Absorber Simulations Update

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Introduction: Approaches to Heat Removal

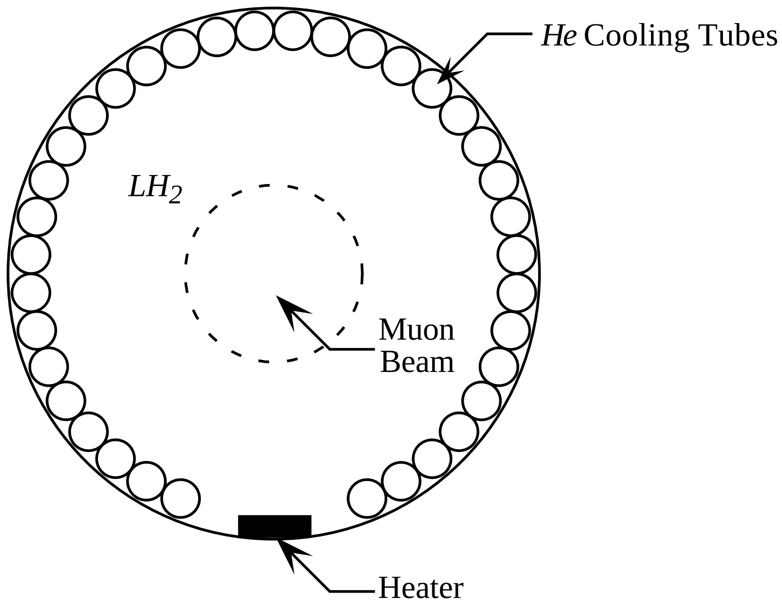
Two approaches under consideration:

① External cooling loop (traditional approach).

☞ Bring the LH_2 to the coolant (heat removed in an external heat exchanger).

② Combined absorber and heat exchanger.

☞ Bring the coolant, i.e. He, to the LH_2 (remove heat directly within absorber).



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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 LH_2.

$+$ Is likely to work best for small aspect ratio (L/R) absorbers.

$-$ May be difficult to maintain uniform vertical flow through the absorber.

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
$\Rightarrow$ more turbulence $\Rightarrow$ enhanced 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.

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Heat Exchanger Analysis

Energy balance between LH_2 and coolant (He).

✓ Parameters:

T_i	=	coolant inlet temperature
T_o	=	coolant outlet temperature
T_{LH_2}	=	bulk temperature of LH_2
A	=	surface area of cooling tubes
h_{LH_2}	=	convective heat transfer coefficient of LH_2
h_{He}	=	convective heat transfer coefficient of He
Δx	=	thickness of cooling tube walls
k_w	=	thermal conductivity of cooling tube walls
c_p	=	specific heat capacity of He

Heat Exchanger Analysis (cont'd)

- ✓ Rate of heat transfer:

$$\dot{q} = -\frac{A(T_o - T_i)}{\left(\frac{1}{h_{LH_2}} + \frac{\Delta x}{k_w} + \frac{1}{h_{He}}\right) \ln\left(\frac{T_{LH_2} - T_o}{T_{LH_2} - T_i}\right)}$$

- ✓ Mass flow rate of He:

$$\dot{m}_{He} = \frac{\dot{q}}{c_p \left(T_o - T_i\right)}.$$

$h_{He} \Rightarrow$ from appropriate correlation (flow through a tube).

h_{LH_2} and $T_{LH_2} \Rightarrow$ from CFD simulations (no correlations for natural convection with heat generation).

Computational Fluid Dynamics (CFD)

Features of the CFD Simulations:

✓ Provides average convective heat transfer coefficient and average LH_2 temperature for heat exchanger analysis.

✓ Track maximum LH_2 temperature (*cf.* boiling point).

✓ Determine details of fluid flow and heat transfer in absorber.

$\Rightarrow$ *Better understanding leads to better design!*

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CFD (cont'd)

Take 1: Results using FLUENT (M. Boghosian):

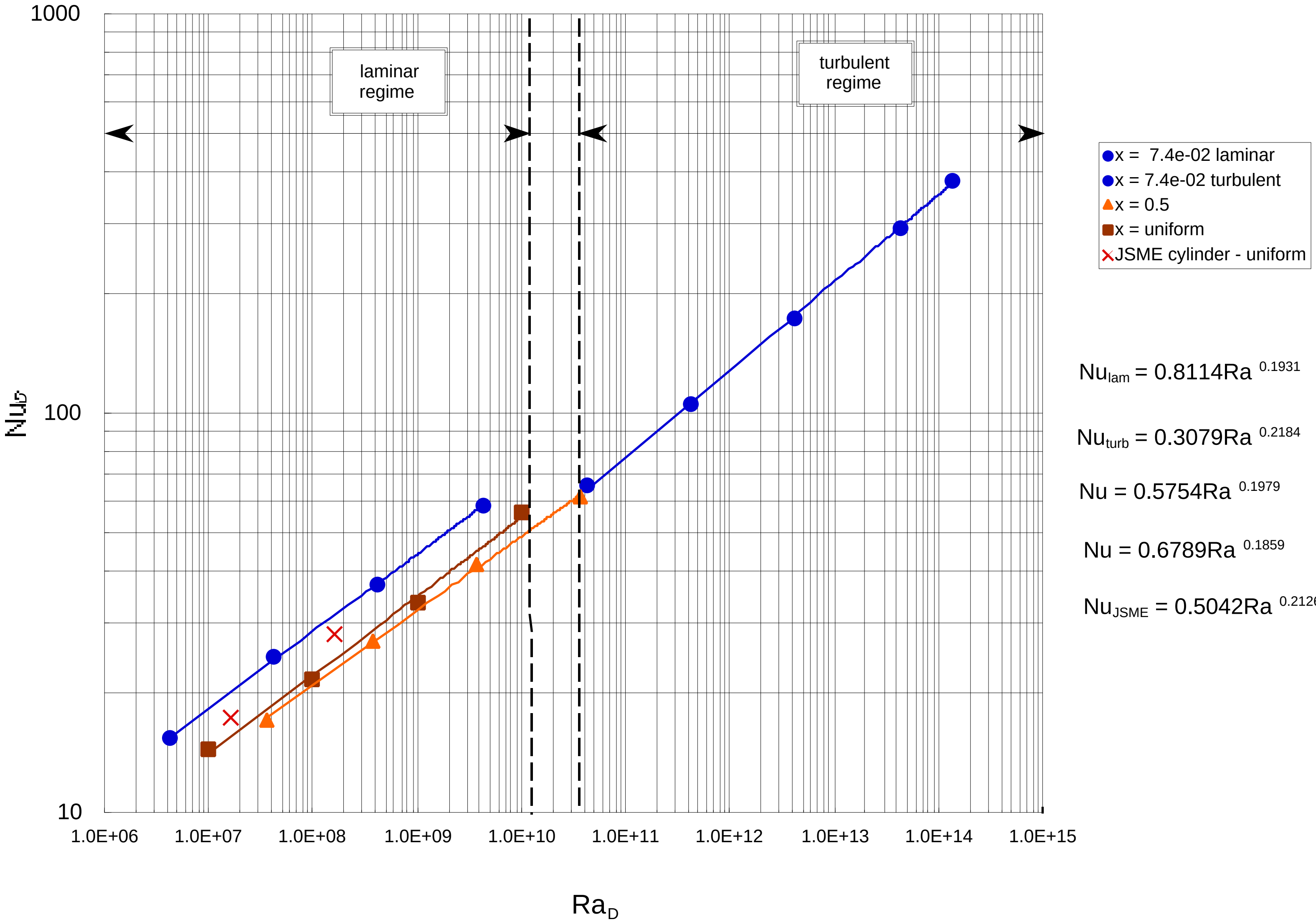
- ✓ Simulate one half of symmetric domain.
- ✓ Steady flow calculations.
- ✓ Heat generation via *steady* Gaussian distribution.
- ✓ Turbulence modeling (RANS) used for $Ra_R \geq 4 \times 10^8$.

Take 2: Results using COA code (A. Obabko and E. Almasri):

- ✓ Simulate full domain.
- ✓ Unsteady flow calculations.
- ✓ All scales computed for all Rayleigh numbers.
  - ➥ Investigate startup behavior, *e.g.* startup overshoot in T_{max}.
  - ➥ Investigate possibility of asymmetric flow oscillations.
  - ➥ Investigate influence of beam pulsing.

FLUENT CFD Results

Average Nusselt Number vs. Rayleigh Number:



$Nu_{lam} = 0.8114Ra^{0.1931}$

$Nu_{turb} = 0.3079Ra^{0.2184}$

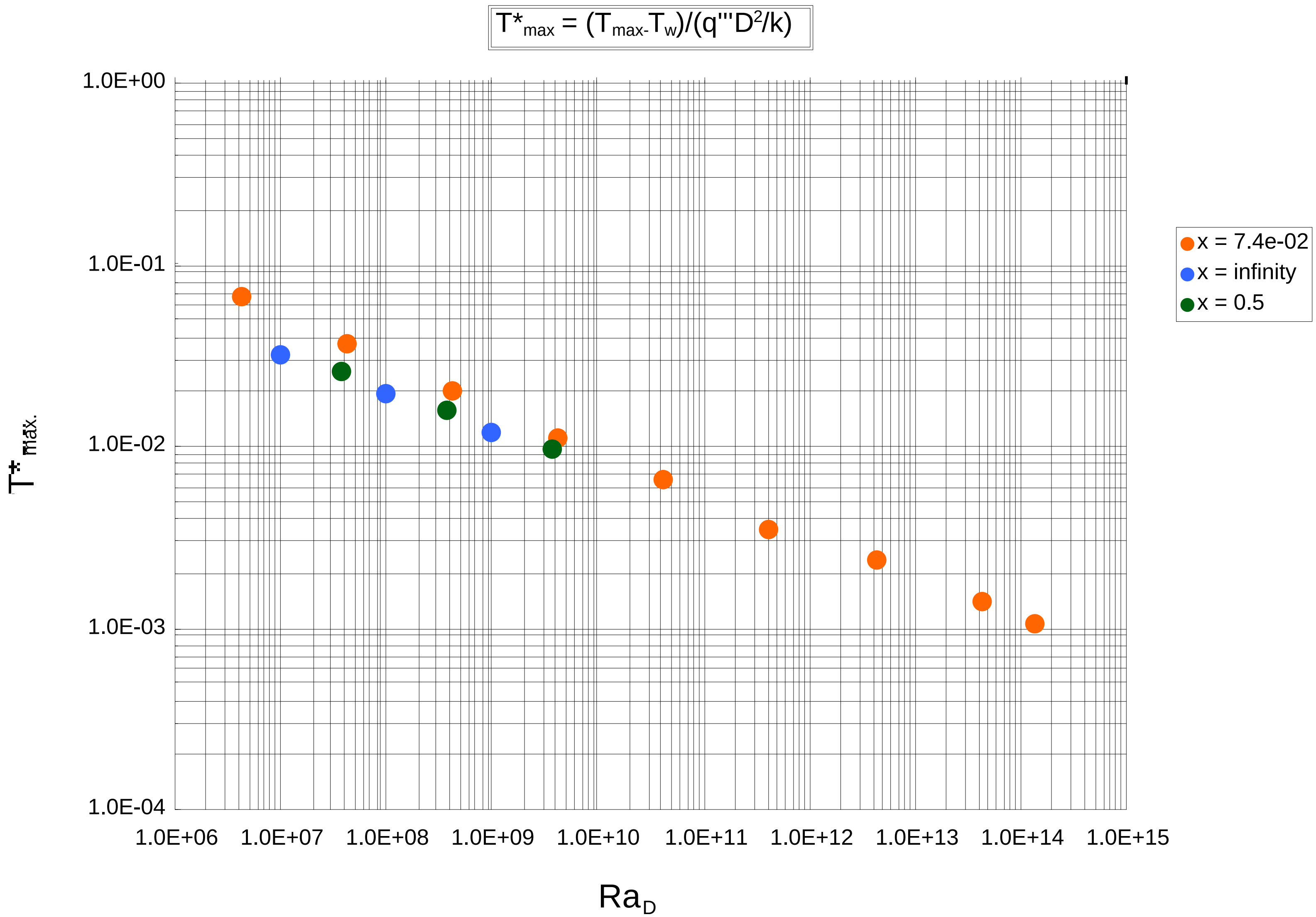
$Nu = 0.5754Ra^{0.1979}$

$Nu = 0.6789Ra^{0.1859}$

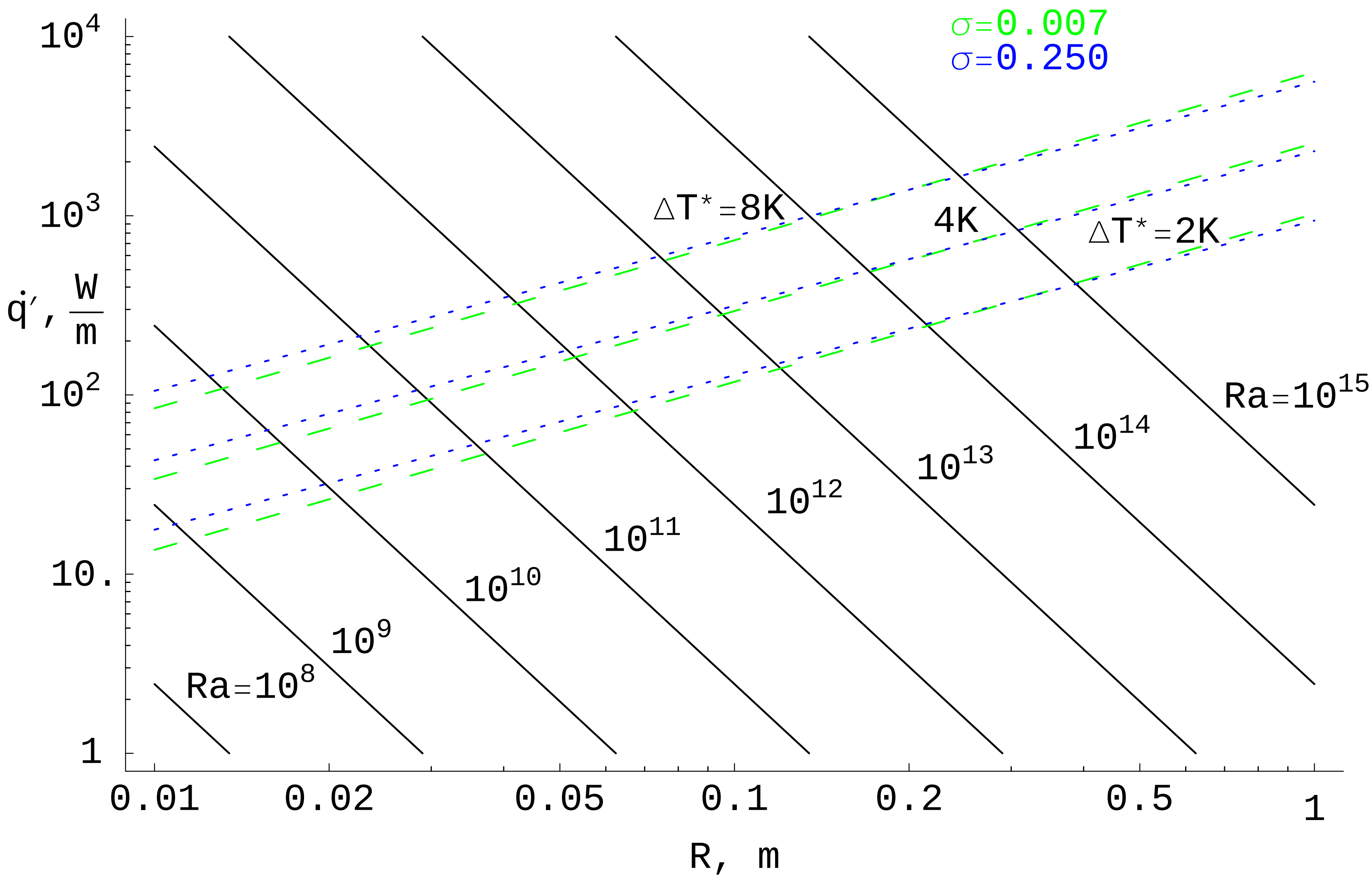
$Nu_{JSME} = 0.5042Ra^{0.2126}$

FLUENT CFD Results (cont'd)

Non-Dimensional Maximum Temperature vs. Rayleigh Number:



Parameter Map for LH_2



Note: Properties taken at 18 K, 2 atm.

Sample Heat Exchanger Analysis

Absorber parameters (single-flip lattice):

$$L = 0.3\,\text{m}, \quad R = 0.2\,\text{m}, \quad \dot{q} = 150\,\text{W} \quad \Rightarrow \quad Ra_R = 1.64 \times 10^{14}$$

Heat exchanger parameters (LH_2 and He at 2 atm):

$$\begin{aligned}
T_i^* &= 14\text{ K} \\
T_o^* &= 15\text{ K} \\
T_{LH_2}^* &= 18.5\text{ K (from CFD results)} \\
T_{max}^* &= 18.9\text{ K (from CFD results)} \\
h_{He} &= 1{,}580\text{ W}/\text{m}^2\text{K} \\
h_{LH_2} &= 210\text{ W}/\text{m}^2\text{K (from CFD results)}
\end{aligned}$$

Results:

Required heat transfer area: $A = 0.20\text{m}^2$

Mass flow rate of He: $\dot{m}_{He} = 0.028$ kg/s (3.9 l/s)

