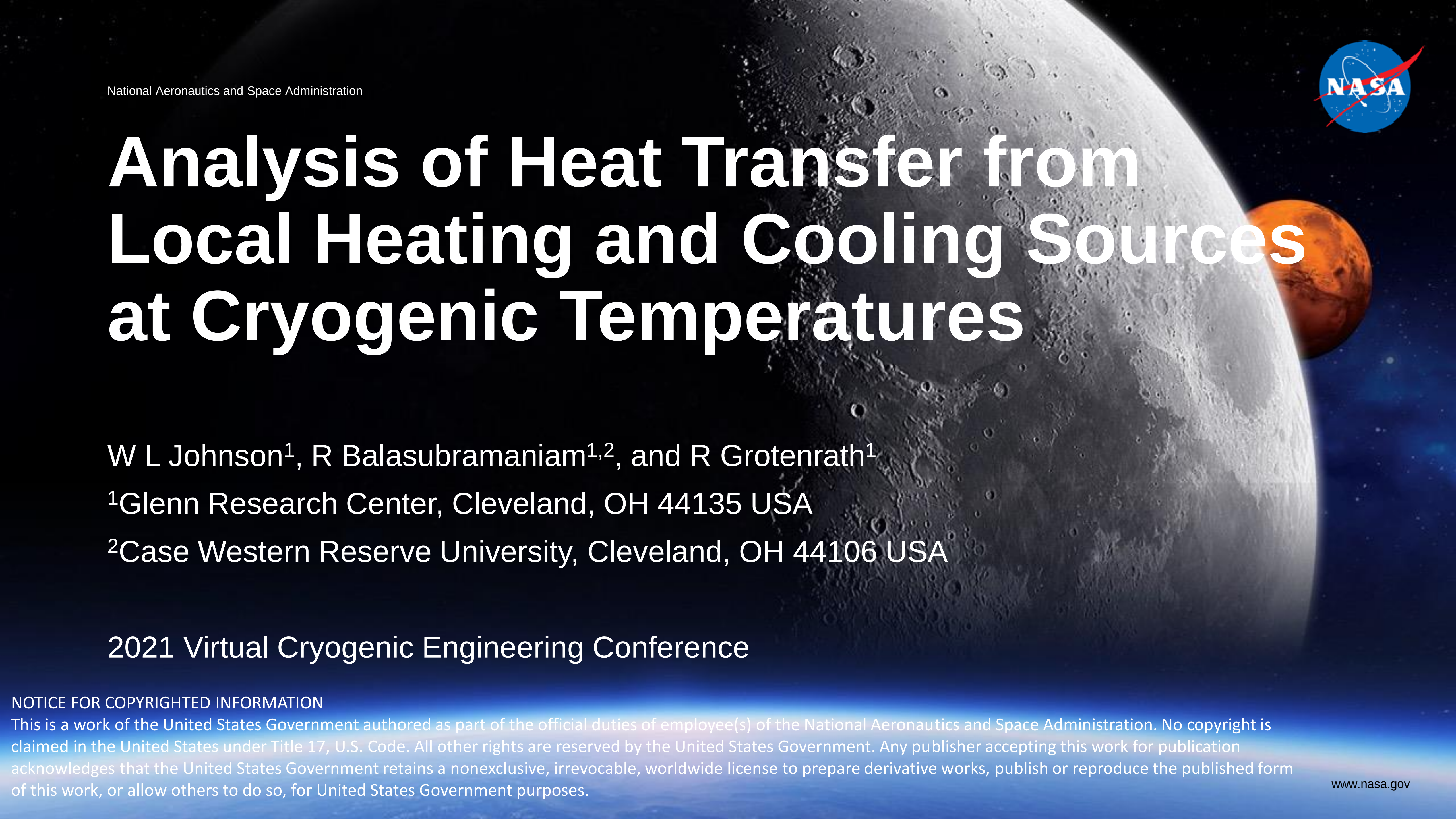

Analysis of Heat Transfer from Local Heating and Cooling Sources at Cryogenic Temperatures

