Optimization of Various Thermal Management Techniques for High Thermal Loads

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ABSTRACT

The increasing thermal demands for spacecraft require the development of new technologies. To efficiently allocate resources to research and development efforts, a tool was developed that can be used to investigate the impact of new thermal technologies on spacecraft mass. The thermal tool is presented here as part of a mass study of three different types of thermal management technologies. The heat pipe radiator is recommended for a thermal load below 700W if the radiator is able to radiate to space from both sides. A heat pipe radiator radiating to space from a single side should be considered for loads at or less than 100W given the linear data trend. Loop heat pipe and single phase pumped looped radiators should be considered for loads exceeding 700W as their configuration allows for deployable radiators. The results indicate that for assumptions used in the paper, technologies associated with the radiator such as alternate materials are desirable above 700W whereas improvements should focus on the pipe and face sheet materials to reduce radiator mass.

INTRODUCTION

Satellites have a need for effective and cost saving thermal management. With the large cost per mass associated with a satellite launch it is advantageous to find a minimum weight solution for thermal management. Presented is a trade study to find the most effective thermal system for specific heat loads. Passive and active systems were analyzed in the form of heat pipes, loop heat pipes and single phase pumped loop systems. Each system has been proven in space. The single phase pumped loop system was used on the space shuttle and International Space Station. The heat pipe and loop heat pipe systems have been utilized on numerous satellites. With a known reliability of these systems, the focus was placed on finding a low mass solution while maintaining heat rejection.

BACKGROUND

The heat rejection needs for a satellite vary greatly and are dependent on operating temperature, orbit and components. Thermal loads for small satellites can range from tens to hundreds of watts. The loads investigated for this study were over a broader range due to the ever increasing satellites thermal needs.

The analytical approach to the heat pipe radiator fin design was based on Chang’s work on “Optimization of a Heat Pipe Radiator Design”, AIAA 1984-1718. In Chang’s paper the heat pipe spacing and face sheet thickness were varied to find the lightest radiator per kilowatt of radiated energy. The same equations and approach were used to optimize the heat pipe radiator in this study.

METHODOLOGY

For this study a standard radiator design was used. It was composed of an aluminum honeycomb core which was used to support the aluminum pipes in the radiator and was effectively non-conductive. Aluminum facesheets were bonded to the top and bottom of the honeycomb core. Lastly, an Optical Solar Reflector (OSR) was bonded to both facesheets. The layout of the radiator is shown in Figure 1.
The properties used for each component are listed in Table 1.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Density (kg/m$^3$)</th>
<th>Conductivity (W/m-K)</th>
<th>Thickness (m)</th>
<th>Emissivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum Facesheet</td>
<td>2700</td>
<td>167</td>
<td>Varied</td>
<td>0.8</td>
</tr>
<tr>
<td>Aluminum Pipe</td>
<td>2700</td>
<td>167</td>
<td>0.001 (LHP,SPP,L)</td>
<td>N/A</td>
</tr>
<tr>
<td>Adhesive</td>
<td>0.1463 (kg/m$^3$)</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Aluminum Honeycomb</td>
<td>36.9</td>
<td>N/A</td>
<td>Varied</td>
<td>N/A</td>
</tr>
<tr>
<td>OSR</td>
<td>0.4879 (kg/m$^3$)</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

The working fluid used was ammonia with the properties listed in Table 2.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Specific Heat (J/kg-K)</th>
<th>Dynamic Viscosity (N-s/m$^2$)</th>
<th>Density (kg/m$^3$)</th>
<th>Thermal Conductivity (W/m-K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ammonia</td>
<td>5000</td>
<td>1.24E-04</td>
<td>577</td>
<td>0.477</td>
</tr>
</tbody>
</table>

It was assumed that the fluid inlet temperature was a constant 323 K and the radiator was rejecting to a sink temperature of 4 K from both sides.

