High Power Targets

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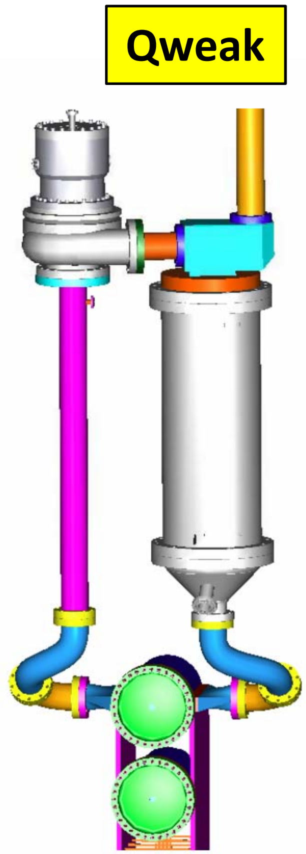
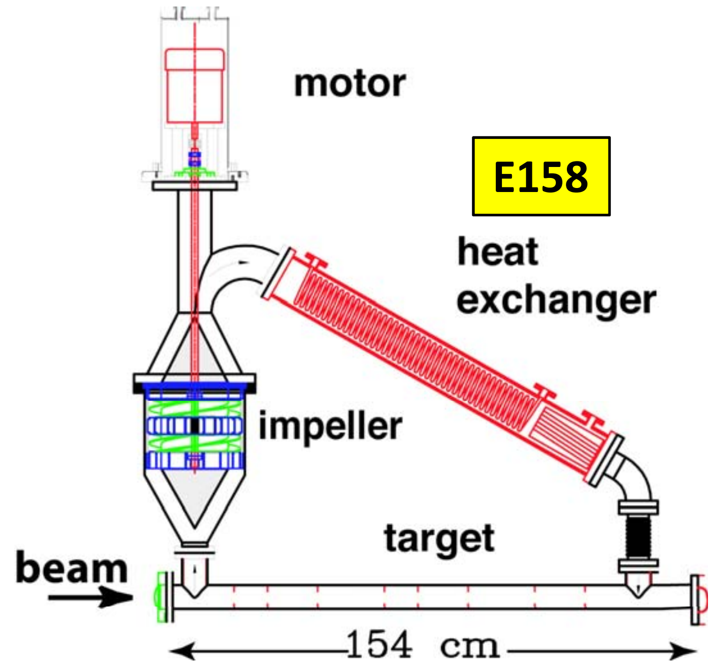
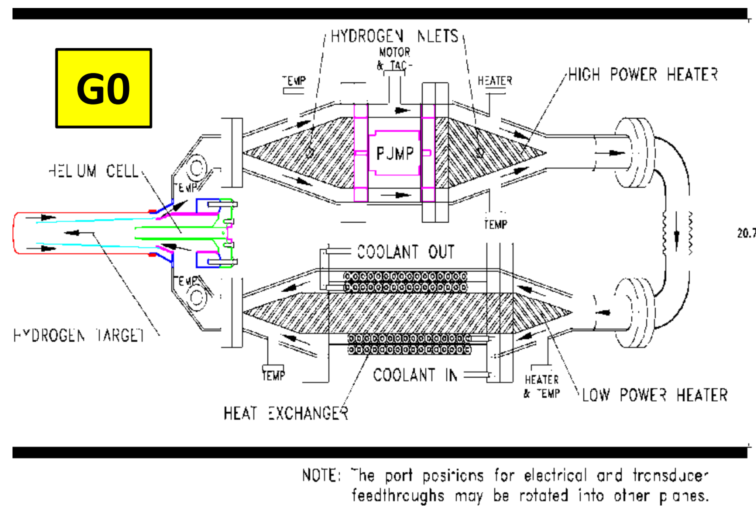
PAVI09 June 24, 2009

- In collaboration with :
  - S. Covrig, JLab
  - J. Dunne, Miss. State (heater)
  - D. Meekins (Jlab)