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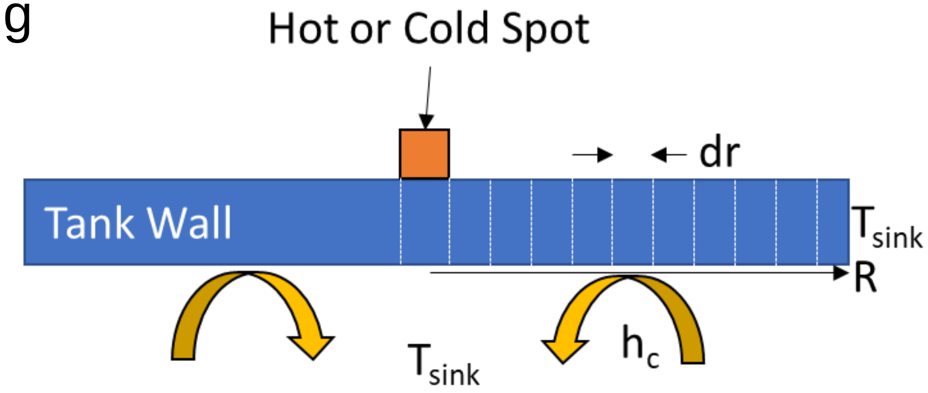
Introduction

- Isothermal wall solutions are the simplest to calculate for evaluating heating inputs into the tank and the effects of the heating on the fluid, these assumptions are often used in terrestrial systems to great effect.
    - Due to significant decreases in natural convection on orbit, an isothermal wall assumption may not be appropriate in microgravity.
    - The most non-isothermal wall locations would be where discrete heating or cooling is incurred on a tank.
- Data from the Tank Pressure Control Experiment, Zero-Boil-off Tank ISS experiment, and Robotic Refueling Mission #3 showed that local heat loads can cause impactful effects on a two-phase fluids system in microgravity.
    - Multiple pressure spikes observed
    - Ullage location effected by temperature gradients as well
- In preparation for future orbital fluids system, the effect of heating and cooling spots on tank wall temperature distribution is explored.

Finite Element Model

- Finite Element Model
  - 11 nodes including central cooling/heating spot (distance can be changed).
  - Heat input/removal connection of finite width (in W).
  - Conduction from node at next outer spot (cylindrical coordinates).
    - Properties not temperature dependent.
  - Convection internally to the tank wall.
  - Bulk fluid, inside the tank and at wall edge boundary kept at constant sink temperature.
  - Neglect any environmental heating/cooling on the wall (i.e. well insulated wall).
  - Solves conservation of energy at each node by iterating temperature.
- Calculates heat flux, temperature at each node
  - Size of nodes can be adjusted to improve accuracy, capture size of area of impact.
- Care most about heat flux and temperatures at central node at hot/cold spot

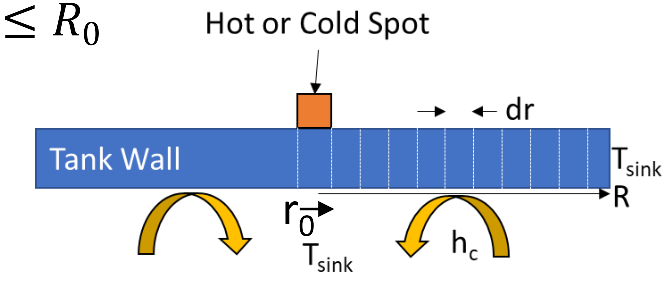


Analytical Solutions

- Point Heat Source on surrounding disk ($r_0 = 0$):

$$T = T_0 + \frac{Q}{2\pi k d}\left\{K_0\left(\sqrt{Bi}\,\frac{r}{R_0}\right) - \frac{K_0\left(\sqrt{Bi}\right)}{I_0\left(\sqrt{Bi}\right)}\, I_0\left(\sqrt{Bi}\,\frac{r}{R_0}\right)\right\}, \qquad 0 < r \le R_0$$

- Finite Heat Source on surrounding disk ($r_0 > 0$):



