum core is very commonly used. The main difference from conventional panels is that a high-temperature resin matrix is required to allow the panel to operate at temperatures up to 550 K. Further explanation for the panel design and materials selection can be found in the companion paper (Anderson and Stern, 2005).

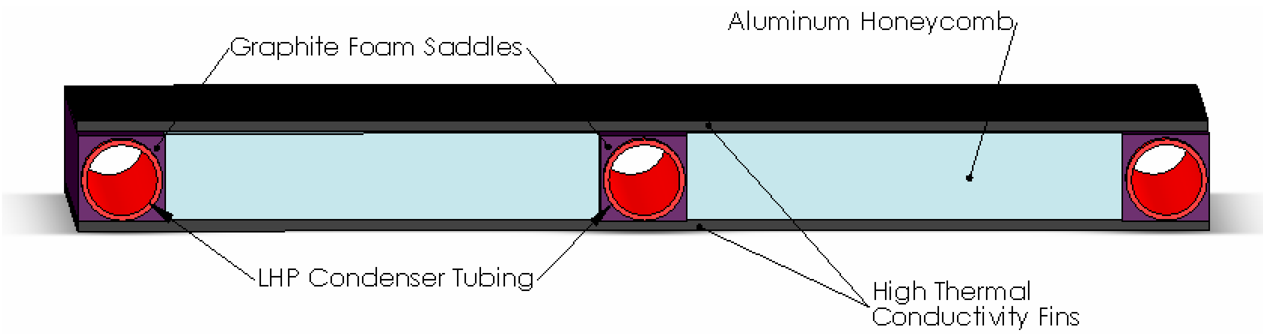

FIGURE 1. Cutaway Section Through The Panel.

LOOP HEAT PIPE DESIGN

During the LHP design and trade study, the evaporator dimensions and properties were fixed at the current dimensions for other aerospace LHPs: 22 mm O.D. by 300 mm long.(0.867" O.D. by 12" long). Similarly, it was assumed that a wick in the appropriate material could be developed with properties similar to current LHP Wicks: 1.2 micron pore size, with a permeability of 1.2×10^{-14} m^2. In contrast to heat pipe designs, the LHP performance is less closely coupled to the evaporator dimensions. In order to include current as well as future technology in the study, two parallel sets of analysis were performed: one based on the "standard" evaporator with a 1200 W limit and another one with an "advanced" evaporator with a 5000 W limit.

Compensation Chamber

In contrast to a heat pipe, a loop heat pipe does not have a wick throughout the system. Because of this, the fluid inventory and volumes must be controlled to insure that the LHP is self-priming, with liquid always in contact with the evaporator. A compensation chamber (connected to the evaporator with a secondary wick) is used to supply liquid to the evaporator, while accommodating the changes in liquid volume in the remainder of the heat pipe.

The LHP detailed design and trade studies assumed a spherical compensation chamber, with the wall thickness set by the pressure and materials properties, as discussed previously. The standard assumptions for sizing the compensation chamber were made: (1) Compensation Chamber is 10% full at the minimum temperature (300 K for water, 250 K for ammonia), and (2) Compensation Chamber is 90% full at the Maximum temperature. For the calculations, the maximum temperature was chosen to be the highest temperature in each of the 5 ranges, i.e., 550 K for the 525 K design.

For a given LHP, the Compensation Chamber volume increases as the temperature increases, since it must accommodate a greater change in liquid density. It also increases as the condenser length increases (for a given diameter), since it must accommodate the entire condenser volume at the maximum operating temperature.

Working Fluids

The potential working fluids with titanium are ammonia and water. Both fluids are compatible with titanium at lower temperatures; life tests are currently underway with water to demonstrate compatibility at the higher temperatures (Anderson and Stern, 2005).

Water is clearly the preferred working fluid at the higher end of the temperature range. It becomes a bit problematic at 375 K because of the low vapor pressure and high vapor velocities. It cannot be used at 325 K where ammonia is the preferred choice. In order to cover the transition range more fully, both fluids were also evaluated at the intermediate temperature of 350 K.

Detailed LHP designs were developed for ammonia and water LHPs, using the standard Advanced Cooling Technologies LHP model. These LHP designs were used to roughly size the LHPs for the trade study, and to determine the cross-over point between water and ammonia. Typical results are shown in Figure 2 for the standard 1200 W power. The ammonia LHP has a lower mass for temperatures below 350 K.