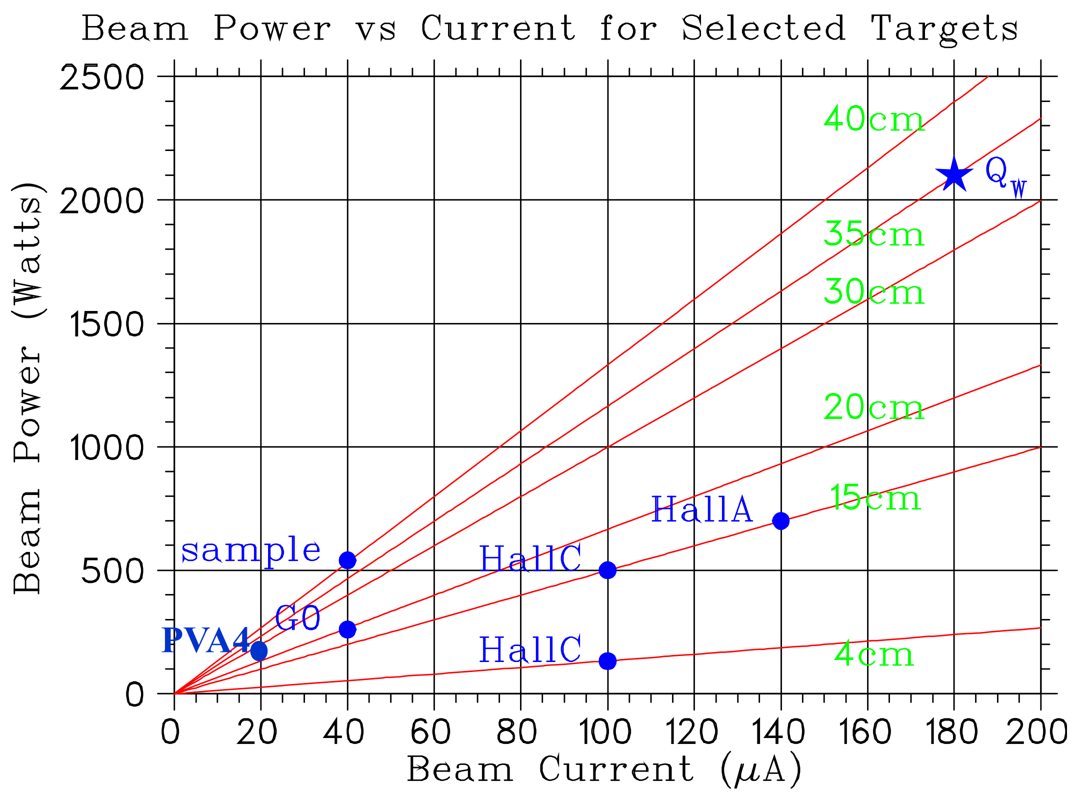


Comparisons

JLab Moller

- G0: LF @ Jlab
- Sample: LF @ Bates
- PVA4: LF @ Mainz
- Hall A & C: Mixed LF & TF, @ Jlab

- Q_{weak}: TF @ Jlab (planned)
- JLab Moller: (probably) LF, Jlab, (planned)



LH_2 Targets for Parity Violation

	p / T / $\dot{m}$ psia/K/kg/s	L cm	P/I / E W / µA / GeV	P_v W/cm^3	d_{beam} mm	Δρ/ρ %	δρ/ρ ppm
sample	25/20/0.6	40	700/40/0.2	396	2	1	<1000 @60Hz
happex	26/19/0.1	20	500/35-55/3	76	5x5 6x3	?	100 @30Hz
pv-a4	25/17/0.13	10	250/20/0.854	310	1.7	0.1	392 @50Hz
e158	21/20/1.8	150	700/12/48	467	1	1.5	<65 @120Hz
g0	25/19/0.3	20	500/40-60/3	346	2x2	<1.5	<238 @30Hz
Q_{weak}	35/20/1	35	2500/180/1.1	245	5x5		<45 @250Hz
møller		150	5000/85/11				<26 @2000Hz

Note: Achieved 68 ppm with double the raster area & double the pump head

Extrapolating Performance

	Raster	L_{tgt}	I_{beam}	Massflow	$n_{helicity}$

$$Q_{weak} = G0 \quad \frac{2x2}{5x5} \quad \frac{35\ \text{cm}}{20\ \text{cm}} \quad \frac{180\ \mu\text{A}}{40\ \mu\text{A}} \quad \frac{7\ \text{l/s}}{15\ \text{l/s}} \quad \left(\frac{30\ \text{Hz}}{250\ \text{Hz}}\right)^{0.4}$$

Q_{weak} = 250 ppm x **0.16** x **1.75** x **4.5** x **0.47** x **0.43** = **63 ppm**

- **Note G0 is longitudinal flow, Q_{weak} is transverse**
- **Assumes massflow dependence linear, although some evidence suggests could be cubic. If cubic → 14 ppm!**
- **Note: expect ΔA_{stat} ~ 140 ppm (@250 Hz) in Q_{weak}**
- **Need ΔA_{boil} < 45 ppm to keep runtime penalty below 10%**