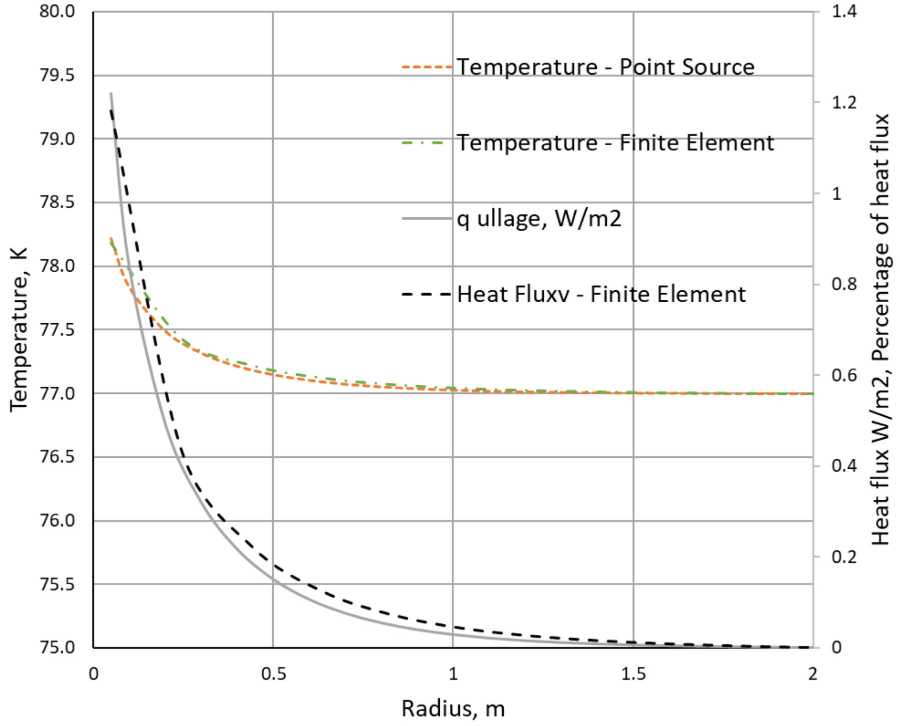
$$T = T_0 + \frac{Q}{2\pi k d}\, B\left\{K_0\left(\sqrt{Bi}\,\frac{r}{R_0}\right) - \frac{K_0\left(\sqrt{Bi}\right)}{I_0\left(\sqrt{Bi}\right)}\, I_0\left(\sqrt{Bi}\,\frac{r}{R_0}\right)\right\} \qquad r_0 \le r \le R_0$$

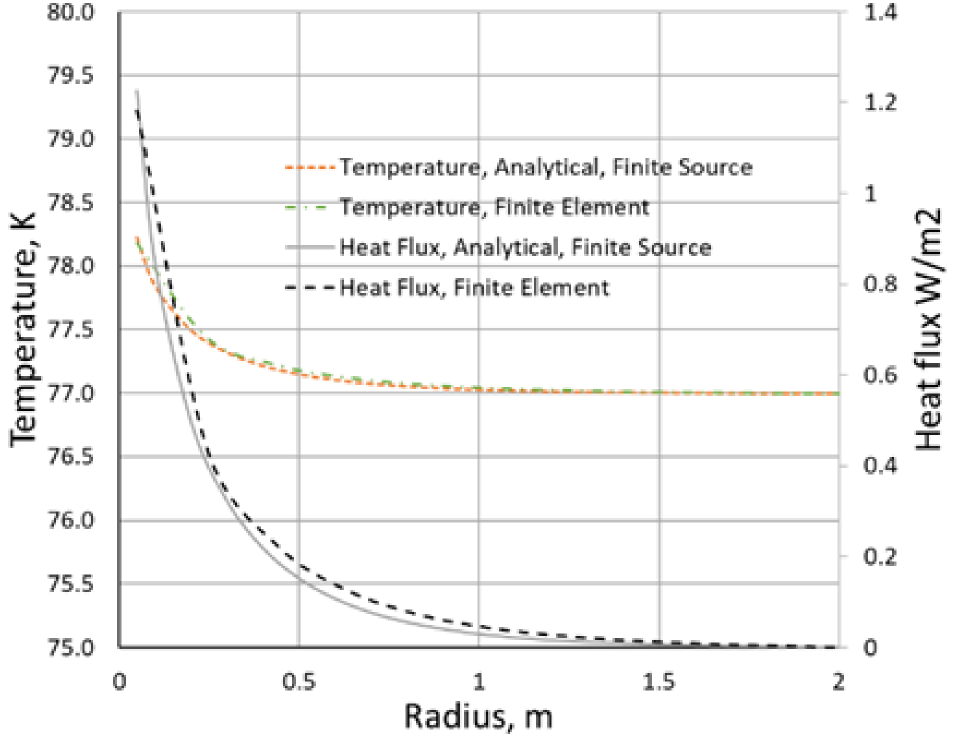
$$B = \left\{\frac{r_0}{R_0}\sqrt{Bi}\left[\frac{K_0\left(\sqrt{Bi}\right)}{I_0\left(\sqrt{Bi}\right)}\, I_1\left(\sqrt{Bi}\,\frac{r_0}{R_0}\right) + K_1\left(\sqrt{Bi}\,\frac{r_0}{R_0}\right)\right] + \frac{1}{2}\left(\frac{r_0}{R_0}\right)^2 Bi\left[-\frac{K_0\left(\sqrt{Bi}\right)}{I_0\left(\sqrt{Bi}\right)}\, I_0\left(\sqrt{Bi}\,\frac{r_0}{R_0}\right) + K_0\left(\sqrt{Bi}\,\frac{r_0}{R_0}\right)\right]\right\}^{-1}$$

d is the thickness of the disk
k is its thermal conductivity
r_0 is the radius of the heat source
R_0 is the outer radius, h is the heat transfer coefficient

Bi = $(hR_0$^2)/kd is the Biot number
T_0 is the bulk fluid temperature
Q is the heat load
$I_n(x)$ and $K_n(x)$ are Bessel's functions of order n

Comparison between Finite Element and Analytical


Point Source


Finite Source

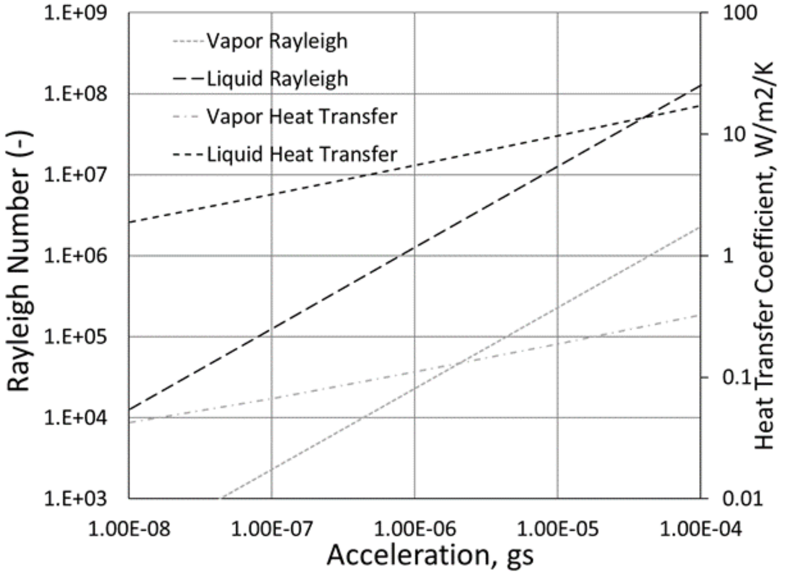
Good agreement

Further results show peak heat flux and temperature

Good agreement

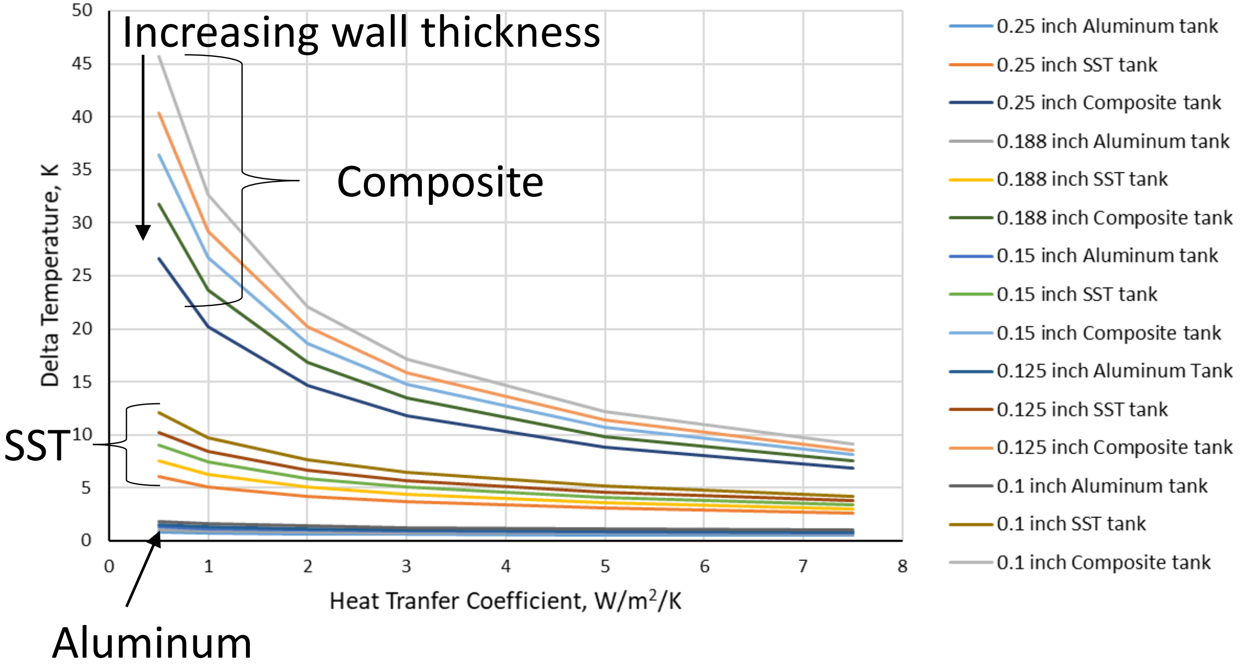
Analysis Run Matrix

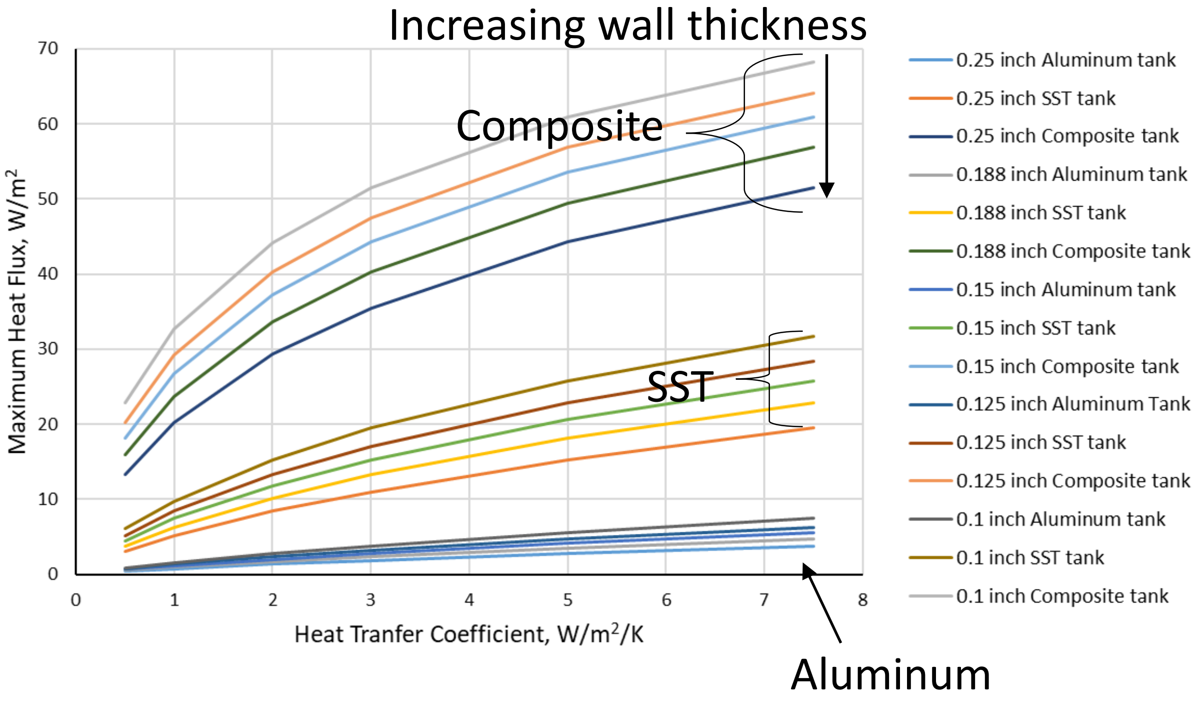
- Multiple different variables were investigated over the range indicated in the table below
- The range of heat transfer coefficients was from analysis for different fluids based on vertical flat plate heat transfer correlations.



Variable	Hot Spot	Cold Spot
Tank Material (thermal conductivity)	90 W/m/K (AL 6061) 8 W/m/K (SST 304) 1 W/m/K (composite)	
Heat Transfer Coefficient	0.5 – 7.5 W/m^2/K	
Heat input/removal rate	0.25 W – 1 W	-5 W – (-10)W
Wall thicknesses	0.00254 m – 0.00635 m (0.1 inch to 0.25 inch)	
Heat Sink Temperature	77 K	
Cryocooler Connection Diameter	0.05 – 0.10 m (2 inch to 4 inch)	

Tank Material Difference for 1 W heat source



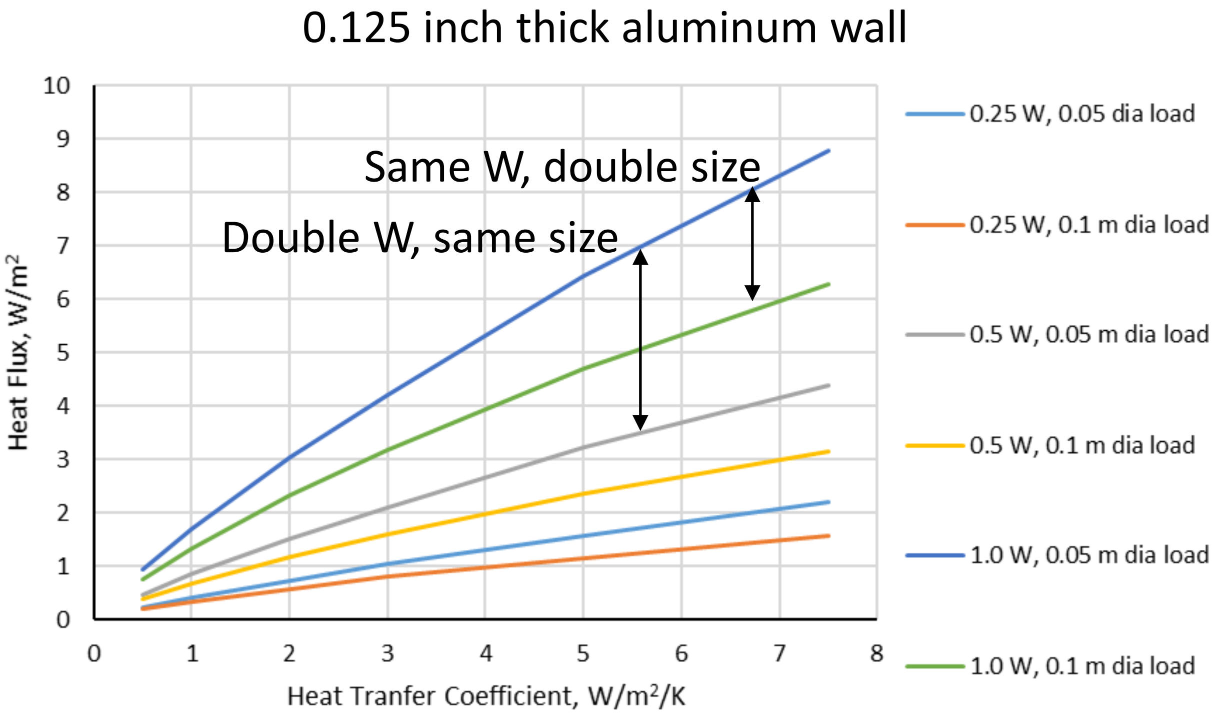


The three dominant variables from this plot, in order of importance, are:

- Tank wall material (result varies by greater than a factor of 10)
- Natural convection heat transfer coefficient (result varies by factor of 3)
- Wall thickness (result varies by factor of 2)

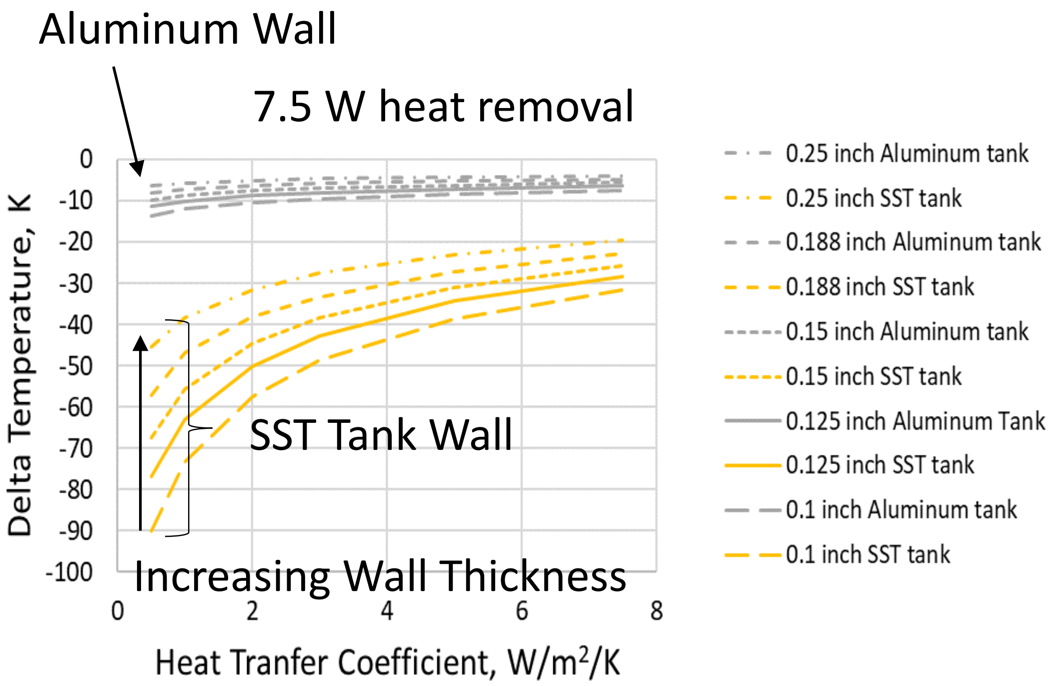
Effect of Heat Load and Interface Size

- Increasing heat input shows higher heat fluxes.
- Interface size of the heat source (load diameter) also has secondary impact on peak heat flux.
  - Increasing the size with the same heat input decreases peak heat flux.



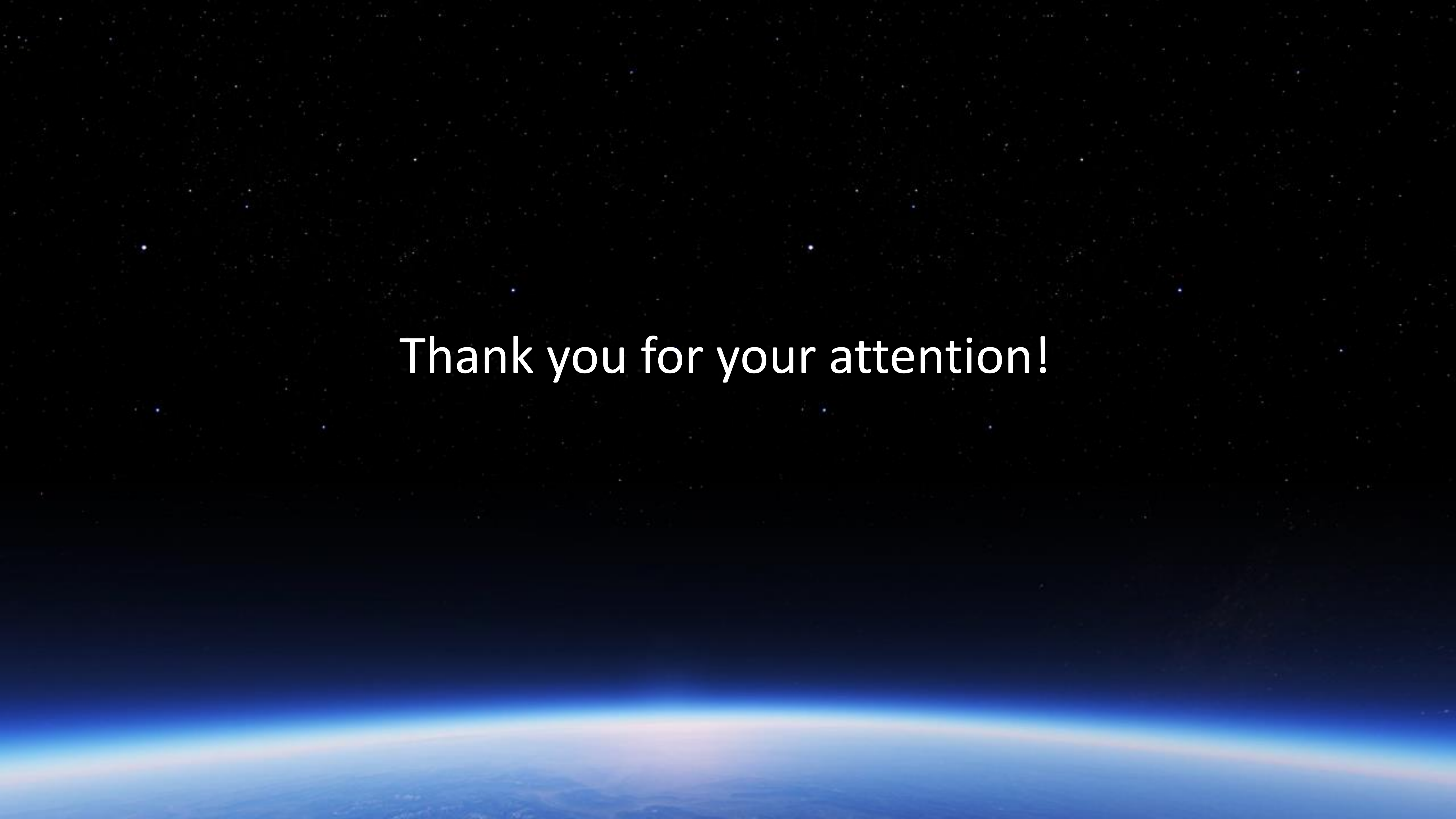
Results: Cold Spot

- The delta temperature shown is a direct thermal penalty requiring the cryocooler to run at a colder temperature to remove all of the energy desired.
- The use of a stainless steel, thin wall tanks with any real level of cooling, appears to be relatively impractical in a microgravity environment.
  - Composite tanks would be even worse, and so were not evaluated.



Conclusions

- A simplified one-dimensional, first order model was developed to look at local heat concentrations from heat sources and sinks.
  - Both a simple finite element model as well as direct analytical solution were developed, with the results being very similar given the same assumptions.
- Concentration of heat flux from a hot/cold spot is highly dependent on the local heat transfer coefficient.
  - On Earth, the natural convection is strong enough to overcome these issues, in microgravity, there is concern.
- Material choices also play a significant role in how the heat spreads within the tank wall.
- Variation of parameters within the model allow for an interesting investigation into multiple effects, all of which could not be demonstrated in the short paper/presentation.



Thank you for your attention!