Absorbers–Simulations and Proposed Flow Test
Introduction: Approaches to Heat Removal

Two approaches under consideration:

1. External cooling loop (traditional approach).
   - Bring the coolant, i.e. $H_2$, to the heater to remove heat removed in an external heat exchanger.

2. Combined absorber and heat exchanger.
   - Bring the coolant, i.e. $H_2$, to the LH$_2$ to remove heat directly within the absorber.
Introduction (cont'd)

Advantages/disadvantages of an external cooling loop:

+ Has been used for several LH$_2$ targets (e.g. SLAC E158).
+ Easy to regulate bulk temperature of an external cooling loop.
+ More likely to work best for large aspect ratio (L/R) absorbers.

May be difficult to maintain uniform vertical flow through the absorber.
+ Is likely to work best for small aspect ratio (L/R) absorbers.
+ More difficult to ensure against boiling at very high Rayleigh numbers.

Advantages/disadvantages of a combined absorber/heat exchanger:

+ Takes advantage of natural convection transverse to the beam path.
+ Flow in absorber is self regulating, i.e. larger heat input increases turbulence and enhances thermal mixing.
+ Is likely to work best for large aspect ratio (L/R) absorbers.

More difficult to ensure against boiling at very high Rayleigh numbers.

May be difficult to maintain uniform vertical flow through the absorber.
+ Is likely to work best for small aspect ratio (L/R) absorbers.
+ Easy to regulate bulk temperature of several LH$_2$ targets (e.g. SLAC E158).
Energy balance between coolant (H) (H) and coolant (HT).

Parameters:

\[ \varepsilon H \]

Specific heat capacity of

\[ \varepsilon H \]

Thermal conductivity of cooling tube walls

\[ \varepsilon H \]

Heat transfer coefficient of cooling tube walls

\[ \varepsilon H \]

Heat transfer coefficients of LH2

\[ \varepsilon H \]

Thickness of cooling tube walls

\[ \varepsilon H \]

Heat transfer coefficient of LH2

\[ \varepsilon H \]

Heat transfer coefficient of He

\[ \varepsilon H \]

Surface area of cooling tubes

\[ \varepsilon H \]

Bulk temperature of LH2

\[ \varepsilon H \]

Outlet coolant temperature

\[ \varepsilon H \]

Inlet coolant temperature

\[ \varepsilon H \]

Parameters:
correlations for natural convection with heat generation.

\[ q = A \left( T_o - T_i \right) \frac{1}{h_{LH}^2} + x_k w + \frac{1}{h_{He}^2} \ln \frac{T_L}{T_o} \]

Mass flow rate of He:

\[ \frac{(iL - oL) d\rho}{b} = \rho_{He} \]

Rate of heat transfer:

\[ \frac{(iL - oL)}{(oL - z_{HTL})} \mu_l \left( \frac{\rho_{He} \mu}{l} + \frac{m_k}{x^2} + \frac{z_{HT} \mu}{l} \right) - \frac{1}{(iL - oL)V} = b \]
Features of the CFD Simulations:

1. Provides average convective heat transfer coefficient and average HT temperature for heat exchanger analysis.
2. Tracks maximum HT temperature (cf. boiling point).
3. Determines details of fluid flow and heat transfer in absorber.
4. Better understanding leads to better design!
Take 1: Results using FLUENT (M. Boghosian):
- Simulate one half of symmetric domain.
- Steady flow calculations.
- Heat generation via steady Gaussian distribution.
- Turbulence modelling (RANS) used for $RaH > 4 \times 10^8$.
- Investigate possibility of asymmetric flow oscillations.
- Investigate startup behavior, e.g. startup overshoot in $T_{max}$.
- All scales computed for all Rayleigh numbers.
- Unsteady flow calculations.