**Heat Pipe Model**

The heat pipes were assumed isothermal at 323°K and used an aluminum wick with 96% porosity, an effective pore radius of 0.023*10$^{-2}$m and permeability of 3.8*10$^{-3}$m$^2$. The heat pipe (HP) model calculates the wick thickness necessary to achieve the minimum summation of the liquid and vapor pressure drop for a given range of diameters, helping to ensure the capillary limit is met. The capillary limit is given by the maximum capillary pressure (Equation (1)) which is a function of the effective porous radius of the wicking materials ($r_e$) and the surface tension of the working fluid ($\sigma_L$). The capillary pressure must exceed the pressure loses from the liquid and vapor movement to sustain the cycle. The program concurrently calculated the pipe wall thickness, using thin wall pressure vessel assumption with a factor of safety of two against yielding.

The program then calculated a mass optimized heat pipe per length that met the boiling limit (Equation (2)). The radius of nucleation ($r_n$) was assumed to be 1.1*10$^{-7}$m. This was found by running the program for Savage’s, “ESTEC Heat Pipe Experiment on SPAS-01”, using their limiting flux and solving for the radius of nucleation. With the mass optimized heat pipe per length found that met the boiling limit, the diameter of the heat pipe was defined. The adiabatic section was assumed to have a length of 1m. The maximum condenser length was then solved for by increasing the pipe length until the capillary limit was met.

The lower length limit of the condenser was found by solving Equation (3) for the necessary length to remove the required heat load. For the coefficient of convection, the change in temperature was assumed to be two degrees and that the pipe operated under a 9.81m/s$^2$ acceleration. Using the condenser’s lower length limit the fins were optimized via Chang’s, “Optimization of a Heat Pipe Radiator” equations (see Heat Pipe Model-Radiator Fins). Within the fin optimization the heat pipe length was increased as necessary to radiate the required thermal load with the calculated fin efficiency. The heat pipe was not allowed to become longer than defined by the upper bound with the capillary limit. The heat pipe radiator has been designed for a given load.

To find the optimum number of heat pipes for the radiator, the thermal load was divided by two to twenty. The program was run again for each load with the denominator being the necessary number of heat pipes to meet the global thermal requirement. These heat pipe configurations were then compared to find the mass optimized radiator for the thermal load.

**Heat Pipe Model-Other Limits**

The sonic limit is a heat flux limitation, (Equation (4)) which occurs when the vapor reaches a sonic speed and...
a change in pressure does not accelerate the fluid. The sonic limit is a function of the latent heat of vaporization of the working fluid ($H_{vp}$), the density of the working fluid ($\rho_v$) and the vapor pressure ($P_v$). The viscous limit (Equation (5))2 is also a heat flux limitation, that occurs when the viscosity of the fluid is not overcome. The radius of the vapor pathway is $r_v$. The effective length ($L_{eff}$) is the length of the adiabatic part of the heat pipe plus half of the condenser length and half the evaporator length. The entrainment limit occurs when the liquid force is unable to overcome the shear force exerted by the vapor in the heat pipe given by Equation (6).2 Entrainment results when the vapor forces the liquid back into the condenser. 

Equations 

$$P_{c,max} = \frac{2\sigma_k}{r_e}$$  

(1)  

$$Q_{max} = \frac{4\pi r_1 k_{eff} \frac{1}{\mu \rho \log \left( \frac{r_1}{r_e} \right) \left( \frac{1}{r_e} - \frac{1}{r_e} \right)}}{H_{vp} \mu \rho}$$  

(2)  

$$Q = h A_s(T_{sat} - T_s)$$  

(3)  

$$h = 0.555 \left( \frac{g \rho_i (\rho_i - \rho_v) k_l H_{vp}}{\mu_i (T_{sat} - T_s) D_v} \right)^{\frac{1}{4}}$$  

$$H_{vp} = H_{vp} + 3 \frac{3}{8} C_p (T_{sat} - T_s)$$  

(4)  

$$Q_{max} = 0.474 H_{vp} (\rho_v P_v)^{\frac{1}{2}}$$  

(5)  

$$Q_{max} = \frac{r_2 H_{vp} \rho \rho_v}{16 \mu + \mu_{eff}}$$  

(6)  