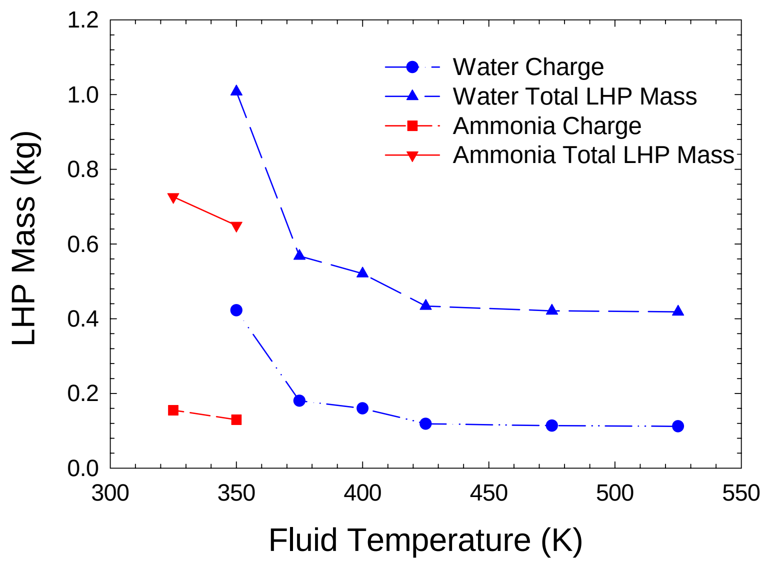

FIGURE 2. LHP Standard Evaporator, Fluid and Total LHP Mass for the Standard Evaporator with 1200 W Power.

LOOP HEAT PIPE TRADE STUDY

Depending on the configuration, portions of the radiator can operate at temperatures from 300 K (27°C) to 550 K (277 °C). Note that an individual heat pipe carries roughly a kW, while the radiator power ranges between 200 and 800 kW, implying somewhere in the range of 100 and 800 individual heat LHPs. In other words, each LHP will carry only a small fraction of the total radiator panel power. This means that we can design an LHP (and associated fins) in isolation from the rest of the system. The temperature range has been broken into 5 ranges, and designs developed for the middle temperature in each range: 325, 375, 425, 475, and 525 K. Assumptions are as follows:

- One dimensional heat transfer from the LHP into the radiator panel
- One dimensional heat transfer in the radiator panel
- Temperatures above 350 K: Titanium, Water working fluid
- At lower temperatures: Titanium, Ammonia working fluid
- Emissivity of the radiator panel is 0.9
- Each LHP and associated radiator fins are independent of all of the others
- Fin thermal conductivity is 2 W/m K out of the high conductivity plane. Fin density is 1.75 gm/cm^3.
- Aluminum honeycomb with a density of 0.05 g/cm^3.
- Foam saddles are oriented with the high thermal conductivity in the heat flow direction.

For the fin thermal conductivity, the fiber orientation must be considered. Unidirectional fibers oriented perpendicular to the heat pipe have the advantage of having a positive C.T.E. along the heat pipe, but are weaker structurally. Isotropic fiber orientation gives a greater strength, but has roughly one-half of the thermal conductivity. For the trade study, we assumed the ability, through the appropriate choice of fibers and ply orientations, to customize the thermal conductivity in the direction perpendicular to the heat pipe. Three different thermal conductivities were examined: 300, 600 and 1000 W/m-K.

Temperature Drops: The following temperature drops occur from the NaK fluid in the secondary loop to the radiator fin:

1. Temperature drop from the secondary loop NaK to the outside of the LHP evaporator (not considered in this study due to the very high thermal conductivity of NaK, about 25 W/m K at 500 K).
2. Temperature drop through the evaporator wall.
3. Temperature drop in the vapor (neglect).
4. Temperature drop through the condenser wall.
5. Temperature drop through the graphite saddle.
6. Temperature drop into the middle of the fin.
7. Temperature drop along the fin.

The trade study examined the design at 5 temperatures that span the range of the possible radiator designs: 325, 275, 425, 475, and 525 K.