Design Considerations

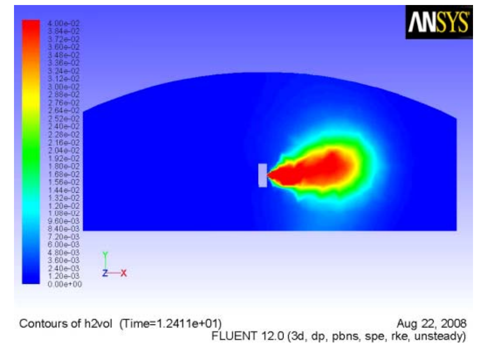
Knobs to turn:

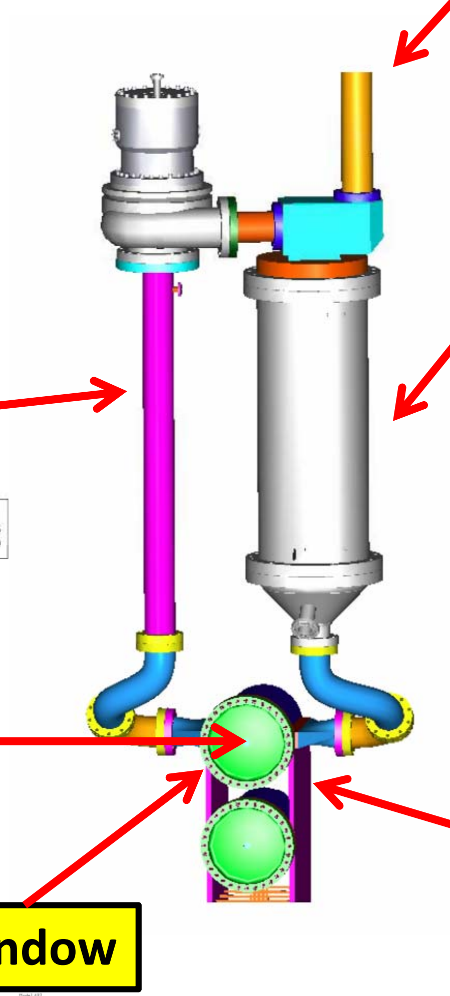
- P & T
- V_{flow}
- A_{raster}
- $n_{helicity}$
- n_{raster}
- Intrinsic ϕ_{beam}
- Cell/Flow design
- Window design

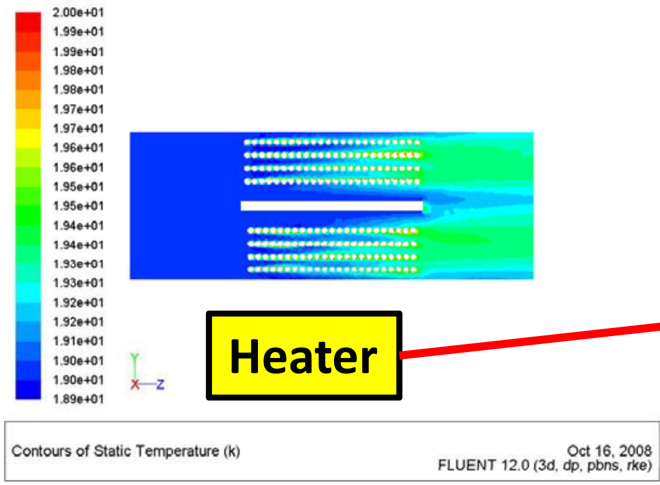
Constraints:

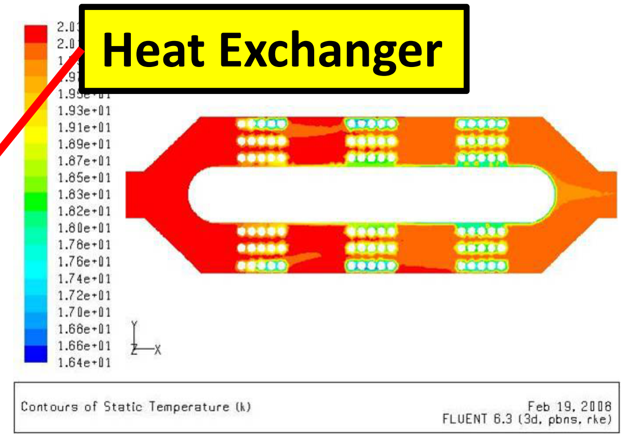
- I_{beam} & L_{tgt}
- Window bkg
- Safety issues
- Available $P_{cooling}$
- Head
- ΔA_{stat}
- Time available
- ASME compliance