Take 2: Results using COA code (A. Obabko and E. Almarsi):
- Simulate full domain.
- Unsteady flow calculations.
- Investigate startup behavior, e.g. startup overshoot in $T_{max}$.
- Investigate possibility of asymmetric flow oscillations.
- Investigate influence of beam pulsing.
Average Nusselt Number vs. Rayleigh Number:

Results from Fluent CFD simulation.
Non-Dimensional Maximum Temperature vs. Rayleigh Number:

\[
\frac{T_{\text{max}}}{(T_{\text{max}} - T_{\text{w}})/(\kappa D_{\text{eff}}/k)} = x
\]
Parameter Map for $LH_2$

Note: Properties taken at 18 K, 2 atm.
COA Formulation

Properties and parameters:

- \( R \) = radius of absorber
- \( T \) = wall temperature of absorber
- \( q_0(r) \) = rate of volumetric heat generation (Gaussian distribution)
- \( q_0 \) = rate of heat generation per unit length
- \( \nu \) = kinematic viscosity of \( LH_2 \)
- \( \lambda \) = thermal diffusivity of \( LH_2 \)
- \( \kappa \) = thermal conductivity of \( LH_2 \)
- \( \alpha \) = coefficient of thermal expansion of \( LH_2 \)

\[ \begin{align*}
\varepsilon & = \varepsilon' \\
\kappa & = \kappa \\
\alpha & = \alpha \\
\nu & = \nu \\
\lambda & = \lambda \\
\beta & = \beta \\
\frac{\partial}{\partial t} & = \frac{\partial}{\partial t'} \\
\frac{m}{L} & = \frac{m}{L} \\
R & = R
\end{align*} \]
Governing Equations (T formulation)

Energy equation:

\[ \frac{\partial \psi}{\partial t} + v \cdot \nabla \psi = \frac{\partial}{\partial r} \left( r \frac{\partial \psi}{\partial r} \right) + \frac{\partial}{\partial z} \left( z \frac{\partial \psi}{\partial z} \right) + q(r) \]

Vorticity-transport equation:

\[ \frac{\partial \psi}{\partial t} + v \cdot \nabla \psi = \frac{\partial}{\partial r} \left( r \frac{\partial \psi}{\partial r} \right) + \frac{\partial}{\partial z} \left( z \frac{\partial \psi}{\partial z} \right) + \frac{\psi}{m \frac{\partial}{\partial \psi}} \]

Streamfunction equation:

\[ \left( \frac{\frac{\partial}{\partial r}}{\frac{\partial}{\partial \psi}} \right) \psi = \frac{\psi}{m \frac{\partial}{\partial \psi}} \]
Formulation (cont'd)

Initial and boundary conditions:

\[
\begin{align*}
\frac{\partial^2 T}{\partial t^2} &= 0 & \frac{\partial^2 \varphi}{\partial t^2} &= 0 \\
\frac{\partial T}{\partial t} &= 0 & \frac{\partial \varphi}{\partial \theta} &= 0 \\
T &\bigg|_{t=0} = T_0 & \varphi &\bigg|_{\theta=1} = \varphi_1
\end{align*}
\]

Non-dimensional variables:

\[
\begin{align*}
\frac{\varphi}{R} &= \phi & \frac{\varphi}{m} &= \phi \\
\frac{\varphi}{\gamma} &= \phi & \frac{\varphi}{\theta_0} &= \phi
\end{align*}
\]

\[
\begin{align*}
\frac{\varphi}{R} &= \phi & \frac{\varphi}{m} &= \phi \\
\frac{\varphi}{\gamma} &= \phi & \frac{\varphi}{\theta_0} &= \phi
\end{align*}
\]

Initial and boundary conditions:

\[
\begin{align*}
T &\bigg|_{t=0} = T_0 & \theta &\bigg|_{t=0} = \theta_0 \\
\end{align*}
\]
Formulation – Non-Dimensional Parameters

**Prandtl Number:**

$$Pr = \frac{v}{\alpha}$$

**Rayleigh Number:**

$$Ra = \frac{g \beta \theta L^3}{\nu \alpha} = \frac{v \alpha}{\alpha^2} \frac{g R^3}{\nu} = \frac{Ra_{MB}}{Ra_{MB}}$$

**Nusselt Number:**

$$Nu = \frac{h L}{k} = \frac{Nu_{MB}}{Nu_{MB}}$$
Results - Flow Regimes

The following flow regimes are observed:

- Steady, symmetric solutions:
  - $\text{Ra} < 1 \times 10^8$
  - $\text{Ra} \geq 1 \times 10^9$

- Unsteady, asymmetric solutions:
  - $1 \times 10^7 < \text{Ra} < 1 \times 10^8$

- Steady, symmetric results for uniform heat generation:
  - $\text{Ra} = 1.5 \times 10^7$

Streamfunction: Temperature: Vorticity:

The following flow regimes are observed:
Nusselt number versus $\theta$ for $Ra_R = 1.57 \times 10^7$ (uniform heat generation):

$Nu$ vs. $\theta$: 

Steady, Symmetric Results (cont'd)
From M. Boghosian's correlation for $Pr = 1.4$, i.e. $\frac{Nu}{MW} = 0.7041 \cdot Ra^{0.1864}$.