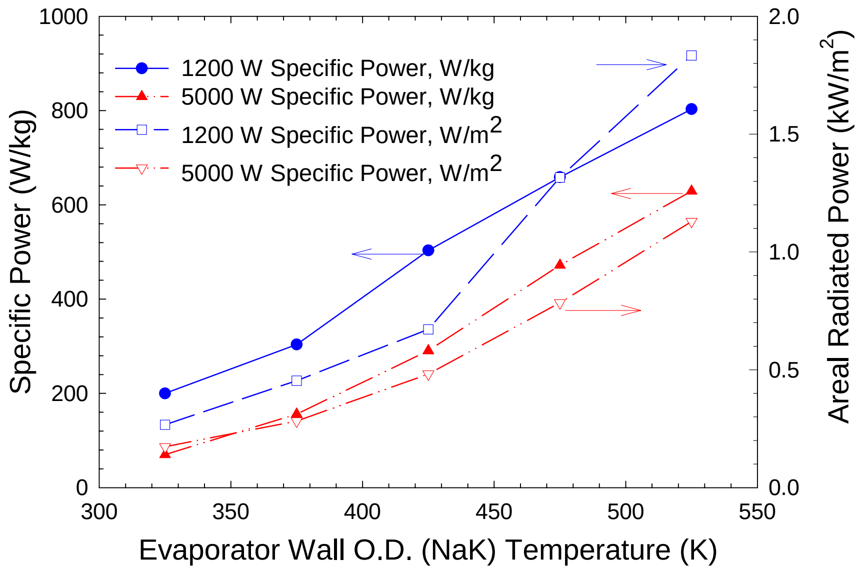

FIGURE 3. LHP Radiator Specific Power, Standard and Advanced Evaporators, 300 W/m K Fins.

LHP Trade Study Results

The radiator performance and mass are calculated while varying the following independent variables:

- Temperature: 550, 500, 450, 400, 350, 300 K
- Fin Thermal Conductivity: 300, 600, 1000 W/m K
- Fin Thickness: 0.01 to 0.1 in. by 0.01 in. (0.254 mm to 2.54 mm by 0.254 mm).
- Overhanging Fin Width: 0.5 to 6.0 inches by 0.5 in. (1.27 cm to 15.2 cm by 1.27 cm).
- Condenser O.D.: 1/8, 3/16, 1/4, 5/16, and 3/8 inch O.D (0.267, 0.425, 0.584, 0.743, 0.902 cm I.D.).
- LHP Power: 1200 and 5000 W

For each case, the temperature drop through the system for the fixed power was calculated, and the condenser length adjusted until the fixed power was radiated at the effective fin temperature. Once the condenser length was known,

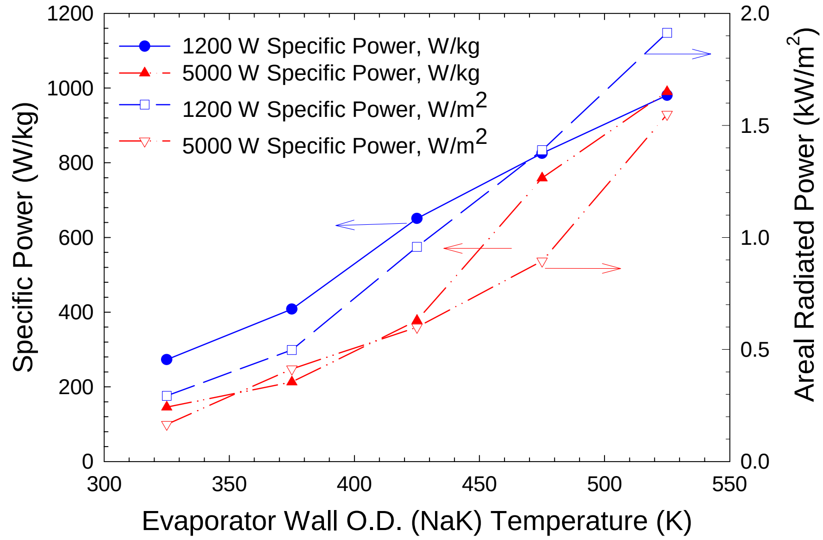

FIGURE 4. LHP Radiator Specific Power, Standard and Advanced Evaporators, 1000 W/m K Fins.

