LH_2 Absorbers – Simulations and Proposed Flow Test

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Introduction: Approaches to Heat Removal

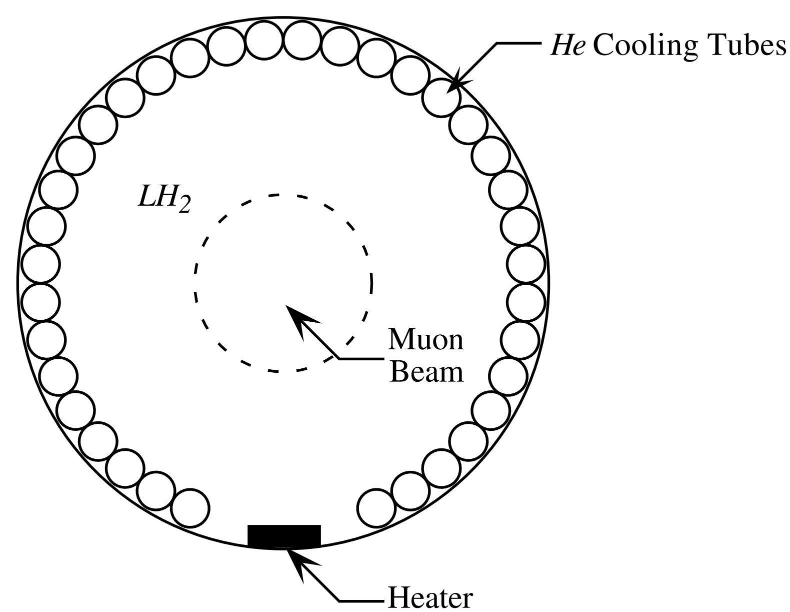
Two approaches under consideration:

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.

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:

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

$h_{He} \Rightarrow$ 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!*

CFD (cont'd)

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

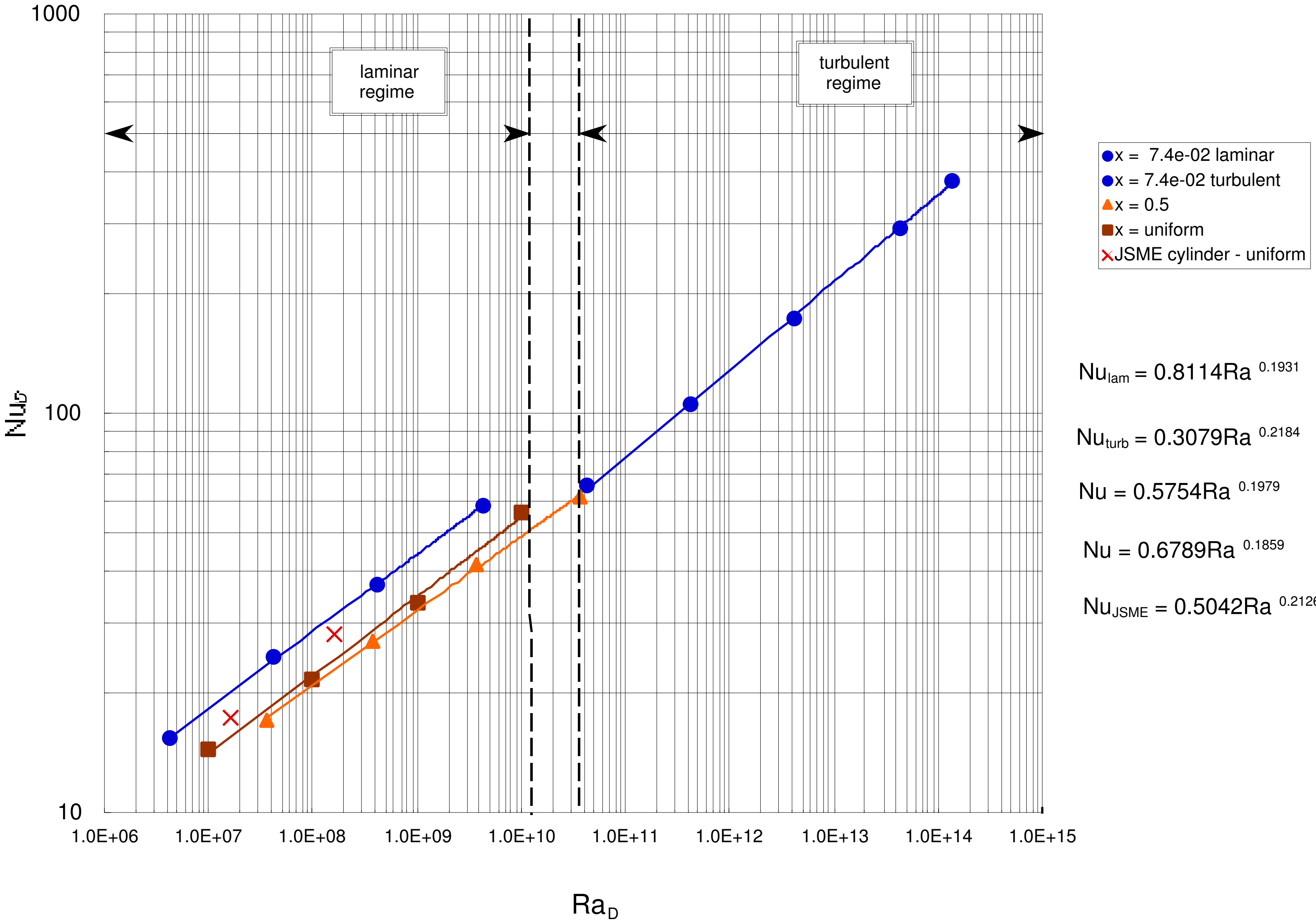
- ✓ Simulate one half of symmetric domain.
- ✓ Steady flow calculations.
- ✓ Heat generation via *steady* Gaussian distribution.
- ✓ Turbulence modeling (RANS) used for $Ra_R \geq 4 \times 10^8$.

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:



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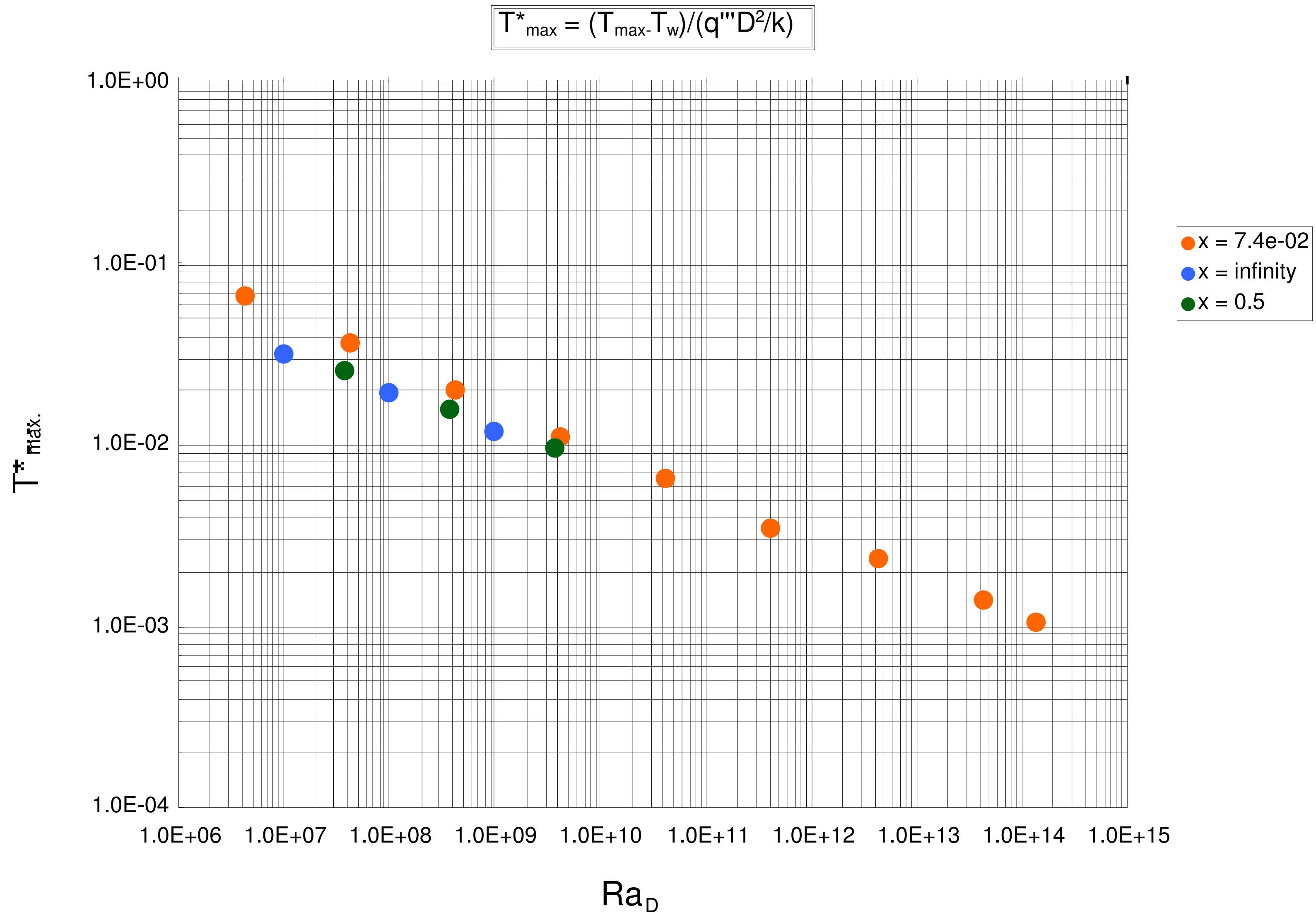
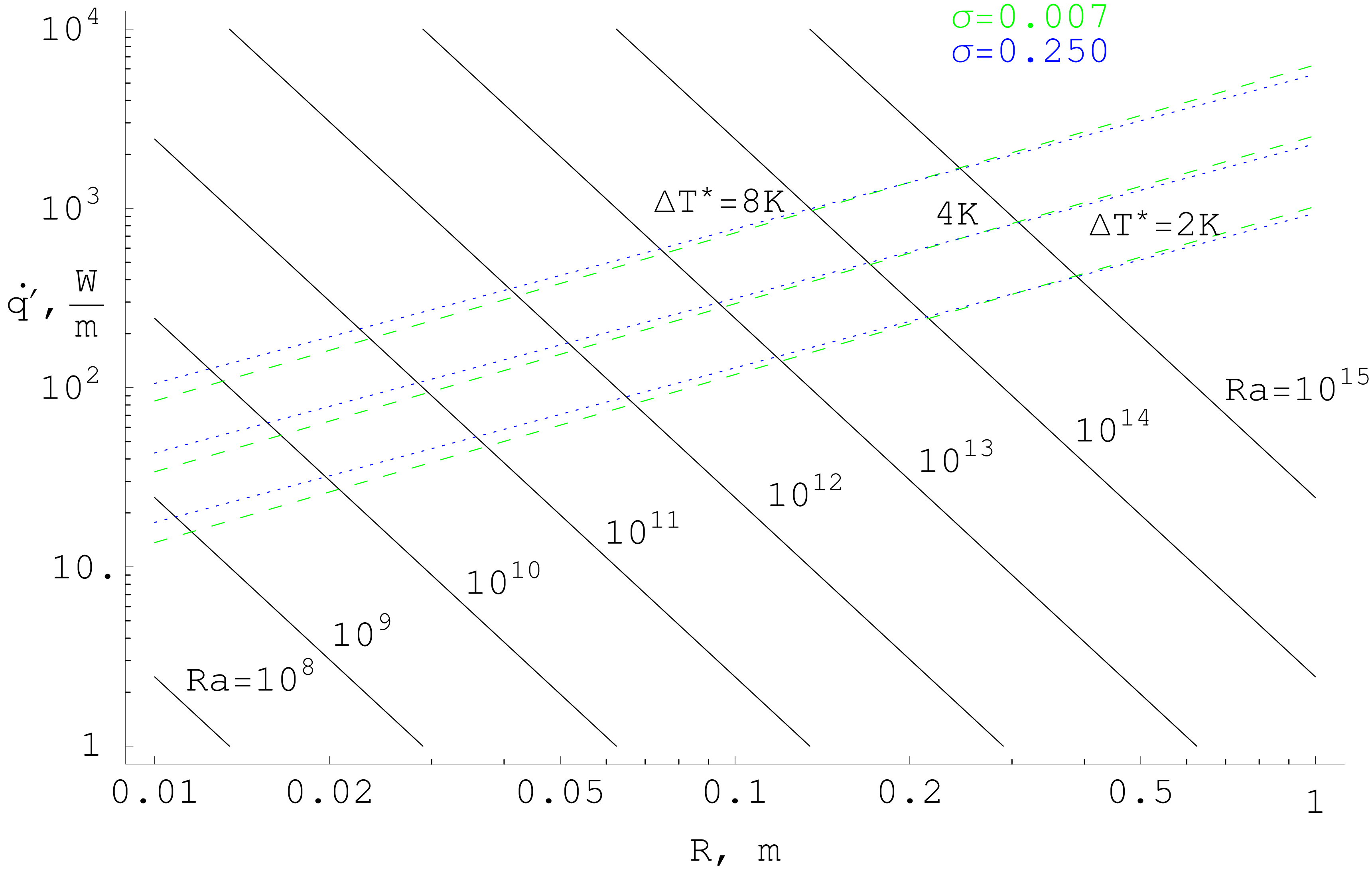
FLUENT CFD Results (cont'd)

