Theoretical Limits of Vapor Cooling Large Cylindrical Structures using Boil-Off

R. Balasubramaniam¹,² and Wesley L. Johnson²

¹ Case Western Reserve University, Cleveland, Ohio
² NASA Glenn Research Center, Cleveland, Ohio

Abstract

The Structural Heat Intercept, Insulation, and Vibration Evaluation Rig (SHIIVER) is planning to demonstrate the performance benefits of using boil-off to reduce the heat load on various upper stage structural members by using the boil-off vapor to intercept some of the heat load on the structures. A first order simplified analytical model was constructed in order to understand sensitivity of various parameters to performance as well as to understand the theoretical maximum performance of the vapor cooling of a thin-walled cylindrical structure. Constant fluid properties and a temperature dependent thermal conductivity of the structure material were assumed. The model showed that vapor cooling of an aluminum cylindrical structure with hydrogen can reduce the heat load along the cylinder by as much as 70%. Also, cooling a portion of the cylinder is quite effective in comparison with cooling it entirely.

1 Introduction

As NASA moves towards exploration beyond Low Earth Orbit, long duration storage and management of cryogenic fluids will become very important. The Structural Heat Intercept, Insulation, and Vibration Evaluation Rig (SHIIVER) is a large scale cryogenic fluid management test bed developed to scale these technologies for inclusion on large, in-space stages (Johnson et al., 2018).

An important objective of the SHIIVER project is the vapor cooling of the forward skirt using the gaseous hydrogen that is produced by the boil-off of the LH₂ stored inside the cryogenic tank. Large heat loads enter through the support structure to the propellant tank, and most launch vehicles use cylindrical skirt type mechanical supports due to the location of the tanks in the vehicle stack. The basic idea for vapor cooling is to use the boil-off H₂ gas that is cool to intercept and remove a part of the structural heat load, and therefore reduce the heat leak into the tank. Helium and other cryogenic fluid based dewars for various orbital telescopes and observatories have long used the boil-off vapor routed around structural elements to reduce the heat coming through the structure (Hopkins and Payne, 1987, Lee, 1990). Most of these dewars used strut based mechanical supports to minimize heat load into their relatively small tanks (Lee, 1990, Read et al., 1999). In a similar fashion, the SHIIVER project will route propellant boil-off vapor around skirts to reduce heat flow into the propellant tanks.
1.1 Problem statement

The SHIIVER project will attempt to characterize vapor cooling on the forward skirt in thermal vacuum tests to be performed at the In-Space Propulsion Facility at NASA's Plum Brook Station. Consider a cylindrical forward skirt as shown in Figure 1. The outer surface of the skirt absorbs heat radiatively from the surroundings. We assume that the inside wall of the skirt is insulated (by suitable layers of Multilayer insulation (MLI)), and the top portion of the skirt is essentially adiabatic because it has a small exposed area. The bottom of the skirt is in thermal contact with the cryo tank. We will assume that the temperature of this surface is nearly the same as the temperature of \( \text{LH}_2 \) \( \approx 20 \) K at 1 atm pressure). At steady state, the radiative heat load on the outer skirt surface from the vacuum chamber is conducted along the skirt, from its top to the bottom, and removed by conduction from the bottom of the skirt. This heat leaks into the tank, causing the LH\(_2\) to boil-off. The vapor cooling concept utilizes the boil-off H\(_2\) gas, that is cool, to flow through channels along the interior wall of the skirt. The H\(_2\) gas flow removes some of the heat that is conducted along the skirt, analogous to a counterflow heat exchanger. The intercepted heat is returned to the surroundings when the hydrogen gas is vented.

It is surmised that there is a limit to the maximum amount of heat reduction by the H\(_2\) gas flow. For the sake of argument, assume that the gas flow rate for vapor cooling can be independently controlled, and consider specifically what happens as the gas flow is increased. We anticipate the gas to intercept more of the heat load from the surroundings. Because of the countercurrent nature of the gas flow, we anticipate a reduction in the temperature gradient near the bottom of the skirt where the cool gas emanates. Therefore, the heat leak into the cryo tank is reduced. In reality, the reduced heat leak will cause a reduction in the boil-off rate, which is counter to the assumption of an increase in the gas flow rate. Were the entire heat load be intercepted by the flowing vapor, the heat leak into the tank will be zero, which will lead to zero boil-off and no availability of vapor for cooling. Thus, we anticipate that the process of vapor cooling of the skirt will be self-limiting, and the heat leak into the tank cannot be completely eliminated.