Results shown are from numerical simulations which compared favorably with experiments.

<table>
<thead>
<tr>
<th>Mitachi et al. (1986, 1987) - COA Code</th>
<th>FLUENT</th>
<th>$Ra_H$</th>
</tr>
</thead>
<tbody>
<tr>
<td>12.0</td>
<td>11.9</td>
<td>14.0</td>
</tr>
<tr>
<td>$7.7$</td>
<td>$7.7$</td>
<td>$8.58$</td>
</tr>
<tr>
<td>$8.2$</td>
<td>$8.2$</td>
<td>$8.2$</td>
</tr>
<tr>
<td>$1.57 \times 10^7$</td>
<td>$1.57 \times 10^6$</td>
<td>$1.57 \times 10^6$</td>
</tr>
</tbody>
</table>

Uniform heat generation ($Pr = \infty$) with $Pr = 1$: $\frac{Nu}{MW}$
Gaussian heat generation:

\[ \mathcal{L} n \times \mathcal{L} n = \frac{1}{2} N \frac{M}{W} \]

From M. Boghosian's correlations:

<table>
<thead>
<tr>
<th>( n )</th>
<th>0.0070</th>
<th>0.0039</th>
<th>0.0060</th>
<th>0.0045</th>
<th>0.1010</th>
<th>1 × 10^9</th>
</tr>
</thead>
<tbody>
<tr>
<td>46</td>
<td>0.0101</td>
<td>0.0169</td>
<td>0.0101</td>
<td>0.0101</td>
<td>0.0101</td>
<td>1 × 10^8</td>
</tr>
<tr>
<td>25.4</td>
<td>0.0111</td>
<td>0.0165</td>
<td>0.0101</td>
<td>0.0101</td>
<td>0.0101</td>
<td>1 × 10^7</td>
</tr>
<tr>
<td>15.6</td>
<td>0.0108</td>
<td>0.0100</td>
<td>0.0169</td>
<td>0.0101</td>
<td>0.0101</td>
<td>1 × 10^6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( n_{\text{max}} )</th>
<th>0.704</th>
<th>0.329</th>
<th>1.380</th>
<th>0.313</th>
<th>0.313</th>
</tr>
</thead>
<tbody>
<tr>
<td>( n_{\text{avg}} )</td>
<td>0.243</td>
<td>0.243</td>
<td>0.243</td>
<td>0.243</td>
<td>0.243</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( \text{COA Code} )</th>
<th>FLUENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>steady laminar, steady RANS (turbulent), unsteady N-S</td>
<td></td>
</tr>
</tbody>
</table>

Clausius-Clapeyron heat generation: \( \varphi = 0.25 \)

Code Comparisons - COA vs. FLUENT

Code Comparisons - COA vs. FLUENT
Steady, Symmetric Results: \( R a R = 1 \times 10^8, \phi = 0.25 \)
Steady, Symmetric Results: $Ra_R = 1 \times 10^8$, $\varphi = 0.25$
Unsteady, Asymmetric Results:

\[ \text{Ra}_R = \frac{10^{10}}{0} \]

\[ \text{Nu} \text{ vs. } \theta \]

\[ \theta \text{ vs. } n \]

\( \theta = 0.25 \)
Unsteady, Asymmetric Results: $Ra_R = 1 \times 10^{10}$, $\omega = 0.25$
Gaseous Absorber Parameters

For $dE/dx = 13$:

$M, 1.5\times10^{14}$ muons/s

$\eta_0 = 332$ W/m

Then at 100 atm and 80 K

$Ra_R = 2.01 \times 10^{12}$ for $R = 0.5$ m.

Characteristics:
- Effect of ionization and magnetic field on fluid flow and heat transfer
- Treatment of actual geometry
- Compressibility?
- Hydrogen absorber
- $Ra_R$ is one order of magnitude higher than in the case of liquid transfer problem:
  More complex and time-consuming to solve the fluid flow and heat + No boiling!