Non-Dimensional Maximum Temperature vs. Rayleigh Number:



Parameter Map for LH_2



Note: Properties taken at 18 K, 2 atm.

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
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

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:

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

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

Formulation (cont'd)

Initial and boundary conditions:

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

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

Non-dimensional variables:

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

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

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

Formulation – Non-Dimensional Parameters

Prandtl Number:

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

Rayleigh Number:

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

Nusselt number:

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

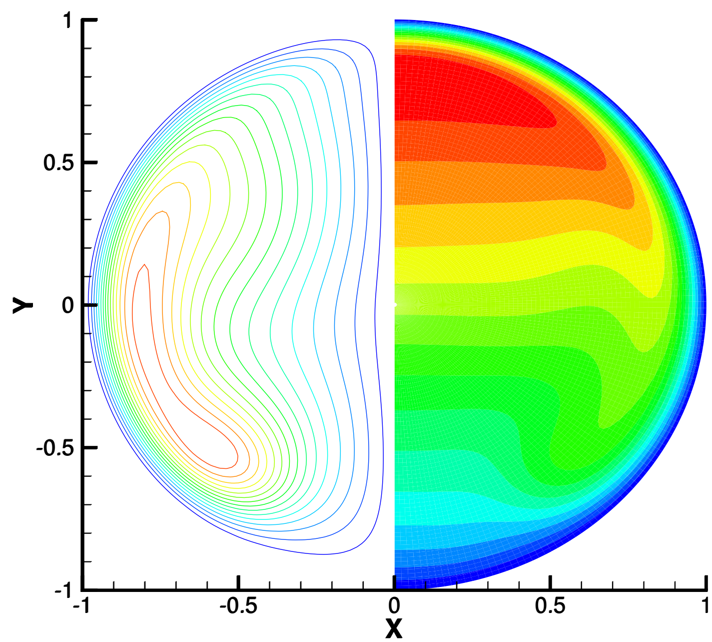
Results – Flow Regimes

The following flow regimes are observed:

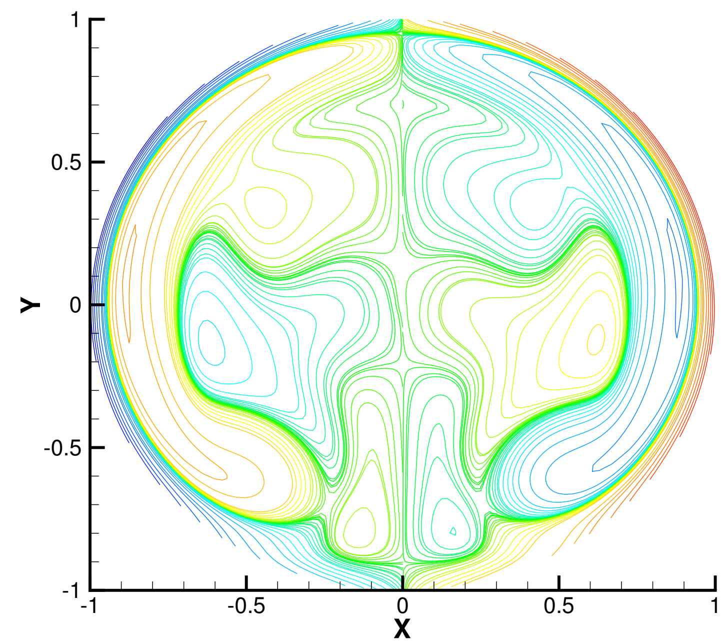
- ☞ Steady, symmetric solutions: $Ra_R \leq 1 \times 10^8$
- ☞ Unsteady, asymmetric solutions: $Ra_R \geq 1 \times 10^9$

Steady, symmetric results for $Ra_R = 1.57 \times 10^7$ (uniform heat generation):

Streamfunction: Temperature:



Vorticity:



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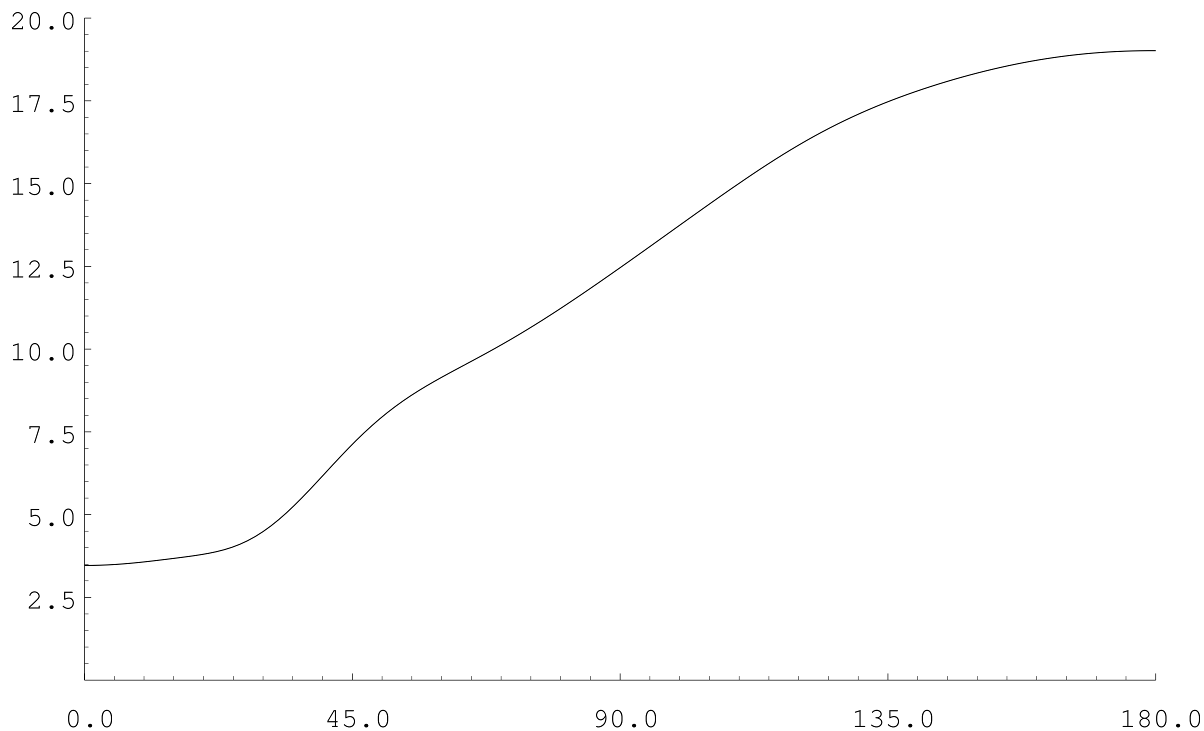
Steady, Symmetric Results (cont'd)

Nusselt number versus θ for $Ra_R = 1.57 \times 10^7$ (uniform heat generation):

Nu vs. θ:



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Code Comparisons – Average Nusselt Number ($\bar{N}u$)

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	8.2
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.* $\bar{N}u_{MB} = 0.7041 \cdot Ra_{MB}^{0.1864}$.

Code Comparisons – COA vs. FLUENT

Gaussian heat generation: $\sigma = 0.25$

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

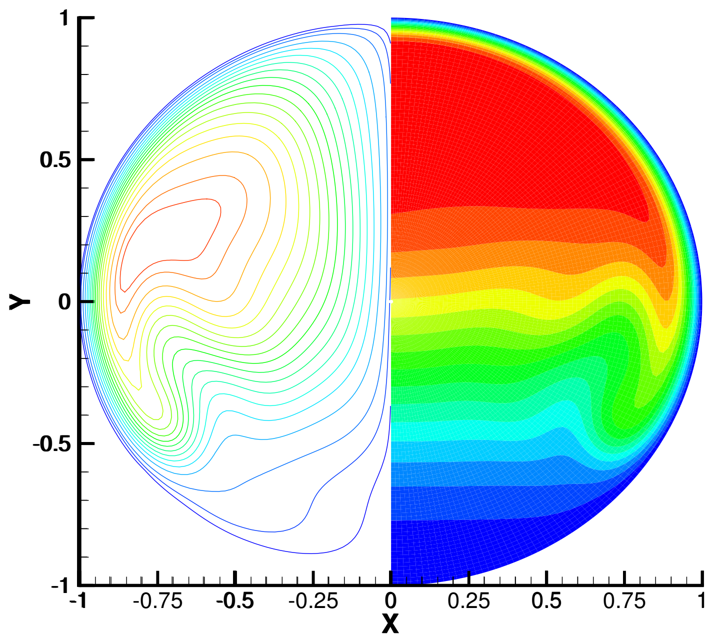
	FLUENT[1]			COA Code		
Ra_R	T_{avg}	T_{max}	$\bar{Nu}$	T_{avg}	T_{max}	$\bar{Nu}$
1×10^8	0.0101	0.0169	16.4	0.0100	0.018	15.6
1×10^9	0.0067	0.0101	25.1	0.0065	0.011	25.4
1×10^{10}	0.0045	0.0060	38.5	0.0039	0.0070	46

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

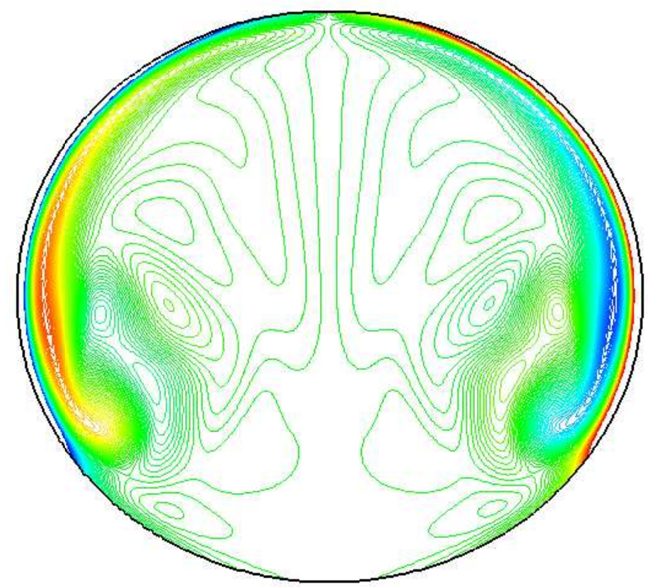
$$T_{avg_{MB}} = 0.3130 \cdot Ra_{MB}^{-0.1771}, \quad T_{max_{MB}} = 1.3597 \cdot Ra_{MB}^{-0.2233}, \quad \bar{Nu}_{MB} = 0.7041 \cdot Ra_{MB}^{0.1852}$$

Steady, Symmetric Results: $Ra_R = 1 \times 10^8, \sigma = 0.25$

Streamfunction: Temperature:

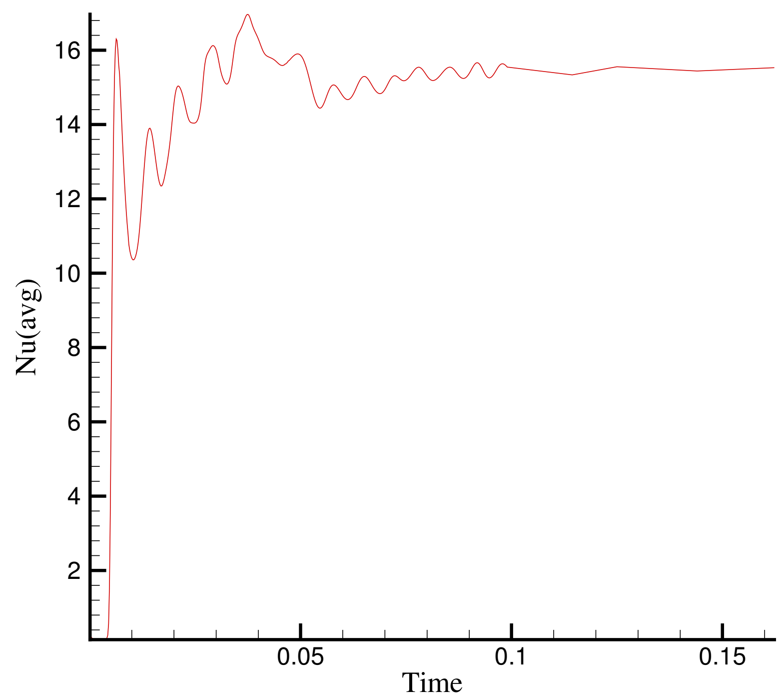


Vorticity:

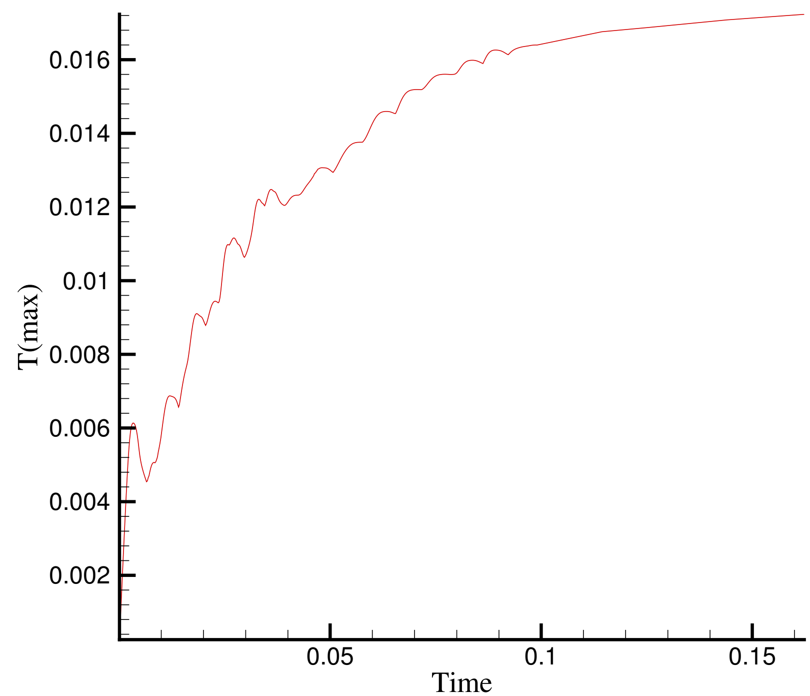


Steady, Symmetric Results: $Ra_R = 1 \times 10^8, \sigma = 0.25$

$\bar{Nu}$ vs. t:



T_{max} vs. t:



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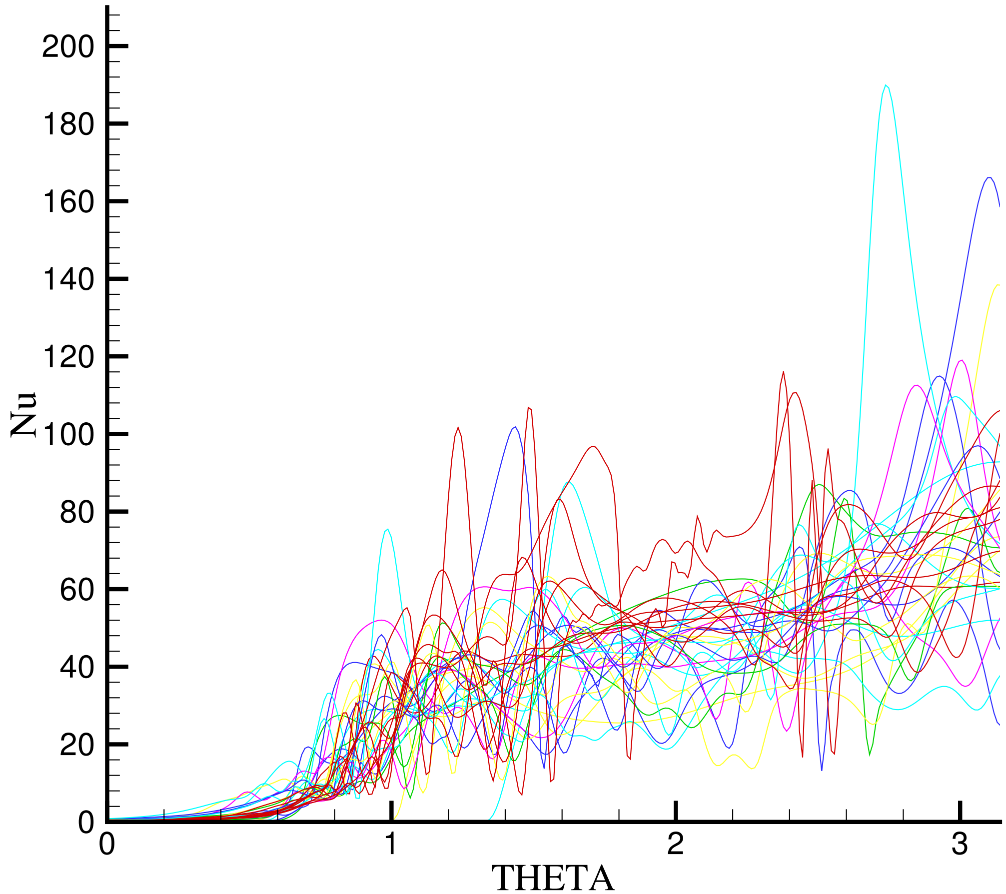
Unsteady, Asymmetric Results: $Ra_R = 1 \times 10^{10}, \sigma = 0.25$

Nu vs. θ:



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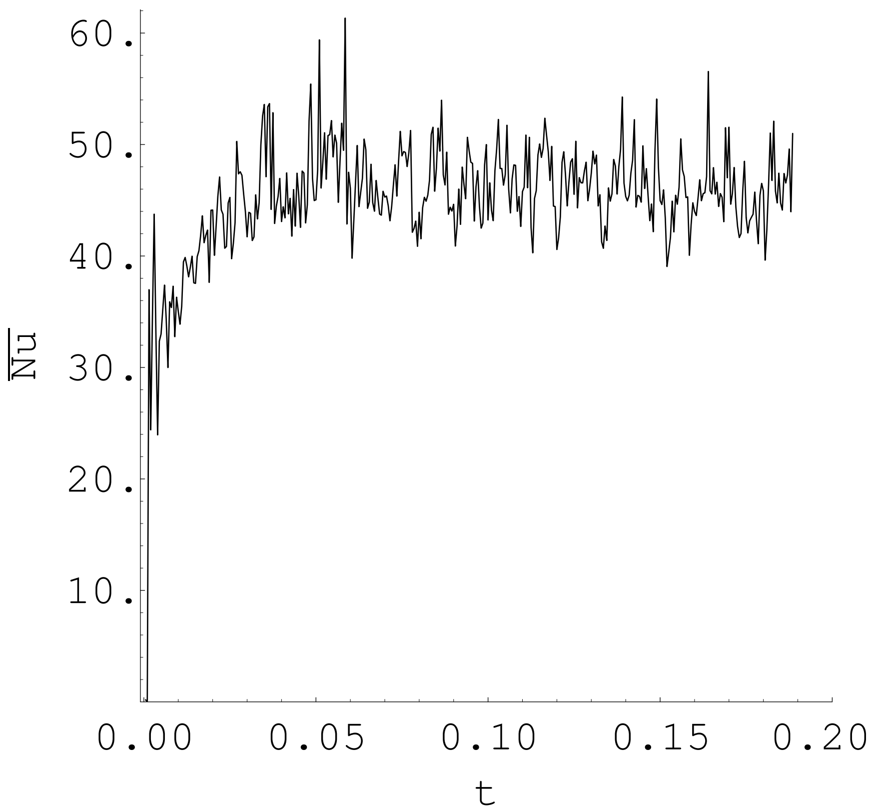
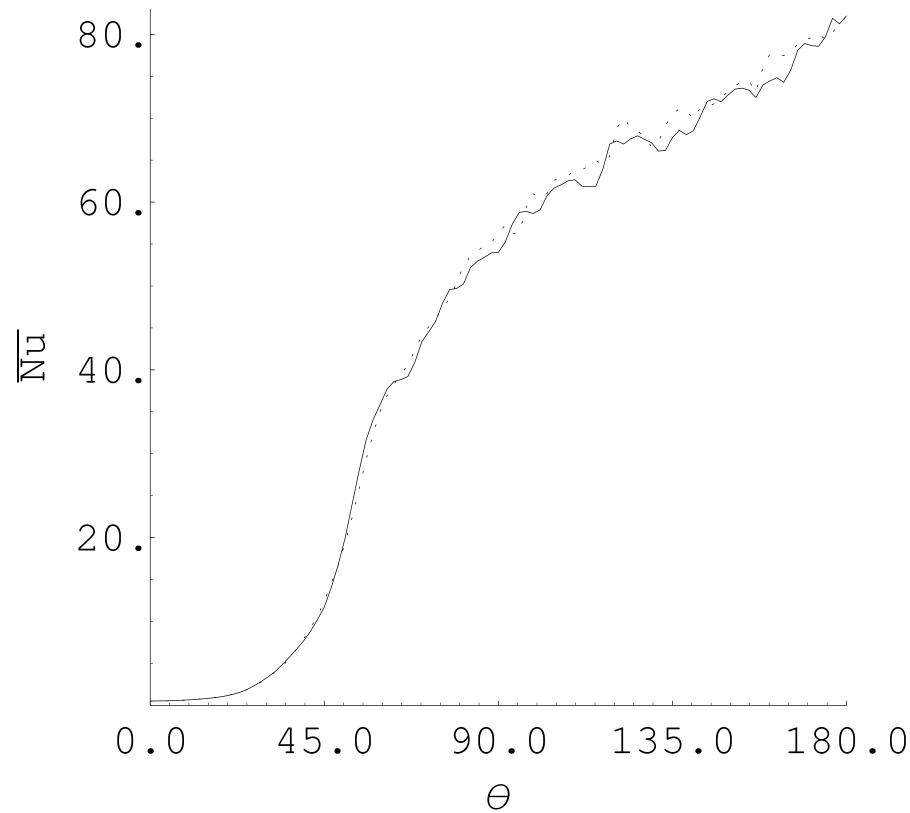
Unsteady, Asymmetric Results: $Ra_R = 1 \times 10^{10}, \sigma = 0.25$

$\bar{N}u$ vs. t:



$\bar{N}u$ vs. θ:



Gaseous Absorber Parameters

For $dE/dx = 13.81$ M, 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).

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:

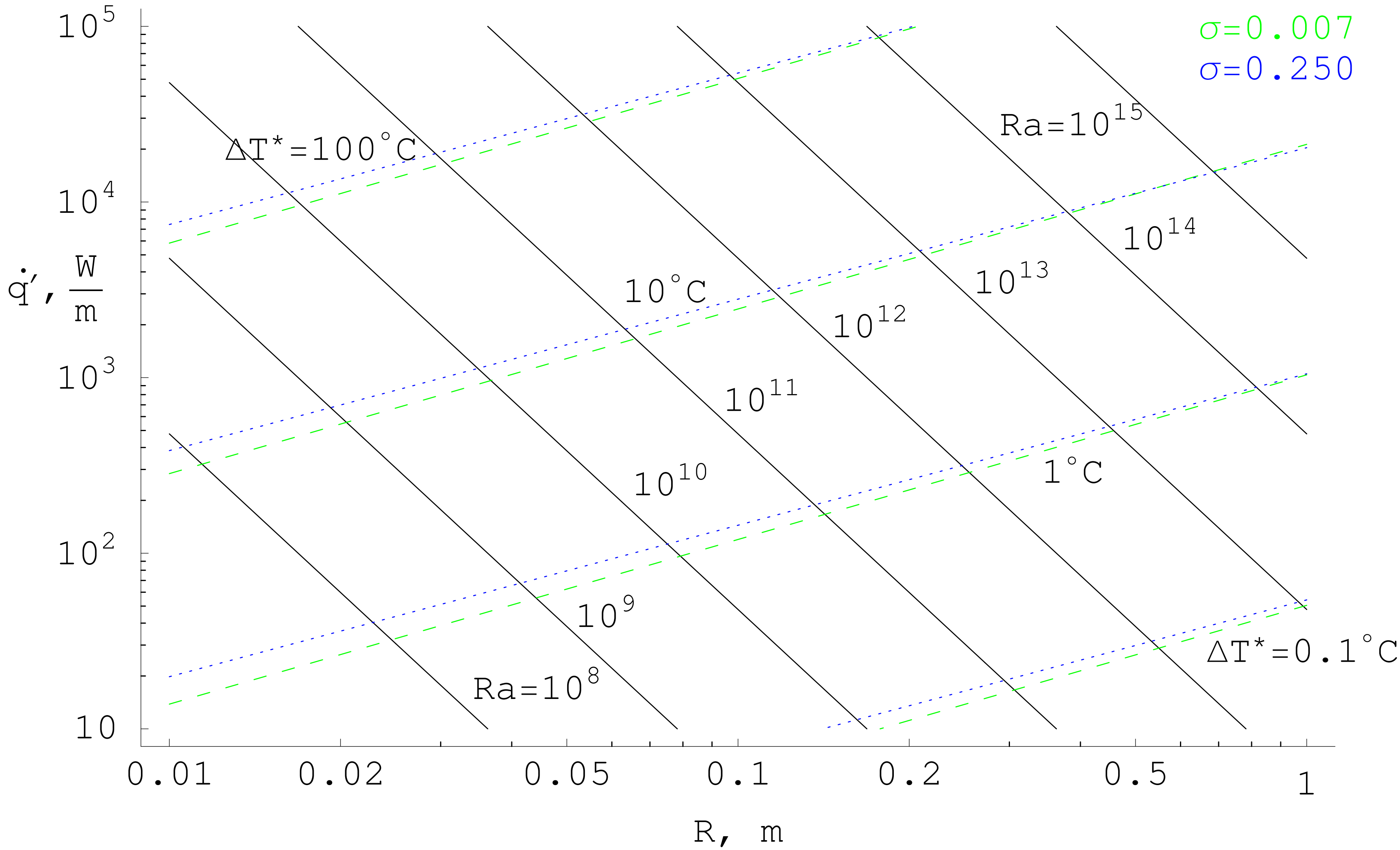
✓ Set up: absorber encased in cooling sheath (similar to actual absorber).

✓ Heat source: electric current in absorber fluid, beam, etc.

✓ Absorber fluid: water is a candidate.

  → 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 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.

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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.

$\Longrightarrow$ 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.