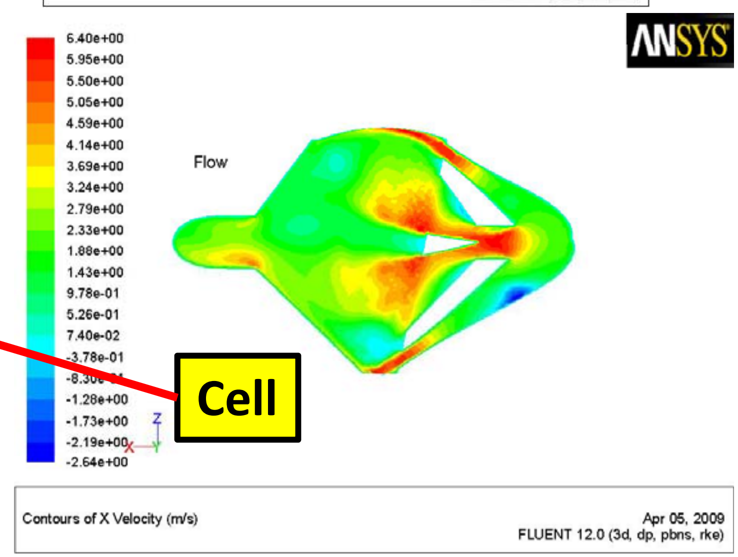
Design by CFD

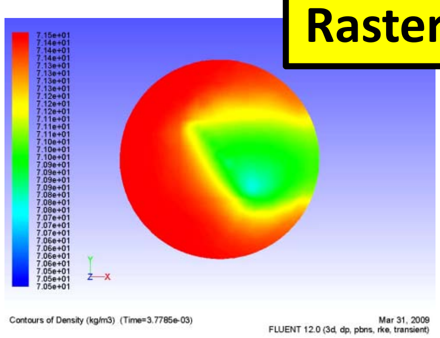


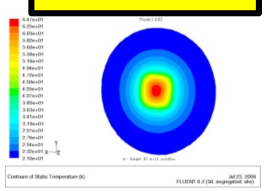








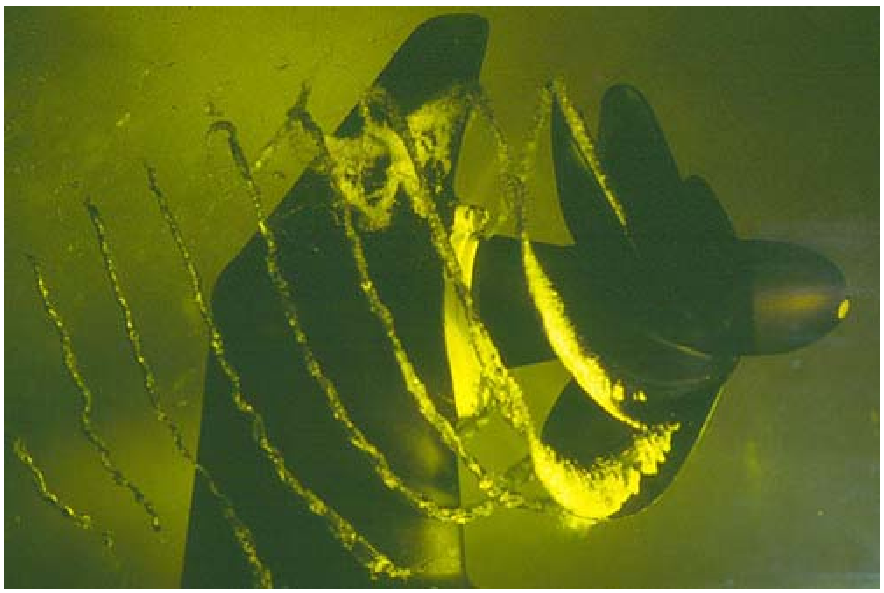




P_{max} Considerations

Lower P:

- Don't go sub-atmospheric
- Thinner windows = less bkg
- Lower warm gas storage P
- Less gas inventory