Characteristics:
- Treatment of actual geometry
- Compressibility?
- Hydrogen absorber
- $Ra_R$ is one order of magnitude higher than in the case of liquid transfer problem:
  More complex and time-consuming to solve the fluid flow and heat + No boiling!
Conclusions

Current COA results compare very well with limited experimental data and FLUENT results (both laminar and turbulent regimes).

Critical Rayleigh number for unsteady, asymmetric behavior is

\[ Ra > 1 \times 10^8 \]  

Roughly corresponds to laminar to turbulent transition in FLUENT results.

No start-up overshoot in temperature at high \( Ra \).

Heater not necessary to improve performance of absorber as heat exchanger.

CFD results offer guidance for gaseous absorber (additional issues must be addressed).
Proposed Flow Test

Wishlist:

1. Near room temperature flow test: maximize possible; minimize cost. (may be difficult depending on heat source)
2. Working fluid that is safe and easy to work with.
3. Allow for flexibility in providing heat source.
4. Maximize information obtained without need for internal measurements.
5. Provide for comparisons of essential data with CFD results.

If such measurements are possible, all the better.
In a typical test one would choose the geometry, working fluid and heat input to give a particular Rayleigh number. Then the temperature (e.g. maximum temperature) and flow conditions would be measured.

The key insight:

Choose the Rayleigh number and determine $L \nabla$.

Choose the Rayleigh number and determine $L \nabla$.

We can measure temperature change by heating from a known wall temperature to boiling, i.e., $T_{\text{w}} = T_{\text{boil}}$.

In the proposed test, the geometry, working fluid and temperature range are chosen, and the required heat input is determined. Choose the Rayleigh number and determine $T$. The key insight:

Choose the Rayleigh number and determine $L \nabla$. In the proposed test, the geometry, working fluid and heat input are chosen, and the required heat input is determined.

Choose the Rayleigh number and determine $L \nabla$. In the proposed test, the geometry, working fluid and heat input are chosen, and the required heat input is determined.
Proposed Flow Test (cont'd)

Features:

- Setup: absorber encased in coolingsheath (similar to actual absorber).
- Heat source: electric current in absorber fluid, beam, etc.
- Absorber fluid: water is a candidate. Could possibly use additive to increase electrical conductivity and/or lower boiling point.
Parameter Map for Water

Note: Properties taken at 100°C, 1 atm.
Proposed Flow Test (cont'd)

Procedure:

1. Choose $T_w = T_{boil}$. 

2. Circulate coolant until absorber fluid reaches uniform temperature equal $\nabla L - \nabla qL = mL$.

3. Turn on heat source and increase in a quasi-steady manner, i.e. slowly, until incipient boiling occurs.

4. Record video to note location of incipient boiling and visualize flow using bubbles.

5. Determine heat output from absorber by measuring mass flow rate and inlet/outlet temperatures of coolant.

6. At conclusion of test, drain fluid from absorber and determine bulk, i.e. average, temperature of absorber, $T_{avg}$.

7. Run test for a series of $T_w$'s.
Proposed Flow Test (cont'd)

Analysis of ow-test results:

1. Determine actual Rayleigh number from magnitude of heat input, i.e. selected $T_{\text{boil}} = T_{\text{w}}$.

2. Determine heat input required to produce temperature rise

3. Compare actual heat input required for boiling with that predicted from CFD, i.e. compare actual and predicted Rayleigh numbers for given $T_{\text{w}}$.

4. Estimate average Nusselt number using actual heat input, heat transfer surface area and $T_{\text{avg}}$.

5. Compare estimated Nusselt number from ow-test with predicted value from CFD.

\[ n \frac{L}{L} - {}^{\text{avg}} \frac{L}{L} = {}^{\text{L}} \frac{L}{L} \text{ necessary to produce boiling, i.e. selected Rayleigh number from test} \]
Proposed Flow Test (cont'd)

Features of ow-test:

Choose

T * rather than Rayleigh number for each test.

No exotic fluid flow or temperature measurements necessary.

We measure the maximum temperature visually by heating until boiling occurs.

The fluid may be heated in the most practical manner without regard for its effect on measurement techniques.

The bubbles provide some limited visualization capability.

Measuring maximum temperature is a quantity that is influenced strongly by both fluid dynamic and heat transfer aspects, i.e. it is a composite of the entire fluid flow and heat transfer environment.

We measure visually by heating until boiling occurs.

No exotic fluid flow or temperature measurements necessary.

Choose T * rather than Rayleigh number for each test.

Features of ow-test: