Pressurization of a Flightweight, Liquid Hydrogen Tank: Evaporation & Condensation at a Liquid/Vapor Interface

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Outline

• Issues with evaporation/condensation at interface
• Temperature jumps at an interface—sharp gradients
• Interface physics: mass and energy transfer
• Model equation: solved two ways
• Numerical methods: subgrid model & coupling
• Results: practical demonstration of method
K-Site: Slow Self-Pressurization Captures Pressure Evolution

![Graph showing pressure evolution over time with different cases: Steady Boil-Off, Isothermal, and Lower T_Bulk Case.](image)
CNES Low-g Slosh: Heavy Boiling Phase with Condensation and Transit

00:26 in data

T=96.75 s in CNES_5C_7
CNES Low-g Slosh: Pressure Evolution

\[ \sigma_{\text{evap}} = \sigma_{\text{cond}} = 1 \times 10^{-4} \]

\[ \sigma_{\text{evap}} = \sigma_{\text{cond}} = 2 \times 10^{-4} \]
Temperature “Jumps” at the Interface

\[ W = -\int p \, dV \]

Work done on the vapor phase

Pressurization caused by:
- Pressurant
- Boiling liquid
- Clouds rise/fall
- Temperature gradients

Temperature gradient established
# Pressurization of a Compressible Gas

<table>
<thead>
<tr>
<th></th>
<th>Latent Heat $\Delta H_{vap}$ (J/kg)</th>
<th>$T_{sat}$ (K)</th>
<th>$\Delta T_{compress}$ 1-&gt;2 atm (K)</th>
<th>Vapor $\Delta H_{vap}/C_p$ (K)</th>
<th>Liquid $\Delta H_{vap}/C_p$ (K)</th>
<th>Vapor $\alpha$</th>
<th>Liquid $\alpha$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Helium</td>
<td>20,752.</td>
<td>4.2304</td>
<td>1.342</td>
<td>2.3</td>
<td>3.9</td>
<td>5.95E-05</td>
<td>2.82E-05</td>
</tr>
<tr>
<td>Methane</td>
<td>510,830.</td>
<td>111.67</td>
<td>20.433</td>
<td>230.3</td>
<td>146.7</td>
<td>2.88E-03</td>
<td>1.25E-04</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>199,178.</td>
<td>77.355</td>
<td>16.942</td>
<td>177.2</td>
<td>97.6</td>
<td>1.45E-03</td>
<td>8.86E-05</td>
</tr>
<tr>
<td>Oxygen</td>
<td>213,050.</td>
<td>90.188</td>
<td>19.752</td>
<td>219.5</td>
<td>125.4</td>
<td>1.93E-03</td>
<td>7.82E-05</td>
</tr>
<tr>
<td>Parahydrogen</td>
<td>445,440.</td>
<td>20.277</td>
<td>4.441</td>
<td>36.4</td>
<td>46.1</td>
<td>1.04E-03</td>
<td>2.81E-06</td>
</tr>
<tr>
<td>Water</td>
<td>2,256,440.</td>
<td>373.12</td>
<td>70.019</td>
<td>1084.9</td>
<td>535.3</td>
<td>2.02E-02</td>
<td>1.68E-04</td>
</tr>
</tbody>
</table>

Compressible Vapor

Incompressible Liquid
Energy/Heat Equation

\[ \rho C_p \frac{\partial T}{\partial t} - k \frac{\partial^2 T}{\partial x^2} = \dot{q}_{\text{interface}} + \dot{q}_{\text{vap}} \]

\[ \alpha = \frac{k}{C_p \rho} \]

Negligible:
- Fluid motion
- Temperature variation in interface plane
Heat Equation: Series of Exact Solutions

\[ \rho C_p \frac{\partial T}{\partial t} - k \frac{\partial^2 T}{\partial x^2} = \dot{q}_{\text{interface}} + \dot{q}_{\text{vap}} \]

\[ T^{\text{vap}}(x, t) = T^{\text{vap}}_\infty(t) + \sum_{j=1}^{t_j \leq t} \frac{Q_j^{\text{vap}}}{(4\pi \alpha_{\text{vap}} (t-t_j))^{1/2}} e^{\frac{-x^2}{4\alpha_{\text{vap}} (t-t_j)}}, \quad x \geq 0 \]

\[ T^{\text{liq}}(x, t) = T^{\text{liq}}_{\infty}(t) + \sum_{j=1}^{t_j \leq t} \frac{Q_j^{\text{liq}}}{(4\pi \alpha_{\text{liq}} (t-t_j))^{1/2}} e^{\frac{-x^2}{4\alpha_{\text{liq}} (t-t_j)}}, \quad x \leq 0 \]

\( T^{\text{liq}}_{\infty}, \) is assumed constant

\( T^{\text{vap}}_\infty(t), \) from isentropic compression

\[ \frac{T_1}{T_0} = e^{\frac{dS}{C_p}} \left( \frac{V_0}{V_1} \right)^{\gamma-1} \]

\[ (-k_{\text{liq}} \frac{dT}{dx})_{\text{interface-liq}} - (-k_{\text{vap}} \frac{dT}{dx})_{\text{interface-vap}} = \dot{q}_{\text{flux}} = m_{\text{flux}} (p_{\text{interface}}, T_{\text{interface}}) \Delta H_{\text{vap}}(T_{\text{interface}}) \]
Heat Equation: Numerical Solutions

\[
\rho C_p \frac{\partial T}{\partial t} - k \frac{\partial^2 T}{\partial x^2} = \dot{q}_{\text{interface}} + \dot{q}_{\text{vap}}
\]

\[
D(T_i^+ - T_i^-) - (T_{i+1}^+ - T_{i+1}^-) + (T_i^+ - T_i^-) + \frac{(\omega - 1)}{\omega} (T_{i+1}^- - T_{i+1}^-) - \frac{(\omega - 1)}{\omega} (T_i^- - T_i^-) = \frac{\Delta x^2}{\omega k} \dot{q}_{\text{vap}}
\]

\[
D = \frac{\rho C_p \Delta x^2}{\omega k \Delta t}
\]