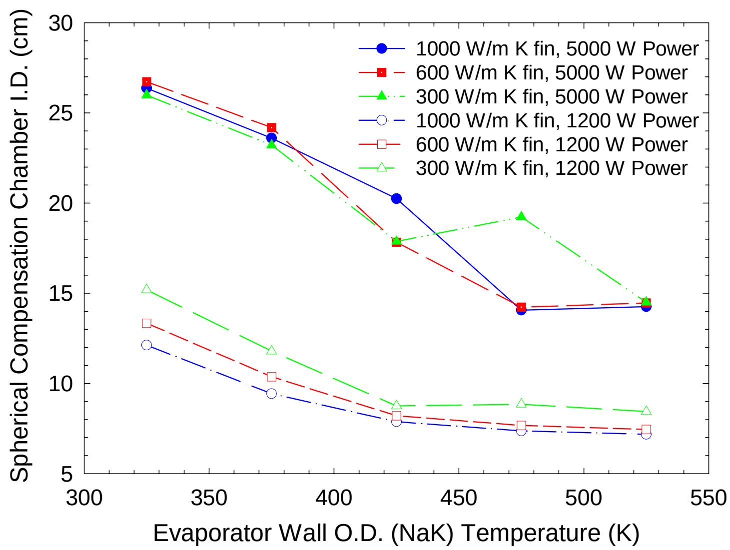

FIGURE 5. LHP Spherical Compensation Chamber Diameter. Standard and Advanced Powers, 300, 600, and 1000 W/m K Fins. The Advanced Power Designs Have A Much Larger Compensation Chamber, Due to the Longer Line Lengths.

the system dimensions and masses were calculated. The power and system mass were then used to find the specific power. During the trade study, the condenser length was checked, and any designs with too long of a length were rejected. The maximum allowable condenser length was the smaller of (1) 50 m, and (2) the maximum length set by the pressure drops in the LHP. The results were then sorted by specific power.

The results are shown in Tables 1 to 3, and Figures 2 to 5. Figures 3 and 4 show the best-case radiator specific power for both the standard and advanced evaporators with 300 and 1000 W/m K fins. For all cases, the 1200 W designs had a higher specific power (W/kg) when compared with the 5000 W designs. This occurs because the 5000 W designs require a longer condenser to reject the heat. The condenser diameter must be increased to compensate for the longer condenser and higher flow rates. In addition to the increased condenser volume, the compensation chamber and fluid charge must also be increased; see Figure 5.

Tables 1 – 3 show the maximum specific power for the 1200 W and the three fin thermal conductivities. The 5000 W results are not shown, since they were always worse than the 1200 W results. Note that the heat exchanger mass, and the mass of the required trace heaters (and the power consumption) are not included.

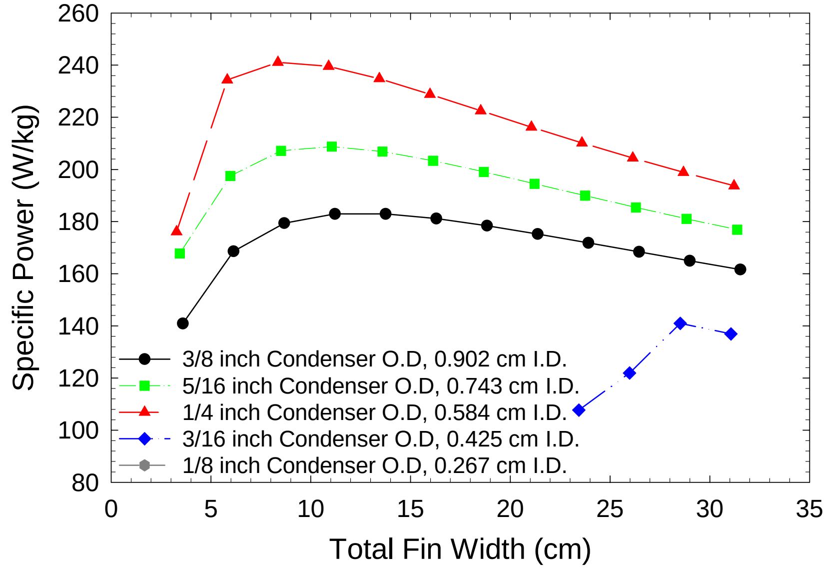

FIGURE 6. LHP Trade Study - Specific Power Versus Fin Width, 325 K, 600 W/m K Fin Thermal Conductivity.

TABLE 1. LHP Trade Results, 1200 W Evaporator, 300 W/m K Fins.