In the analysis we will explore two cases: (i) where the majority of the heat flow is through the skirt and all the boil-off is used for vapor cooling (ii) where there are other heat sources.

2 Analysis

The goal of the current analysis is to determine the extent of reduction in the heat leak into the tank when vapor cooling is used. While the vapor cooling tubes are spiraling the inside surface of the forward skirt in Figure 1, we assume an axial flow of the vapor over the inside skirt surface in the model. We assume that:

1. The forward skirt heat transfer can be modeled as a radiating fin – the skirt receives a radiative heat load on its outer surface, which is absorbed and transferred by conduction axially from its top to bottom. The skirt temperature depends only on the axial position \( z \) – the temperature distribution is axisymmetric and the temperature variation over the skirt thickness is assumed negligible. Indeed, the Biot number (based on radiative heat load on the skirt, and its thickness and thermal conductivity) is quite small, of \( O(10^{-4}) \) or less.

2. There is a radiative heat exchange between the skirt outer surface and the environment, whose temperature is assumed constant.

3. The top of the skirt is adiabatic and the bottom is at a fixed temperature.

4. Vapor cooling is modeled by assuming a heat exchange between the inner surface of the skirt and H\(_2\) gas flowing over that surface (in an annulus) over an axial length \( 0 \leq z \leq L_1 \); \( L_1 \leq L \), where \( L \) is the height of the skirt. The gas is assumed to flow over the entire interior surface (and not along any specific channels or loops). A constant heat transfer coefficient \( h \) between the skirt and the flowing gas is assumed. The gas
Figure 1: Schematic of forward skirt vapor cooling.
temperature at \( z = 0 \) is \( T_{f,\text{in}} \) and the gas temperature at \( z = L_1 \) is to be determined. The gas is vented at \( z = L_1 \) and carries away the amount of heat transferred to it. A sketch of the model is shown in Figure 2.

5. The thermal conductivity of the skirt is assumed to be temperature dependent, with a temperature relationship for aluminum (aluminum 6061-T6) from Marquardt et al. (2000).

6. A steady state is assumed. The boil-off rate is taken to be commensurate with the heat transferred from the bottom of the skirt, with the entire heat assumed to leak into the tank and cause \( \text{LH}_2 \) to vaporize.

The governing equations for the skirt temperature \( T \) and the gas temperature \( T_f \) are

\[
A \frac{d}{dz} \left( k \frac{dT}{dz} \right) + \epsilon \sigma P \left( T_a^4 - T^4 \right) = \dot{m} C_p \frac{dT_f}{dz} = \pi D h (T - T_f), \quad 0 \leq z \leq L_1 \tag{1}
\]

\[
A \frac{d}{dz} \left( k \frac{dT}{dz} \right) + \epsilon \sigma P \left( T_a^4 - T^4 \right) = 0, \quad L_1 \leq z \leq L \tag{2}
\]

\[
T = T_a, \quad T_f = T_{f,\text{in}} \text{ at } z = 0 \tag{3}
\]

\[
\frac{dT}{dz} = 0 \text{ at } z = L \tag{4}
\]

\[
T \text{ and } \frac{dT}{dz} \text{ are continuous at } z = L_1 \tag{5}
\]

\[
\dot{m} = f \frac{A}{h_{fg}} \left( k \frac{dT}{dz} \right)_{z=0} \tag{6}
\]