Solved as a tridiagonal matrix

\[T_{\text{liq}}^{-\infty}, \text{ is assumed constant}\]

\[T_{\text{vap}}^{\infty}(t), \text{ from isentropic compression}\]

\[
\frac{T_1}{T_0} = e^{\frac{dS}{C_p}} \left( \frac{V_0}{V_1} \right)^{\gamma - 1}
\]

\[
\begin{align*}
(-k_{\text{liq}} \frac{dT}{dx})_{\text{interface - liq}} - (-k_{\text{vap}} \frac{dT}{dx})_{\text{interface - vap}} &= \dot{q}_{\text{flux}} = \dot{m}_{\text{flux}} (p_{\text{interface}}, T_{\text{interface}}) \Delta H_{\text{vap}} (T_{\text{interface}})
\end{align*}
\]
Temperature “Jumps” at the Interface

\[ W = - \int p \, dV \]

\[ \Delta T \text{ due to work done on vapor} \]
Intermission- Mid-Review

• Thermal Layers: role of heat near the interface
• Exact & numerical solutions: verification
• Evaporation/Condensation rates:
  – Temperature gradients at interface, O(1 mm)
  – Heat transfer near interface is important -- if not rate limiting
• From Physics, application to CFD simulation
CFD: Subgrid Model for Interface

• Fine grid needed to resolve thermal layers ~1mm
• Interface can move and curve
  – grid generation nightmare, even unstructured, adaptive grid
• Subgrid model moves with the interface
• Solves the 1-D heat equation normal to interface
• Four couplings between subgrid model and Fluent
• Energy & mass source terms in liquid/vapor equations
Coupling Between Subgrid Model and Simulation

Coupling 1:
\[ x_{\text{interface}} = \frac{\sum_{\Omega} x \varphi (1 - \varphi)}{\sum_{\Omega} \varphi (1 - \varphi)} \]
\[ y_{\text{interface}} = \frac{\sum_{\Omega} y \varphi (1 - \varphi)}{\sum_{\Omega} \varphi (1 - \varphi)} \]

Coupling 2:
\[ p_{\text{interface}} = \frac{\int_{\Omega} p \varphi_{\text{vap}} dV}{\int_{\Omega} \varphi_{\text{vap}} dV} \]

Coupling 3: Fluent Source Term (liquid & vapor)
\[ m_{\text{interface}} \]

Coupling 4: Fluent Source Term (liquid & vapor)
\[ q_{\text{interface}} \]

Mass and Energy are conserved
Must be careful about sizing Fluent source terms!!
EDU Tank

- 2219 Aluminum; Volume 4.34 m$^3$; I.D. 1.70 m; I. H. 2.33 m
- 1.25” SOFI, 60 layers MLI; 2.54 mm wall thickness
EDU CAD Geometry

- Axisymmetric geometry/grid
EDU CAD Geometry

- Diffuser Supply Line
- Tank Penetration
- Bolted Joint
- Unsubmerged Diffuser
- Tank Wall
- Tank Axis
Phase A Test Data

- Test HT-15, 16 on day 3 of Phase A testing
- 90% Fill level
- Pressurant gas at 290 K through the unsubmerged diffuser supply line
- Small drain flow, less than 1% of volume
$Q_{\text{vapor}} = -4.04 \text{ W/m}^2$; $Q_{\text{liquid}} = -54.03 \text{ W/m}^2$;
Condense Heat Flux $= 52.7 \text{ W/m}^2$; Condense Mass Rate $= -1.13e-4 \text{ kg/m}^2\text{s}$;
Results

- Good measure of condensation rate?
- For duration, pressurant inflow to condensation is between 1.5:-1 and 2:-1
- After 123 s, pressurant declines
- Assuming pressure release after 131 s
Conclusions

• Proof of concept for improved interface mass & energy transfer
• Accommodation coefficient of 1.0
• Extension to curved surfaces, multiple surfaces
• Need to examine other problems in the context of this result
Mass & Heat Equations

\[ \rho C_p \frac{\partial T}{\partial t} - k \frac{\partial^2 T}{\partial x^2} = \dot{q}_{\text{interface}} + \dot{q}_{\text{vap}} \]

\[ \dot{m}_{\text{flux}} (p_{\text{interface}}, T_{\text{interface}}) = \frac{2}{2 - \sigma_{\text{cond}}} \frac{\sqrt{MW}}{2\pi R_u} \left( \sigma_{\text{evap}} \frac{p_{\text{sat}}(T_{\text{liq}})}{\sqrt{T_{\text{liq}}}} - \sigma_{\text{cond}} \frac{p_{\text{vap}}}{\sqrt{T_{\text{vap}}}} \right) \]

\[ \dot{q}_{\text{flux}} = \dot{m}_{\text{flux}} (p_{\text{interface}}, T_{\text{interface}}) \Delta H_{\text{vap}} (T_{\text{interface}}) \]

\[ \frac{T_1}{T_0} = e^{\frac{dS}{C_p}} \left( \frac{V_0}{V_1} \right)^{\gamma - 1} = e^{\frac{dS}{C_p}} \left( \frac{p_1}{p_0} \right)^{1 - \frac{1}{\gamma}} \]