Heat Pipe Model - Radiator Fins 

To model the radiator the Stefan-Boltzmann Law (Equation (7)) was used to calculate the heat transfer from the radiator. The equation was formatted to allow different emissivities and sink temperatures for each side of the radiator. For the study, the base temperature ($T_{base}$) was held at 323 K and radiated to deep space ($T_{ks}$) at 4 K. The emissivity was held constant at 0.8 and irradiation was assumed to be zero. The efficiency ($\eta_e$) was calculated through Chang’s fin efficiency equations (Equation (8-11)).1 For the model it was assumed that the only the face sheets conducted and radiated out, acting as fins. In Equation (8) the thickness of the base ($t_b$) is two times the face sheet thickness, ($k$) is the conductivity of the face sheet material and ($L$) is the spacing between the heat pipes. The Matlab code ran numerous combinations of face sheet thicknesses and heat pipe spacing to give a mass optimized radiator configuration. 

Equations 

$$Q = \sum_{k=1}^{2} \varepsilon_k \eta_e \sigma \left( T_{base}^4 - T_{ks}^4 \right)$$  

(7)  

$$\eta_e = (1 = 1.25 \xi + 1.6\xi^2)(1 - \theta^{*4})$$  

(8)  

$$for \ 0.01 \leq \xi \leq 0.2$$  

$$\eta_e = (-0.4049 \log \xi + 0.5321)(1 - \theta^{*4})$$  

(9)  

$$for \ 0.2 \leq \xi \leq 2.0$$  

$$\xi = \frac{\sigma \left( T_{base}^{1/4} \right)^2}{4 \kappa T_{base}}$$  

(10)  

$$\theta^{*4} = \frac{\varepsilon_1 \theta_1 + \varepsilon_2 \theta_2}{\varepsilon_1 + \varepsilon_2}$$  

(11)  

where $\theta_1 = \frac{T_{1s}}{T_{base}}, \theta_2 = \frac{T_{2s}}{T_{base}}$

Single Phase Pumped Loop Model 

For the single phase pumped loop (SPPL) system, assumptions were made to simplify the calculations necessary to find the heat rejection across the radiator. First, the radiator was assumed to be square with the geometry being dependent on the length and size of the loop. Also, for a pumped loop system, the weight of the pump must be considered when finding the mass of the system. For this study it was assumed that the pump had a base mass of 3 kg with an additional 29 kg/kW to take into account the influence an increase in power rejection has on the mass of the pump.5 The radiator considered was defined by the diameter of the pipe, which affected the pipe and radiator thickness, the pipe length and number of pipe runs across the radiator, which defined the radiator size, facesheet thickness and mass flow rate of the fluid, which affect the heat rejection of the radiator. 

The final heat rejection from the radiator was calculated by using heat transfer equations to determine the energy balance across the radiator. The radiator was divided into sections and each section was represented by a node. This node was assigned an energy equation that was composed of the different heat transfer equations for conduction, convection and radiation depending on the location of the node on the radiator. These equations were then solved through an iterative method until the energy was balanced across the radiator. Once the final temperatures on the radiator were found, they were
used in Equation (12) to find the total heat rejected from the radiator.

\[ Q = \int \varepsilon \sigma (T(x,y)^4 - T_{space}^4) \, dA \]  

(12)

In this equation, \( \varepsilon \) is the emissivity of the facesheet, \( \sigma \) is the Stefan-Boltzmann constant, \( T \) is the temperature across the radiator as a function of location and \( T_{space} \) is the sink temperature.

The fluid properties are also necessary to find the mass of the radiator since they affect the pump mass and the pipe. Using these properties and the fluid velocity, the Reynolds, Nusselt and Prandlt numbers were calculated. These numbers were then used to find the pressure drop across the system, which was used along with the mass flow rate to find the pump mass needed for the system. The final mass of the system was found by adding the masses of the glue, aluminum facesheet, OSR, aluminum honeycomb, pipe, fluid and pump.

The goal was to create a minimum mass radiator system. This was done by varying the pipe diameter, mass flow rate of the fluid, facesheet thickness, pipe length and number of runs of pipe across the radiator within the procedure described. From the radiator configurations generated the minimum mass solution was found.

**Loop Heat Pipe Model**

The objective of this model was to simulate the operation of a loop heat pipe (LHP) given a specified configuration and to observe the effects of varying certain parameters on the overall performance of the LHP. Loop heat pipes are two-phase thermal management devices that pump a working fluid via capillary action in a remote evaporator. As a passive system, the LHP’s heat rejection ability is subject to various limits. One such limit is when the wicking material in the evaporator cannot provide a pumping head greater than the frictional losses in the system. At this point, only a fixed amount of coolant per time can be evaporated, and the available heat rejection of the LHP reaches a maximum. Another limiting factor is the fluid condensation. The coolant must fully condense before returning to the evaporator for wicking to occur. Given this, the condenser tubing of the LHP must be of sufficient length to condense the coolant at a reasonable temperature.

Code was written in MATLAB to model an LHP. The program was given the heat load at the evaporator and calculated the operating temperature of the loop given a steady state energy balance on the system. The model showed that for a given LHP configuration, the operating temperature will increase with increasing heat load. Assuming 323°K is the upper temperature limit of the spacecraft systems being cooled, there exists a maximum heat load that can be safely dissipated by the LHP for any given specifications.

**RESULTS AND DISCUSSION**

The optimized heat pipe, single phase pumped loop and loop heat pipe radiators are compared in Figure 2. The heat pipe radiator was found to be the lightest configuration for a load of less than 700W. The liquid loop pump mass was initial significant, driving its comparative mass difference between the other radiators. The loop heat pipes were initially heavy due to evaporator and compensation chamber mass. The radiators became approximately equivalent after a one kW load, here the radiators are believed to approach an ideal radiating surface.

![Figure 2: Comparison of Heat Pipe, Single Phase Pumped Loop and Loop Heat Pipe Radiators](image)

The slight nonlinearity for the heat pipe radiator was caused by a restriction to whole number configurations of heat pipes for a given power load. This created variations in the heat loads the heat pipes were designed for in the form of (kW/heat pipe). Also, the 200W and 600W loads optimized using a 2.2cm diameter pipe while the other power loads investigated were optimized using a 2cm diameter. With a 2cm diameter the 200W load were heavier, as a two pipe configuration was needed to pass the capillary limit while only one pipe was needed if the 2.2cm diameter was used. The 10kW heat pipe radiator was calculated by placing ten of the 1kW radiators in parallel.

The upturns of the loop heat pipe line in Figure 2 are explained by an increased pipe diameter. The optimal loop heat pipe went from a 2mm diameter to a 5mm
diameter at 0.9kW due to the failure of the capillary limit in the 2mm diameter pipe. An increase in the pipe diameter to 5mm decreased the pressure losses and enabled the loop heat pipe to meet the capillary limit of its wicking material. At approximately 4.5kW the 5mm diameter transitions to the 10mm diameter loop heat pipe. At 10.5kW the 10mm diameter loop heat pipe fails the capillary limit and a larger diameter is recommended beyond this point.

For the single phase pumped loop the large increase in mass beyond 0.4kW was due to the increase in pipe length, which results in a large increase in area. The length increase also affected the pump mass since more pumping power was required to maintain the mass flow rate.

Numerical data for the mass and area of the radiators is visible in Table 3. The required area of the LHP and SPPL was similar to the heat pipe radiator. However, the use of a heat pipe radiator at high thermal loads is impractical as it is not deployable and requires a significant area. The necessary width of the heat pipe radiator at a 10kW load was 9.1m, well above the normal rocket payload threshold. The radiator could be divided into sections; however, it would still be large. The SPPL area requirements became larger than the other systems as the working fluid temperature drop effected its radiation.

| Table 3: Radiator Mass and Top Surface Area for Selected Loads |
|-----------------|-----------|---------|
| Radiator | Load (kW) | Mass (kg) | Area (m²) |
| HP | 0.1 | 0.95 | 0.13 |
| | 1 | 8.39 | 1.27 |
| | 10 | 83.9 | 12.7 |
| LHP | 0.13 | 2.26 | 0.30 |
| | 1.02 | 10.09 | 1.37 |
| | 10.27 | 105.31 | 12.39 |
| SPPL | 0.08 | 3.57 | 0.09 |
| | 1.50 | 11.67 | 2.14 |
| | 6.16 | 52.26 | 13.38 |

The percent mass breakdown for the heat pipe radiator is shown in Figure 3. The dominant portion of the mass was the heat pipes, as this mass included the evaporator, adiabatic and condenser length. The loop heat pipe mass breakdown is given in Figure 4. The dominant portion of the mass was the facesheets and OSR. The variation between the heat pipe and loop heat pipe mass breakdowns is explained by the wicking material mass and the difference in pipe diameters. The optimal heat pipe diameter was 2cm for the 0.1kW, 1kW (10kW) loads. Two centimeters was the smallest diameter the program was run for to help meet the capillary limit. The diameters used for the loop heat pipes were smaller as the wicking material is not internalized in them, which allows for the full use of the pipe diameter for the liquid and vapor flows. Also, the wicking material for the heat pipe must run the length of the pipe adding more mass to the pipe. The optimal loop heat pipe diameters were 0.2cm, 0.5cm and 1cm respectively for the 0.1kW, 1kW and 10kW loads. With smaller diameters the pipe mass and liquid mass decreased significantly for the loop heat pipe.
exceeding 0.7kW as the pump mass is very significant at lower levels.

Figure 5: Single Phase Liquid Loop Radiator Mass Breakdown.

The practicality of a high thermal load heat pipe system was in question if a header heat pipe system is utilized. Header heat pipes are larger heat pipes used to transport heat from the heat source to the heat pipes in the radiator. The use of a heat pipe radiator with a single or double header heat pipe is believed to become impractical at 1kW, as the mass of the header heat pipes should exceed the radiator mass. Using only a few heat pipes to construct the header heat pipes forces the diameter of the pipes to be large to meet their design limits. Figure 2, does not account for any header heat pipe mass as it is a radiator comparison.

Header heat pipes are not necessary if the heat pipes in the radiator are laid out to allow contact with the internal cold plate of the satellite. However, this is challenging if the satellite architecture has the heat pipe bending multiple times to reach the cold plate as it is difficult to manufacture a heat pipe with multiple bends.

Validations

Thermal Desktop and FlowCAD were used to check the results of the single phase pumped loop and the heat pipe system. A representative case was chosen and modeled, and the resulting heat rejection was compared to the heat rejection calculated by the optimization function. An example output from Thermal Desktop is shown in Figure 6.

Figure 6: Sample Thermal Desktop Radiator for the Single Phase Pumped Loop

The results from the Thermal Desktop model showed an 11% difference between them, which is a reasonable correlation between an ideal and an experimental model.

The heat pipe radiator model was also validated using Thermal Desktop. Using the optimum radiator as the test case, a difference of 9.4% was found between the two models.

CONCLUSIONS

Three different types of radiators were optimized and compared. The heat pipe radiator is recommended for a thermal load below 700W if the radiator is able to radiate to space from both sides. A heat pipe radiator radiating to space from a single side should be considered for loads at or less than 100W given the linear data trend. Loop heat pipe and single phase pumped looped radiators should be considered for loads exceeding 700W as their configuration allows for deployable radiators. For radiators operating above 700W alternate materials for the face sheets and piping should be sought to lower the system mass. The practicality of the heat pipe radiator is believed an issue at or greater than 1kW if header heat pipes are used due to the system’s mass. A rigid heat pipe radiator would be unrealistic with a 10kW or greater thermal load due to the necessary radiator size. The loop heat pipe is an attractive alternative to the pumped loop for smaller systems as it doesn’t require power to function.

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REFERENCES