where the following quantities pertain to the skirt: \( A = \pi D t \) is the cross-section area, \( D \) is the diameter, \( t \) is the thickness, \( P = \pi D \) is the circumference, \( \epsilon \) is the emissivity of the skirt surface, \( k \) is the thermal conductivity. The ambient temperature is \( T_a \). \( C_p \) is the specific heat of the gas, \( h_{fg} \) is the latent heat of vaporization of \( \text{LH}_2 \), and \( h \) is the heat transfer coefficient between the interior surface of the skirt and the gas. In most of the calculations that we report, \( C_p \) is assumed to be a constant for simplicity, though the equations given above are valid even
when $C_p$ is temperature dependent. We will discuss selected results for a temperature dependent $C_p$. We will use $h = 50$ and 100 $W/(m^2K)$ in the calculations as well as address the limit of very large $h$. The hydrogen flow rate is expected to be less than a few grams per second. For example, consider the flow of hydrogen at 1 g/s in an annular space of width $\sim 5$ mm on the inner surface of the skirt. The Reynolds number for the flow is around 25. Laminar fully developed flow and heat transfer will prevail, with a heat transfer coefficient estimated to be around 30 $W/(m^2K)$. The factor $f$ in Eq (6) is nominally equal to 1, for which the boil-off rate is commensurate with the heat leak. $f > 1$ reflects that there are other (parasitic) sources of heat leak into the tank, while $f < 1$ means that the vapor cooling is under-sized, i.e., not all boil-off is routed along the skirt. Unless otherwise stated, we assume $f = 1$ in the calculations that we report; we do present some results with $f \neq 1$ later on. One might argue that $f \neq 1$ is not a fundamental case, as it leaves room for proper sizing of vapor cooling, or cooling the source of the parasitic heat leak.

Dimensionless quantities are defined as follows.

$$\theta = \frac{T}{T_a}, \quad \theta_f = \frac{T_f}{T_a}, \quad \xi = \frac{z}{L_1} \quad 0 \leq z \leq L_1$$

$$\theta_1 = \frac{T}{T_a}, \quad \xi_1 = \frac{L - z}{L - L_1}, \quad L_1 \leq z \leq L$$

The dimensionless equations and boundary conditions are

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta_f}{d\xi}$$

$$\theta = \theta_f + \beta h M \frac{d\theta_f}{d\xi}$$

$$\frac{d}{d\xi_1} \left( \hat{k} \frac{d\theta_1}{d\xi_1} \right) + \lambda_1 (1 - \theta_1^4) = 0$$

$$\theta(0) = \theta_c, \quad \theta_f(0) = \theta_{f,in}; \quad M = f \hat{k}(0) \theta'(0)$$

$$\theta_1'(0) = 0; \quad \theta(1) = \theta_1(1); \quad \theta'(1) + \frac{L_1}{L - L_1} \theta'_1(1) = 0$$

$$\hat{k} = \frac{1}{k_0} k(T); \quad \lambda = \frac{\epsilon \sigma P L_1^2 T_a^3}{Ak_0}; \quad \lambda_1 = \frac{\epsilon \sigma P (L - L_1)^2 T_a^3}{Ak_0}; \quad \beta = \frac{C_p T_a}{h_{fg}}; \quad \beta_h = \frac{k_0 t}{h L_1^2 \beta}$$

where $k_0$ is a reference value for the thermal conductivity, for example $k_0 = k(T_a)$.

When the heat transfer coefficient $h$ is large, the energy transfer from the skirt to the flowing gas occurs with little thermal resistance. In this limit the magnitude of $\theta - \theta_f$ (that depends on $\beta_h \theta'_f$ from Eq 10) can be estimated as follows. Since $\theta$, $\theta_f$ and $\theta'_f$ are all expected to be $O(1)$, and $\beta_h M \sim \frac{k(T_a) t}{h L_1^2} \frac{C_p T_a}{h_{fg}} = 0.009 \sim O(10^{-2})$ with $h = 50 W/(m^2K)$ and $L_1 = L = 1.5$ m, $\theta \approx \theta_f$ is justified. In the limit $h \gg 1$, $\beta_h \to 0$, and Eq (9) can be simplified to

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

$$\frac{d}{d\xi} \left( \hat{k} \frac{d\theta}{d\xi} \right) + \lambda (1 - \theta^4) = \beta M \frac{d\theta}{d\xi}$$

5
The system of equations has been solved with aluminum as the skirt material with the following values for various parameters: $D = 4 \text{ m}$, $L = 1.5 \text{ m}$, $t = 4.76 \text{ mm}$, $\epsilon = 0.14$, 0.7 and 1, $T_c = 20 \text{ K}$. Unless otherwise stated, the calculations assume that $T_{f,in} = T_c$. The emissivity values are for polished aluminum, white-painted aluminum and a black body. The temperature dependent conductivity of aluminum from Marquardt et al. (2000) is used (see Figure 3).

$$k(T) = 10^{(a+b \log_{10}T+c(\log_{10}T)^2+d(\log_{10}T)^3+e(\log_{10}T)^4+f(\log_{10}T)^5+g(\log_{10}T)^6+h(\log_{10}T)^7)} \frac{W}{m \cdot K}$$

(16)

where $T$ is in degrees Kelvin and $a = 0.07918$, $b = 1.0957$, $c = -0.07277$, $d = 0.08084$, $e = 0.02803$, $f = -0.09464$, $g = 0.04179$, $h = -0.00571$.

**Table 1: Heat transfer to the cryo tank and vapor flow with and without vapor cooling, for various values of emissivity, heat transfer coefficients, and cooling height.** Hydrogen flow fraction $f = 1$, temperature $T_f = 20 \text{ K}$, environment temperature $T_a = 293 \text{ K}$. 

![Figure 3: Thermal conductivity of aluminum.](image-url)
| $\epsilon$ | f | $T_a$ (K) | $T_f$ (K) | $h$ (W/(m$^2$ K)) | Tank (W) |
|---|---|---|---|---|---|
| 0.14 | 0 | 250 | 20 | infinite | 571 |
| 0.14 | 0 | 275 | 20 | infinite | 824 |
| 0.14 | 0 | 293 | 20 | infinite | 1047 |
| 0.14 | 1 | 250 | 20 | infinite | 207 |
| 0.14 | 1 | 275 | 20 | infinite | 261 |
| 0.14 | 1 | 293 | 20 | infinite | 357 |
| 0.14 | 1 | 293 | 30 | 100 | 304 |
| 0.14 | 1 | 293 | 40 | 100 | 415 |
| 0.14 | 1 | 293 | 50 | 100 | 547 |
| 1 | 0 | 225 | 20 | infinite | 1772 |
| 1 | 0 | 250 | 20 | infinite | 2403 |
| 1 | 0 | 275 | 20 | infinite | 3145 |
| 1 | 0 | 293 | 20 | infinite | 3752 |
| 1 | 1 | 225 | 20 | infinite | 498 |
| 1 | 1 | 250 | 20 | infinite | 625 |
| 1 | 1 | 275 | 20 | infinite | 764 |
| 1 | 0.75 | 293 | 20 | infinite | 1043 |
| 1 | 1 | 293 | 20 | infinite | 873 |
| 1 | 1.25 | 293 | 20 | infinite | 756 |
| 1 | 1 | 293 | 20 | 100 | 917 |
| 1 | 1 | 293 | 30 | 100 | 1026 |
| 1 | 1 | 293 | 40 | 100 | 1179 |
| 1 | 1 | 293 | 50 | 100 | 1409 |

Table 2: Heat transfer to the cryo tank for various values of emissivity, hydrogen flow fraction and temperature, and environment temperature. The entire skirt is cooled.
3 Results

The chief results from the calculations are presented in Tables 1 and 2. Table 1 shows the heat leak into the cryo tank, and the heat intercepted and carried away by the flowing vapor. Cases with and without vapor cooling are shown for skirt emissivity $\epsilon = 0.14$, 0.7 and 1. With vapor cooling, the cooling starts at the bottom of the skirt and ends at either quarter, half, or full height of the skirt. For each value of $\epsilon$ and cooling height, the heat leak into the tank as a percent of the corresponding heat leak without any vapor cooling is also shown. For the results in Table 1, the ambient temperature is 293 K, the hydrogen flow fraction $f = 1$ (i.e., the boil-off is commensurate with the heat leak from the skirt into the tank), and the temperature of the incoming hydrogen vapor is $T_f = 20$ K. The heat leak into the tank and the heat intercepted by the vapor are presented for various values of the heat transfer coefficient. Table 2 shows the heat leak into the tank when $f$, $T_a$ and $T_f$ are varied. For all the results reported in Tables 1 and 2, the temperature at the bottom of the skirt is $T_c = 20$ K.

The salient features from the results are noted below.

1. Vapor cooling reduces the heat leak into the tank in all cases. Figure 4(a) shows the heat load into the tank with and without vapor cooling and its dependence on the skirt emissivity. Figures 5(a) and 5(b) shows the ratio of the tank heat leak with and without vapor cooling versus cooling height, for the three values of heat transfer coefficient $h$ given in Table 1. For $\epsilon = 0.14$ the lowest value of the heat leak ratio is around 30%, and is around 25% for $\epsilon = 1$. There is only a slight dependence of the heat leak ratio on the value of $h$, especially for $\epsilon = 0.14$. The heat leak ratio increases with decreasing ambient temperature and the vapor cooling is correspondingly less, especially for $\epsilon = 0.14$, as shown in Figure 6 (entire skirt cooled; $h = \infty$). A cooling height of a quarter of the skirt height appears adequate and compares very well with cooling the entire skirt. A quantitative measure of the effectiveness of quarter cooling over full cooling can be defined by a cooling efficiency as $\eta_{25} = \frac{Q_{T_{\text{no cooling}}} - Q_{T_{\text{quarter cooling}}}}{Q_{T_{\text{no cooling}}} - Q_{T_{\text{full cooling}}}}$, where $Q_T$ is the rate of heat transfer to the tank. $\eta_{25}$ is 0.88, 0.95 and 0.97 for $\epsilon = 0.14$, 0.7 and 1, respectively, with $h = \infty$. Curiously, for a cooling height of one eighth of the skirt height that we calculated for selected cases, the corresponding cooling efficiency $\eta_{12.5}$ is reduced to 0.75, 0.88 and 0.90.

2. The total heat load from the environment (i.e., the heat leak to the tank and that intercepted by the vapor flow) increases with vapor cooling, as shown in Figure 4(b). This is because, with vapor cooling, more of the skirt is cooled by the vapor and the average skirt temperature decreases. Therefore, the temperature

![Figure 4: (a) Tank heat load and (b) Environmental heat load versus skirt emissivity, with and without vapor cooling. $T_a = 293$ K, $h = \infty$, with the entire skirt cooled.](image)
Figure 5: Tank heat leak ratio (ratio of tank heat leak with and without vapor cooling) versus cooling height fraction: (a) $\epsilon = 0.14$, (b) $\epsilon = 1$.

Figure 6: Tank heat leak ratio versus environment temperature. Entire skirt cooled; $h = \infty$.

Figure 7: Temperature distribution along the skirt versus scaled skirt height: (a) $\epsilon = 0.14$, (b) $\epsilon = 1$; red curve: no vapor cooling, blue curve: with vapor cooling. Entire skirt cooled; $T_a = 293$ K; $h = \infty$. 
differential between the ambient and the skirt increases, leading to an increase in the radiative heat load. The skirt temperature distribution with and without vapor cooling (entire skirt cooled; \( h = \infty \)) is shown in Figure 7. When \( h = \infty \), there is no difference between the skirt temperature and that of the hydrogen vapor, at all skirt locations. Figure 8 shows the difference between the skirt and hydrogen vapor temperatures along the height of the skirt, for \( \epsilon = 0.14 \) and 1, and \( h = 50 \) and 100 \( \frac{W}{m^2 K} \). Cooling a quarter of the skirt with the lower \( h \) has the maximum temperature difference, which is about 3.2 K for \( \epsilon = 0.14 \) and around 17 K for \( \epsilon = 1 \).

3. The effect of an undersized vapor cooling or a parasitic heat load has been modeled by varying the factor \( f \) in Eq(6) (or Eq(12)) between 0.75 and 1.25. In this calculation we have assumed \( T_a = 293 \) K, with full vapor cooling and \( h = \infty \). The heat leak into the tank is plotted against \( f \) in Figure 9, which shows that the reduced or increased hydrogen flow rate around the boil-off rate commensurate with the heat leak (i.e., \( f = 1 \)) has only a very modest effect on the tank heat load.

Figure 8: Difference between skirt and vapor temperature (K) for quarter, half and full vapor cooling with heat transfer coefficient \( h = 50 \) (dashed line) and 100 (solid line) \( \frac{W}{m^2 K} \). (a) \( \epsilon = 0.14 \), (b) \( \epsilon = 1 \). red curve: quarter cooling, green curve: half cooling, blue curve: full cooling.

Figure 9: Effect of variation of hydrogen flow rate: (a) \( \epsilon = 0.14 \), (b) \( \epsilon = 1 \); red curve: no vapor cooling, blue curve: with vapor cooling. Entire skirt cooled; \( T_a = 293 \) K; \( h = \infty \).
4. Figure 10 shows that varying the inlet temperature of the hydrogen gas has a relatively big impact on the performance of vapor cooling, especially for $\epsilon = 0.14$. When the incoming hydrogen is warmer, it is unable to effectively intercept and carry away energy from the skirt. For a fixed value of $\epsilon$, of all the parameters we have looked at, the temperature of the incoming hydrogen has the most influence on vapor cooling.

[Graph showing the effect of varying hydrogen inlet temperature on vapor cooling performance.]

5. The results presented thus far use a constant value of $C_p = 12$ KJ/(kg K), which is approximately the average value of the specific heat of parahydrogen vapor (see Figure 11) between $T = 20$ K and $120$ K, at a pressure of 0.1 MPa. As mentioned before, our formulation is valid even when $C_p$ is temperature dependent. We now provide selected results when $C_p$ varies as shown in Figure 11. (i) For full vapor cooling with $\epsilon = 0.14$ and $h = \infty$, the heat load to the tank and that carried by the vapor flow are 315 and 777 W, respectively. This compares with 304 and 787 W (see Table 1) when $C_p$ is constant. For $\epsilon = 1$, the corresponding heat loads are 877 and 5835 W for a temperature dependent $C_p$, versus 873 and 5358 W when $C_p$ is constant. (ii) For full vapor cooling with $\epsilon = 0.14$, $h = 100 \frac{W}{m^2K}$ and vapor inlet temperature $T_f = 50$ K, the heat load to the tank and vapor flow are 526 and 572 W, respectively, compared to 547 and 552 W when $C_p$ is constant. The corresponding heat loads for $\epsilon = 1$ are 1398 and 6014 W for variable $C_p$, and 1409 and 5821 W for constant $C_p$. Thus, in these results the heat load to the tank for constant and variable $C_p$ differ by less than 4% for $\epsilon = 0.14$, and less than 1% for $\epsilon = 1$.

[Graph showing the specific heat of parahydrogen versus temperature at a pressure of 0.1 MPa.]
4 Conclusions

We have analyzed a simple model to understand the performance of vapor cooling of the forward skirt of a cryogenic liquid hydrogen tank. The results show that vapor cooling using boil-off hydrogen vapor is very effective in reducing heat load from skirt to tank and thus reducing boil-off. A maximum boil-off reduction of around 70% is predicted by the simplified analysis, when the hydrogen flow rate for vapor cooling is commensurate with the heat leak. Also cooling a quarter of the skirt from the cold end is sufficient, and performs very well compared to cooling the entire skirt. Effectiveness of vapor cooling is sensitive to the vapor inlet temperature, and is quite insensitive to skirt-to-vapor flow heat transfer coefficient and small changes in hydrogen flow rate.

References

R.A. Hopkins and D.A. Payne, “Optimized support systems for spaceborne dewars,” Cryogenics, vol. 27, no. 4, pp. 209 - 216, 1987.

https://webbook.nist.gov/chemistry/fluid/ (Accessed 2 February 2020).

W.L. Johnson, L.M. Ameen, F. D. Koci, D. Oberg and J.G. Zoeckler, “Structural Heat Intercept, Insulation, and Vibration Evaluation Rig (SHIIVER),” SP-2018-179, Space Propulsion 2018, 14-18 May 2018, Seville, Spain.

J.H. Lee, “Thermal performance of a five year lifetime superfluid helium dewar for SIRTF,” Cryogenics, vol. 30, no. 3, pp. 166-172, 1990.

E.D. Marquardt, J.P. Le and R. Radebaugh,“Cryogenic Material Properties Database,” 11th International Cryocooler Conference, Keystone, Colorado, June 20-22, 2000.

D.C. Read, R.T. Parmley, M.A. Taber, D.J. Frank, and D.O. Murray, “Status of the relativity mission superfluid helium flight dewar,” Cryogenics, vol. 39, no. 4, pp. 369-379, 1999.