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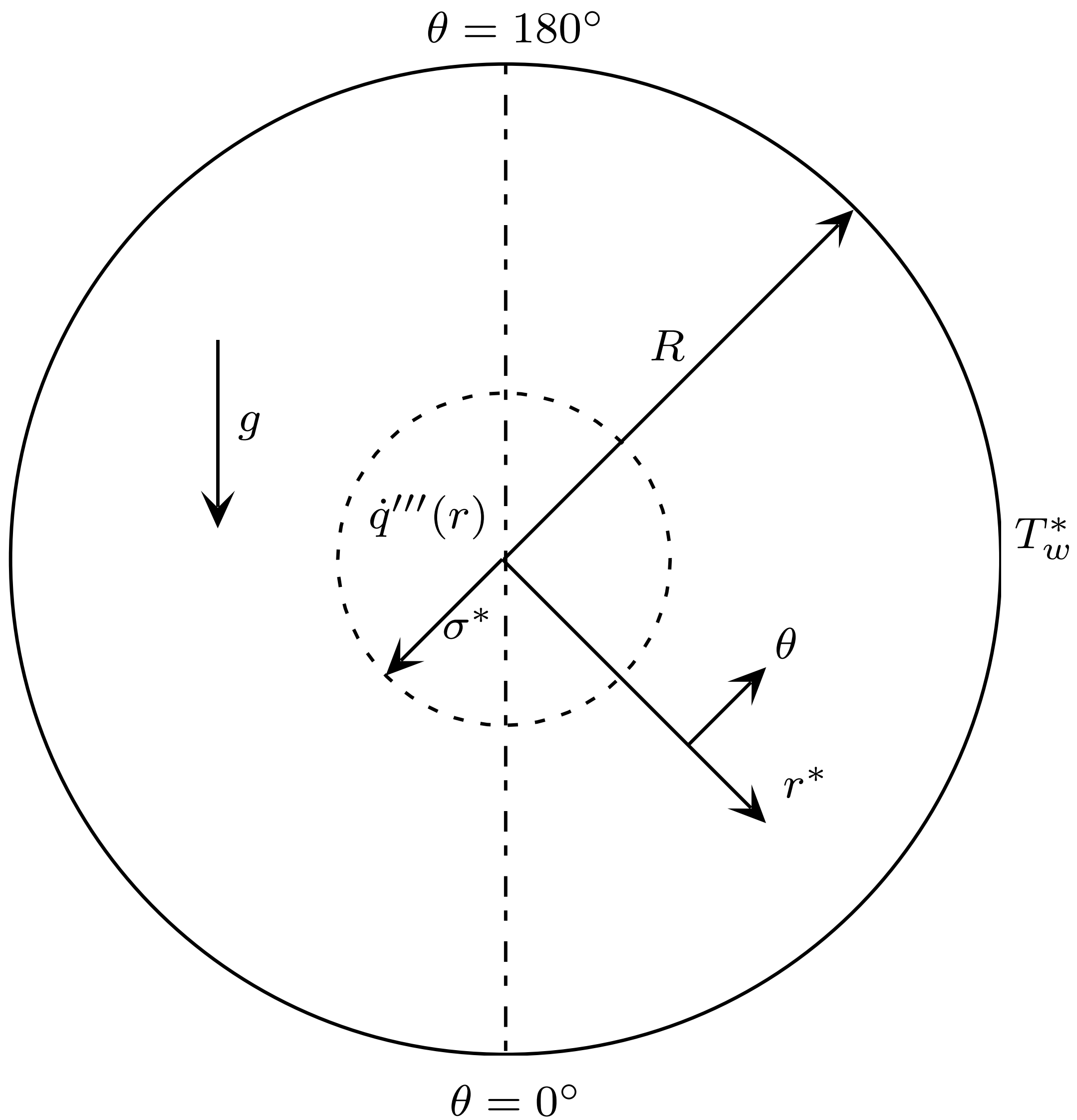
Schematic



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COA Formulation

Properties and parameters:

R	=	radius of absorber
T_w	=	wall temperature of absorber
$\dot{q}'''(r)$	=	rate of volumetric heat generation (Gaussian distribution)
$\dot{q}'$	=	rate of heat generation per unit length
σ^*	=	standard deviation of heat generation Gaussian distribution
ν	=	kinematic viscosity of LH_2
α	=	thermal diffusivity of LH_2
k	=	thermal conductivity of LH_2
β	=	coefficient of thermal expansion of LH_2
g	=	acceleration due to gravity

Formulation (cont'd)

Non-dimensional variables:

$$r = \frac{r^*}{R}, \quad v_r = \frac{v_r^*}{\alpha/R}, \quad v_\theta = \frac{v_\theta^*}{\alpha/R}, \quad t = \frac{t^*}{R^2/\alpha},$$

$$T = \frac{T^* - T_w}{\dot{q}'/k}, \quad \psi = \frac{\psi^*}{\alpha}, \quad \omega = \frac{\omega^*}{\alpha/R^2},$$

$$q(r) = \frac{\dot{q}'''(r)}{\dot{q}'/R^2} = \frac{1}{2\pi\sigma^2} e^{-\frac{r^2}{2\sigma^2}}, \quad \sigma = \frac{\sigma^*}{R}.$$

Initial and boundary conditions:

$$T = \omega = \psi = v_r = v_\theta = 0 \quad \text{at} \quad t = 0,$$

$$T = \psi = v_r = v_\theta = 0 \quad \text{at} \quad r = 1.$$

Governing Equations (T - ω - ψ formulation)

Energy equation:

$$\frac{\partial T}{\partial t} + v_r \frac{\partial T}{\partial r} + \frac{v_\theta}{r}\frac{\partial T}{\partial \theta} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + q(r)$$

Vorticity-transport equation:

$$\begin{aligned}\frac{\partial \omega}{\partial t} + v_r \frac{\partial \omega}{\partial r} + \frac{v_\theta}{r}\frac{\partial \omega}{\partial \theta} &= Pr\left[\frac{\partial^2 \omega}{\partial r^2} + \frac{1}{r}\frac{\partial \omega}{\partial r} + \frac{1}{r^2}\frac{\partial^2 \omega}{\partial \theta^2}\right] \\ &\quad + Ra_R Pr\left[\sin\theta \frac{\partial T}{\partial r} + \frac{\cos\theta}{r}\frac{\partial T}{\partial \theta}\right]\end{aligned}$$

Streamfunction equation:

$$\frac{\partial^2 \psi}{\partial r^2} + \frac{1}{r}\frac{\partial \psi}{\partial r} + \frac{1}{r^2}\frac{\partial^2 \psi}{\partial \theta^2} = -\omega$$

$$v_r = \frac{1}{r}\frac{\partial \psi}{\partial \theta}, \quad v_\theta = -\frac{\partial \psi}{\partial r}$$

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Formulation – Non-Dimensional Parameters

Prandtl Number:

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

Rayleigh Number:

$$Ra_R = Gr\,Pr = \frac{gR^3\beta\dot{q}'/k}{\nu\alpha}\left(= \frac{\pi}{32}Ra_{MB}\right)$$

Nusselt number $=$ nondimensional convective heat transfer coefficient, h_{LH_2}:

$$Nu_R = \frac{h_{LH_2}R}{k}\left(= \frac{Nu_{MB}}{2}\right)$$

$\overline{Nu}$ = Nusselt number averaged over inner surface of cylinder.

$\langle\overline{Nu}\rangle$ = Nusselt number averaged over time interval.

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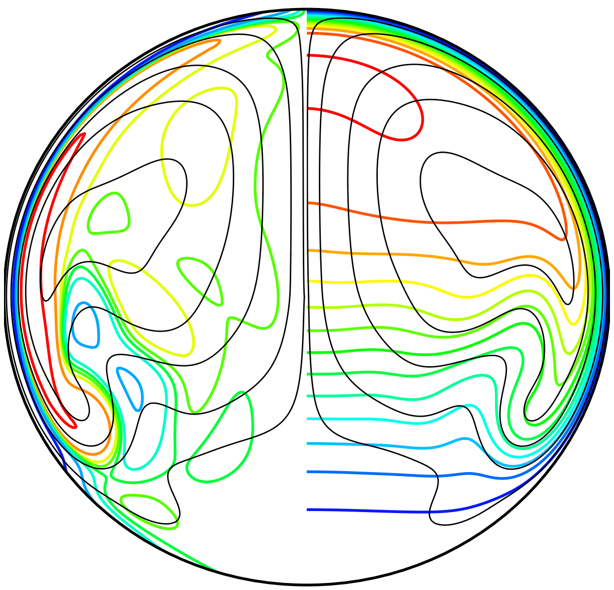
Results – Flow Regimes

The following flow regimes are observed:

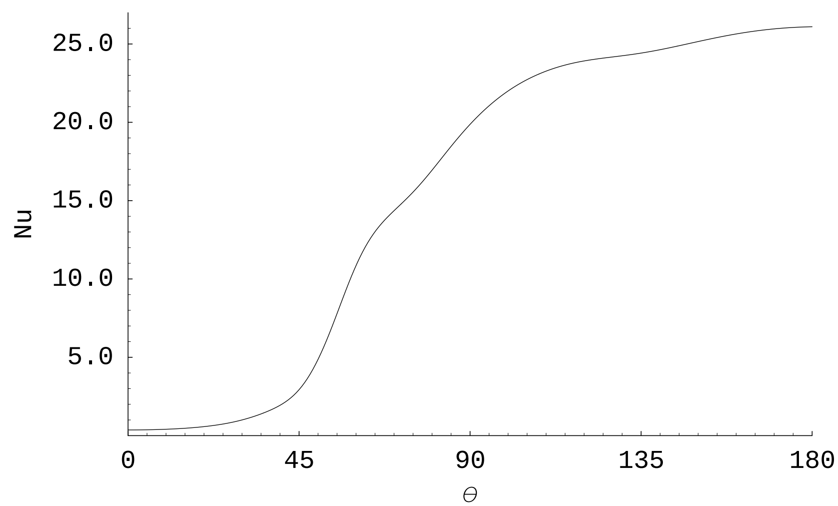
- Steady, symmetric solutions: $Ra_R \leq 1 \times 10^8$
- Unsteady, asymmetric solutions: $Ra_R > 1 \times 10^8$

Steady, symmetric results for $Ra_R = 10^8, \sigma = 0.25$:

ψ and ω: ψ and T:

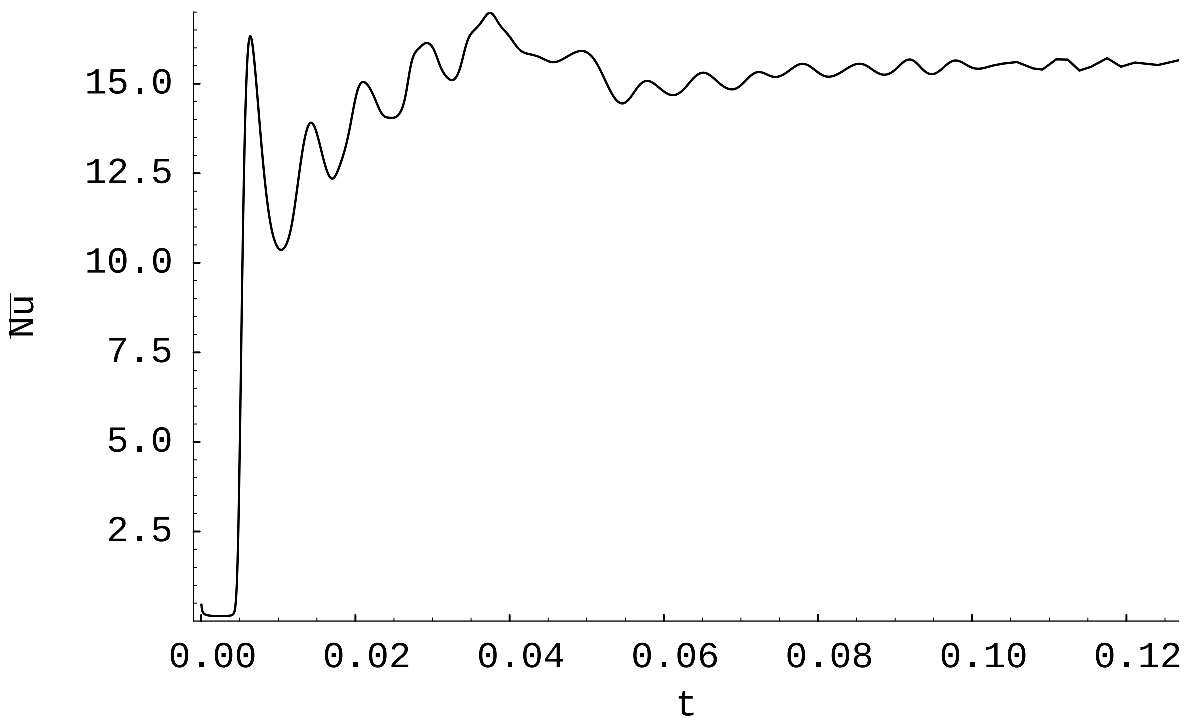


Nu vs. θ:



Results: $Ra_R = 10^8, \sigma = 0.25$

$\overline{Nu}$ vs. t:



Code Comparisons – Average Nusselt Number ($\overline{Nu}$)

Uniform heat generation ($\sigma \longrightarrow \infty$) with $Pr = 1$:

Ra_R	Mitachi *et al.* [1]	FLUENT[2]	COA Code
1.57×10^6	8.58	7.7	8.23
1.57×10^7	14.0	11.9	12.0

[1] Mitachi *et al.* (1986, 1987) - Results shown are from numerical simulations which compared favorably with experiments.

[2] From M. Boghosian's correlation for $Pr = 1.4$, *i.e.* $\overline{Nu}_{MB} = 0.7041 \cdot Ra_{MB}^{0.1864}$.

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Code Comparisons – COA vs. FLUENT

Gaussian heat generation: $\sigma = 0.25$

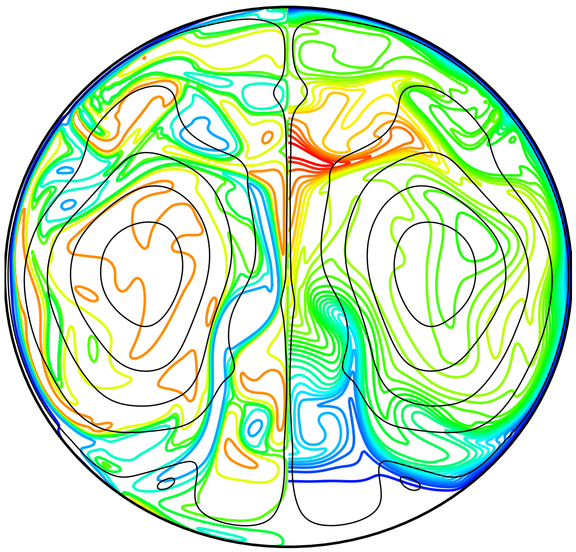
steady laminar, steady RANS (turbulent), unsteady N–S

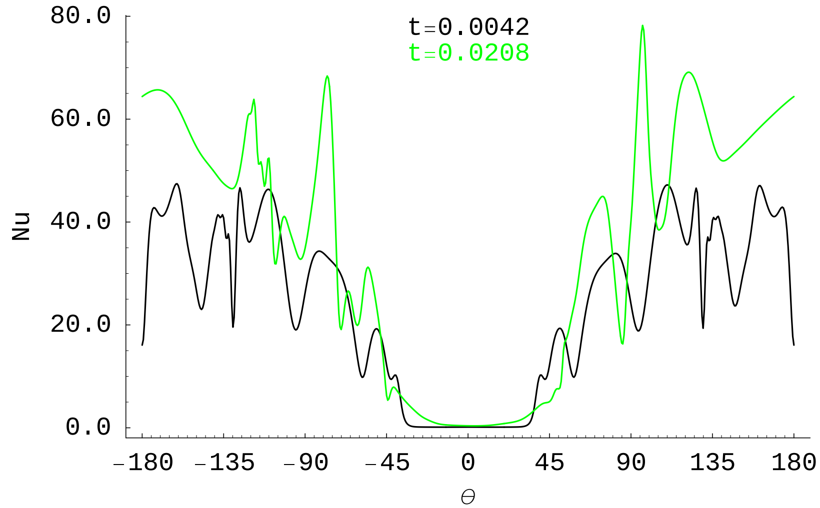
	FLUENT[1]			COA Code		
Ra_R	T_{avg}	T_{max}	$\overline{Nu}$	$\langle T_{avg} \rangle$	$\langle T_{max} \rangle$	$\langle \overline{Nu} \rangle$
1×10^8	0.0101	0.0169	16.4	0.0100	0.018	15.6
1×10^9	0.0067	0.0101	25.1	0.0055	0.0084	23.7
1×10^{10}	0.0045	0.0060	38.5	0.0038	0.0059	38.4

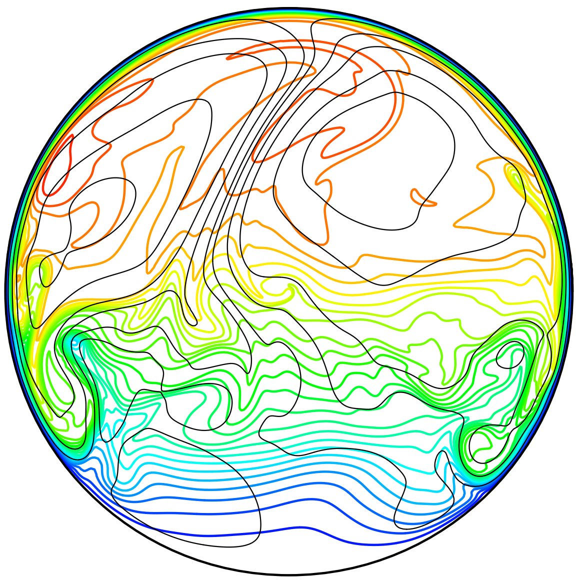
[1] From M. Boghosian's correlations $(T_{MB} = \frac{\pi}{4} T)$:

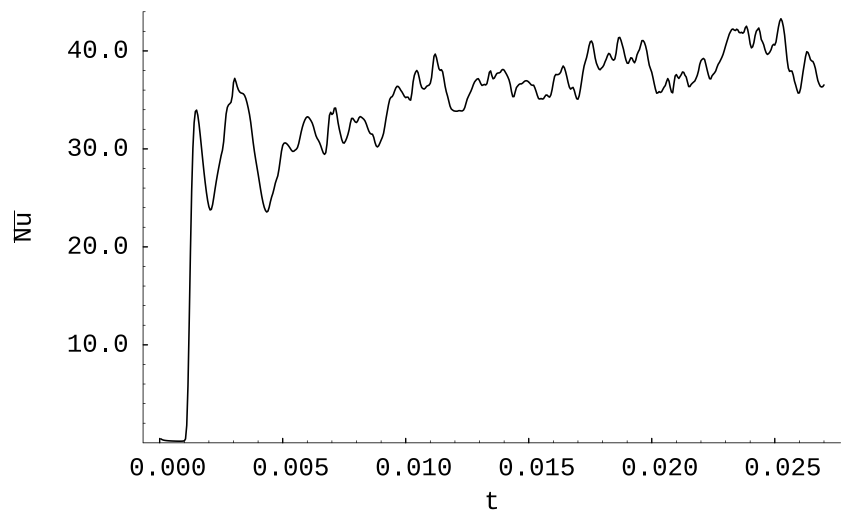
$$T_{avg_{MB}} = 0.3130 \cdot Ra_{MB}^{-0.1771}, \quad T_{max_{MB}} = 1.3597 \cdot Ra_{MB}^{-0.2233}, \quad \overline{Nu}_{MB} = 0.7041 \cdot Ra_{MB}^{0.1852}$$

Unsteady, Asymmetric Results: $Ra_R = 10^{10}, \sigma = 0.25$





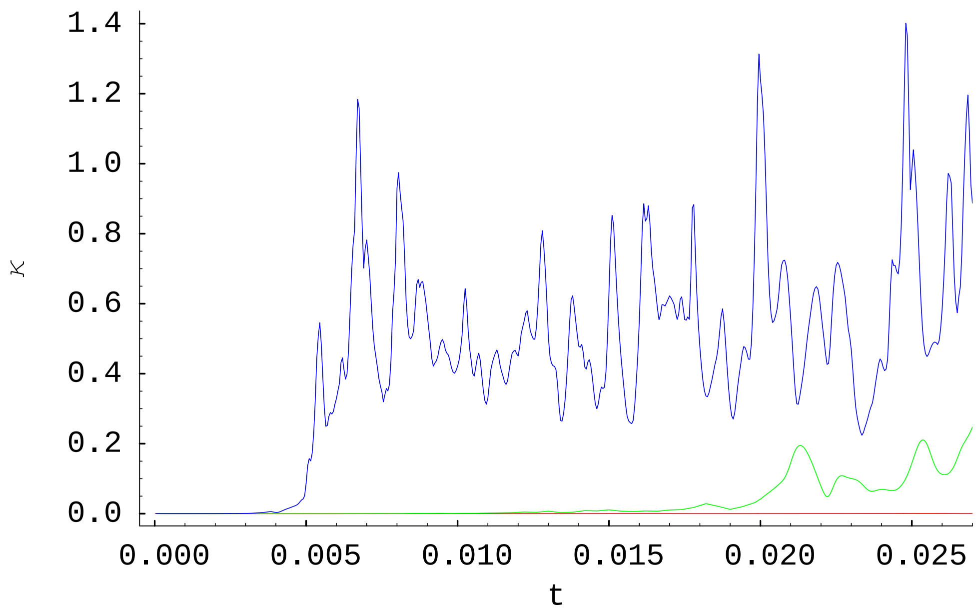




Unsteady, Asymmetric Results

Asymmetry parameter (normalized kinetic energy per unit mass crossing vertical symmetry line, *i.e.* $-1 \leq y \leq 1$):

$$\kappa = \left(\frac{\int_{-1}^{1} v_\theta^2 \, dy}{\int_{-1}^{1} v_r^2 \, dy} \right)^{1/2}$$



$Ra_R = 10^8$, $Ra_R = 10^9$, $Ra_R = 10^{10}$

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Gaseous Absorber Parameters

For $dE/dx = 13.81$ MeV, 1.5×10^{14} muons/s $\Rightarrow \dot{q}' = 332$ W/m.

Then at 100 atm and 80 K $\Rightarrow Ra_R = 2.01 \times 10^{15}$ for $R = 0.5$ m.

Characteristics:

$+$ No boiling!

$-$ More complex and time-consuming to solve the fluid flow and heat transfer problem:

- ☞ Ra_R is one order of magnitude higher than in the case of liquid hydrogen absorber.
- ☞ Compressibility?

? Treatment of actual geometry.

? Effect of ionization and magnetic field on fluid flow and heat transfer characteristics.

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 $Ra_R > 1 \times 10^8$.

  $\Rightarrow$ Roughly corresponds to laminar to turbulent transition in FLUENT results.
- No start-up overshoot in temperature at high Ra.

  $\Rightarrow$ Heater not necessary to improve performance of absorber as heat exchanger.
- CFD results offer guidance for gaseous absorber (additional issues must be addressed).

Current and Future Efforts

- Obtain solutions at higher Rayleigh numbers (target $Ra_R \sim 10^{14}$).
  - Incorporate grid transformation in COA code.
- Compare high-Rayleigh number COA solutions (unsteady) with FLUENT results (steady RANS).
- Evaluate influence of σ, *i.e.* ratio of beam size to absorber size, on heat transfer.
- Investigate influence of pulsed beam on fluid dynamics and heat transfer.
  - Note that at 15 Hz, one pulse corresponds to 2.4×10^{-7} non-dimensional time units (*cf.* $\Delta t = 10^{-8}$).
- Comparisons of CFD predictions with flow tests:
  - J. Norem's beam tests at Argonne.
  - MTA test of KEK absorber with temperature probes.

Proposed Flow Test

Wish list:

- ✓ Near room temperature flow test $\Rightarrow$ minimize cost; maximize possible sites for test.
- ✓ Working fluid that is safe and easy to work with.
- ✓ Allow for flexibility in providing heat source.
- ✓ Maximize information obtained without need for internal measurements (may be difficult depending on heat source).
  - $\Rightarrow$ If such measurements are possible, all the better.
- ✓ Provide for comparisons of essential data with CFD results.

Proposed Flow Test (cont'd)

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.

$\Rightarrow$ Choose the Rayleigh number and determine ΔT^*.

The key insight:

☞ We can *measure* temperature change by heating from a known wall temperature to boiling, i.e. $\Delta T^* = T^*_{boil} - T^*_w$.

$\Rightarrow$ In the proposed test, the geometry, working fluid and temperature range are chosen, and the required heat input is determined.

$\Rightarrow$ Choose the ΔT^* and determine Rayleigh number.

Proposed Flow Test (cont'd)

Features:

- ✓ Set up: absorber encased in cooling sheath (similar to actual absorber).
- ✓ Heat source: electric current in absorber fluid, beam, etc.
- ✓ Absorber fluid: water is a candidate.
  - $\longrightarrow$ Could possibly use additive to increase electrical conductivity and/or lower boiling point.

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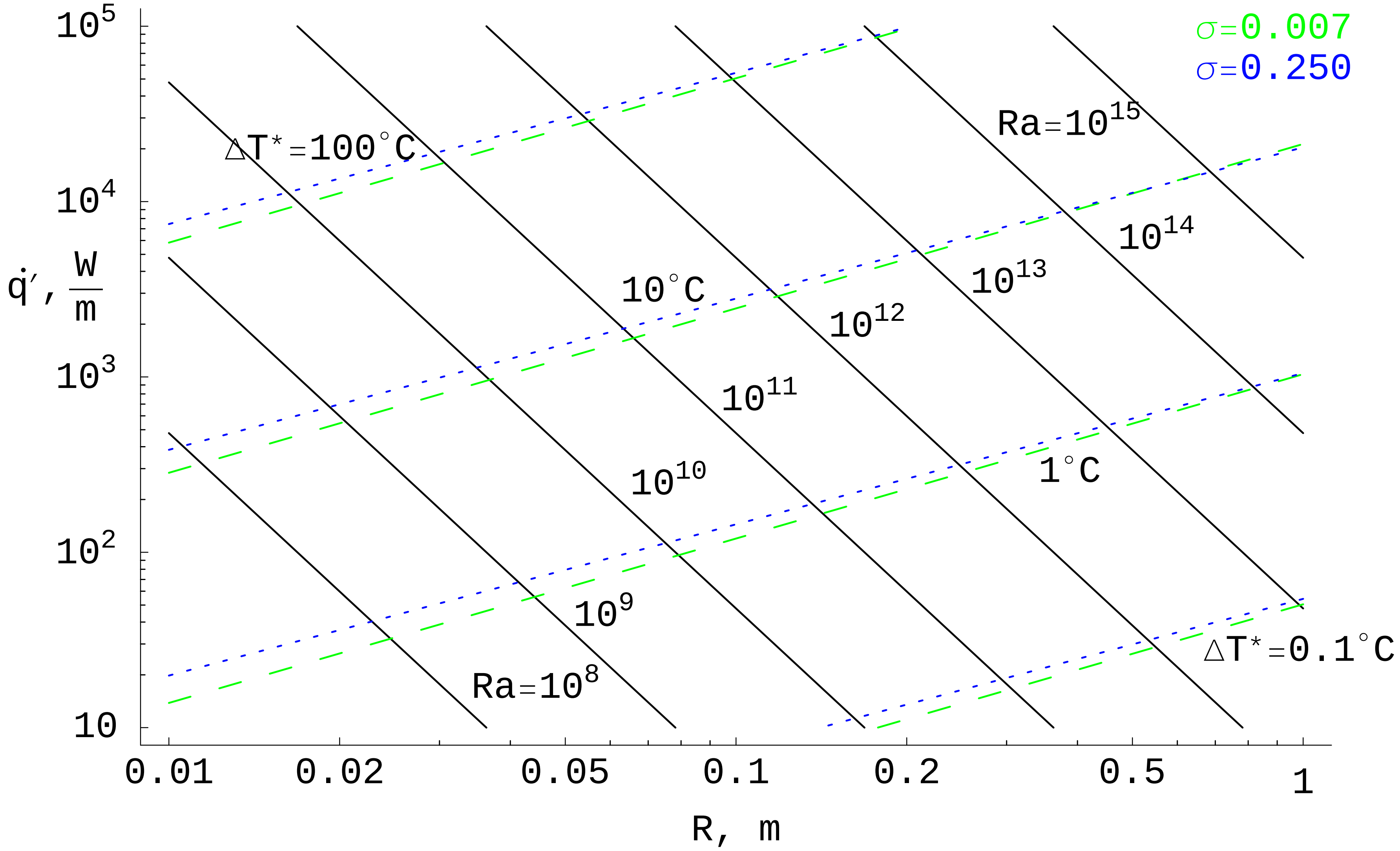
Parameter Map for Water



Note: Properties taken at 100°C, 1 atm.

Proposed Flow Test (cont'd)

Procedure:

① Choose ΔT^* $\Rightarrow$ Absorber wall temperature $T_w^* = T_{boil}^* - \Delta T^*$.

② Circulate coolant until absorber fluid reaches uniform temperature equal to T_w^*.

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, T_{avg}^*, of absorber fluid, i.e. drain at constant, known mass flow rate and measure time series of temperature of draining fluid.

⑦ Run test for a series of ΔT^*'s.

Proposed Flow Test (cont'd)

Analysis of flow-test results:

① Determine actual Rayleigh number of test from magnitude of heat input necessary to produce boiling, i.e. selected $\Delta T^* = T^*_{boil} - T^*_w$.

② Determine heat input predicted from CFD to produce temperature rise corresponding to ΔT^*.

③ Compare actual heat input required for boiling with that predicted from CFD, i.e. compare actual and predicted Rayleigh numbers for given ΔT^*.

④ Estimate average Nusselt number using actual heat input, heat transfer surface area and $T^*_{avg} - T^*_w$.

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

Proposed Flow Test (cont'd)

Features of flow-test:

- ✓ Choose ΔT^* rather than Rayleigh number for each test.
- ✓ No exotic fluid flow or temperature measurements necessary.

  $\longrightarrow$ We *measure* the maximum temperature visually by heating until boiling occurs.

  $\Rightarrow$ 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.