T_{NaK} K	Cond. O.D. in.	Fin Width cm	Fin Thick mm	Cond. Length m	T_{Root} K	Specific Power W/kg	Power/Area KW/m^2	CC ID cm	Fluid Charge kg
325	1/4	5.82	0.25	38.7	319.5	199.89	0.267	15.20	0.89
375	1/4	5.82	0.25	22.7	369.0	303.57	0.454	11.79	0.80
425	3/16	5.66	0.25	15.8	418.0	503.23	0.672	8.76	0.36
475	3/16	3.12	0.25	14.6	467.8	658.88	1.316	8.85	0.34
525	3/16	3.12	0.25	10.5	516.9	803.02	1.834	8.44	0.28

TABLE 2. LHP Trade Results, 1200 W Evaporator, 600 W/m K Fins.

T_{NaK} K	Cond. O.D. in.	Fin Width cm	Fin Thick mm	Cond. Length m	T_{Root} K	Specific Power W/kg	Power/Area KW/m^2	CC ID cm	Fluid Charge kg
325	1/4	8.36	0.25	25.6	319.1	241.06	0.280	13.34	0.62
375	1/4	8.36	0.25	15.1	368.4	362.30	0.476	10.38	0.57
425	3/16	5.66	0.25	12.7	417.4	590.11	0.833	8.21	0.31
475	3/16	5.66	0.25	8.9	466.3	751.85	1.196	7.67	0.25
525	3/16	5.66	0.25	6.5	514.8	899.60	1.637	7.45	0.21

TABLE 3. LHP Trade Results, 1200 W Evaporator, 1000 W/m K Fins.

T_{NaK} K	Cond. O.D. in.	Fin Width cm	Fin Thick mm	Cond. Length m	T_{Root} K	Specific Power W/kg	Power/Area KW/m^2	CC ID cm	Fluid Charge kg
325	1/4	10.90	0.25	18.8	318.8	272.90	0.293	12.13	0.47
375	1/4	10.90	0.25	11.1	367.8	408.25	0.498	9.43	0.45
425	3/16	5.66	0.25	11.1	417.0	650.88	0.958	7.88	0.28
475	3/16	5.66	0.25	7.6	465.6	825.19	1.390	7.37	0.23
525	3/16	5.66	0.25	5.5	513.9	980.88	1.913	7.18	0.19

The results discussed above are for the maximum specific power. It might be desirable to use a different fin spacing from an overall systems design. Figures 6 and 7 are typical graphs of specific power versus fin width. In general, the smallest LHP condenser gives the best results. The exception occurs when the heat pipe diameter is small enough that the maximum condenser length is a limiting factor. In this case, the smaller diameter curve is often incomplete. For example, the 1/8-inch diameter (0.267 cm I.D.) results in Figure 7 are much heavier than optimum designs with larger diameters. This occurs because the maximum allowable length of the 1/8-inch condenser is short enough that only LHPs with relatively thick, massive fins have a short enough condenser length. These thick fins increase the mass above the larger diameter condenser designs, which can use thinner fins.

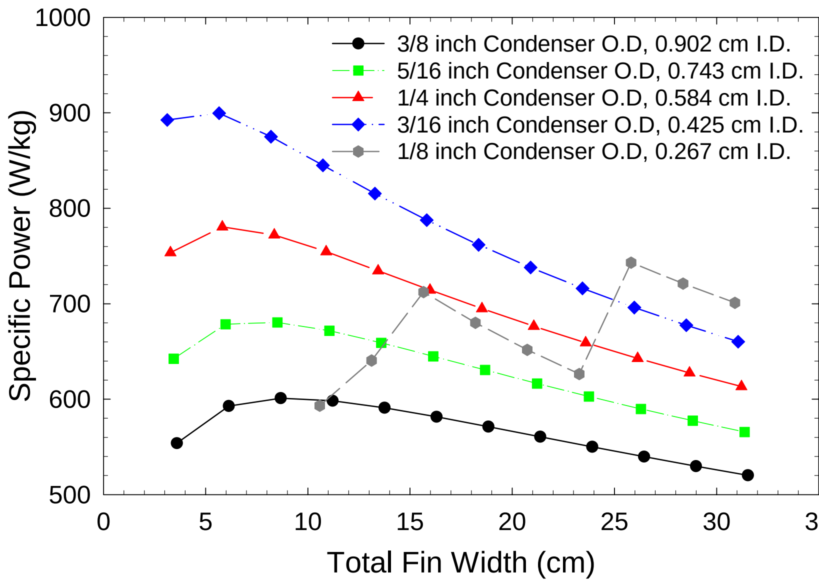

FIGURE 7. LHP Trade Study - Specific Power Versus Fin Width, 525 K, 600 W/m K Fin Thermal Conductivity.

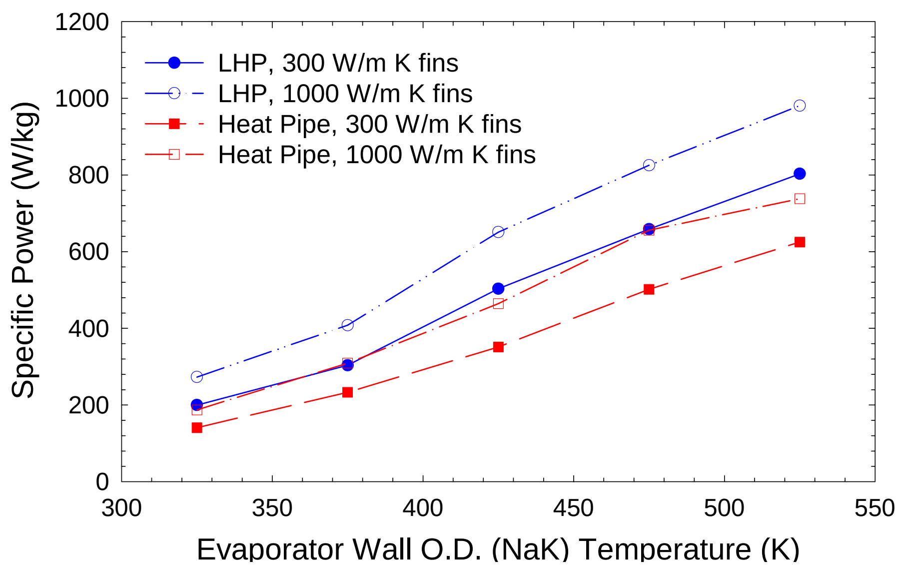

FIGURE 8. LHP Versus Heat Pipe Specific Power, 300 and 1000 W/m K Fins.

Loop Heat Pipe Versus Heat Pipe Radiators

There are a number of advantages to Loop Heat Pipes (LHPs), when compared with heat pipes: (1) Wick only located in the evaporator, (2) Can use smaller pore sizes, allowing higher pumping capabilities, (3) Designed to be self priming, (4) Give the advantages of arteries while remaining tolerant to non condensable gas, and (5) Can be ground tested in any orientation. Heat pipes may be able to tolerate a one-inch adverse elevation, while water LHPs at lower temperatures have operated with more than 10 meters adverse elevation.

A companion trade study calculated the specific powers for heat pipe radiators under comparable conditions (Anderson and Stern, 2005). Figure 8 compares the LHP and Heat Pipe specific powers for the 300 and 1000 W/m K fins, respectively. In both cases, the LHP has a significantly higher specific power, as much as 50 percent higher at for temperatures above 500 K. The LHP radiator with 300 W/m K fins has a specific power that is comparable to the heat pipe radiator with 1000 W/m K fins.

There are a number of potential disadvantages: (1) Large Compensation Chamber – Packaging, (2) Longer Lengths of Serpentine Tubing, (3) More complicated to integrate in panel versus straight heat pipes, and (4) Freeze/Thaw requires trace heating - not included in the masses above.

CONCLUSIONS

A series of trade studies were conducted for flight radiator designs operating at temperatures between 300 and 550 K. The trade study started with a detailed design of the LHP, which determined the maximum allowable condenser length. The trade studies varied fin thermal conductivity, fin thickness, condenser spacing, condenser diameter, power, and temperature. Designs with powers higher than the LHP could carry were rejected. Two powers were examined for a standard evaporator size: 1200 and 5000 W. The higher power LHPs had a lower specific power, due to the larger compensation chamber required.

LHP radiators have significantly higher specific power than heat pipe radiators, and should be developed for inclusion in future radiators. High-temperature water LHPs should be developed for inclusion in future radiators. While a heat pipe radiator could be built with current technology, LHPs require significant additional research and development. For example, a suitable wick would need to be developed, freeze/thaw would need to be examined, and packaging of the compensation chamber would need to be studied in greater detail.

ACKNOWLEDGMENTS

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