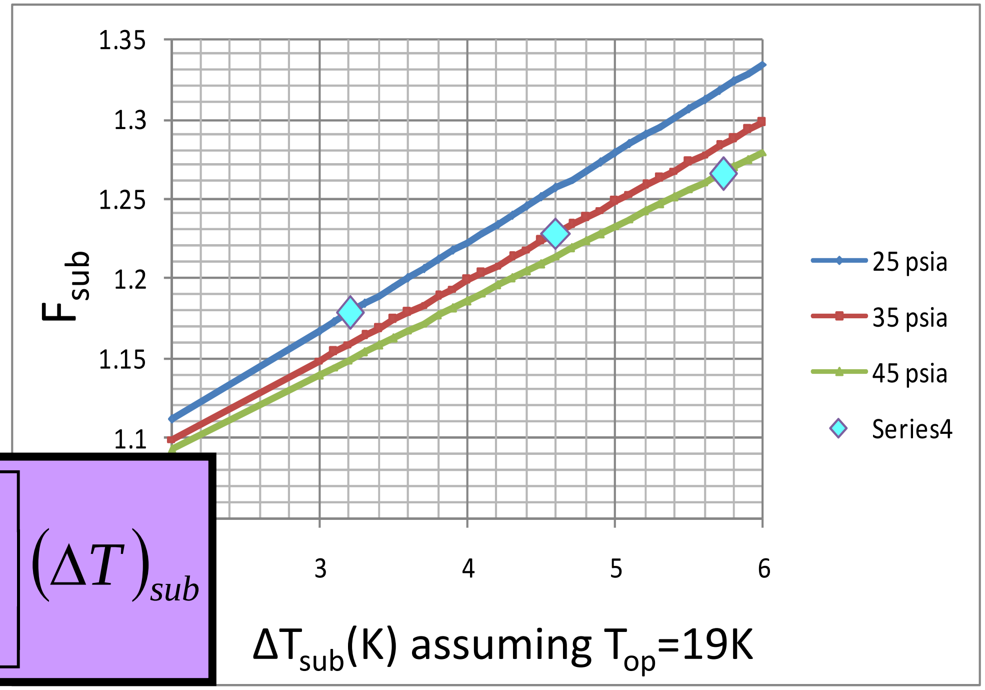
Higher P:

- More cavitation headroom = $P_{op} - P_{VP}$. Cavitation occurs at trailing edge of pump blades when $P < P_{VP}$. For LH_2 $P_{VP}(19K)$ ~ 10 psia.
- Higher boiling temperatures
  - Run at higher T → more cooling power
  - Run at fixed T → more subcooling
    - Less film boiling at windows?

Can subcooling mitigate film boiling at window?

$$F_{sub} \equiv \frac{q_{subcooled}}{q_{saturation}}$$

where q=heat flux across film at window



NBS Technical Note 317 (E.G. Brentari, 1965):

$$F_{sub} = 1.0 + 12.236\left[\frac{\left(C_p \rho k\right)_l^{0.5}\left(\Delta\rho\right)^{0.125}}{\lambda \rho_v^{0.75}\sigma^{0.375}}\right]\left(\Delta T\right)_{sub}$$

C_pf	10959	12100	13198	J/(kgK)	Specific Heat
k_f	0.1041	0.1032	0.1020	W/(mK)	Thermal Conductivity
T_w	40.00	40.00	40.00	K	Wall/Exit window temp
P_tgt	25	35	45	psia	Operating pressure
T_l	22.21	23.60	24.73	K	Saturation temp
T_f	31.10	31.80	32.36	K	Film temp, assumed to be the mean of the wall and liquid
dT	17.79	16.40	15.27	K	$\Delta T = T_w - T_l$
mu_f	1.63E-06	1.68E-06	1.72E-06	Pa-s	Viscosity
k_f	0.0250	0.0260	0.0269	W/(mK)	Thermal Conductivity
rho_l	68.461	66.580	64.900	kg/m^3	Liquid density
rho_f	1.419	1.980	2.549	kg/m^3	Vapor density
deltarho_f	67.04	64.60	62.35	kg/m^3	liquid-film density diff
C_pf	11137	11473	11827	J/(kgK)	Specific Heat
sigma	1.62E-03	1.40E-03	1.22E-03	N/m	Liquid surface tension
lambda	4.36E+05	4.24E+05	4.11E+05	J/kg	Heat of vaporization
lambda'	5.81E+05	5.62E+05	5.43E+05	J/kg	

So the effect is there, but it's too small to be very interesting.

P (psia)	25	30	35	40	45
T_vp (K)	22.21	22.94	23.60	24.19	24.73
dT (K)	3.21	3.94	4.60	5.19	5.73
		0.74	0.65	0.59	0.54
F_sub	1.18		1.23		1.26
Gain	-		4%		7%

Benefits of Higher Operating Pressure

- **Normal: 19K, 25 psia**
  - **Fix T → Run more subcooled**

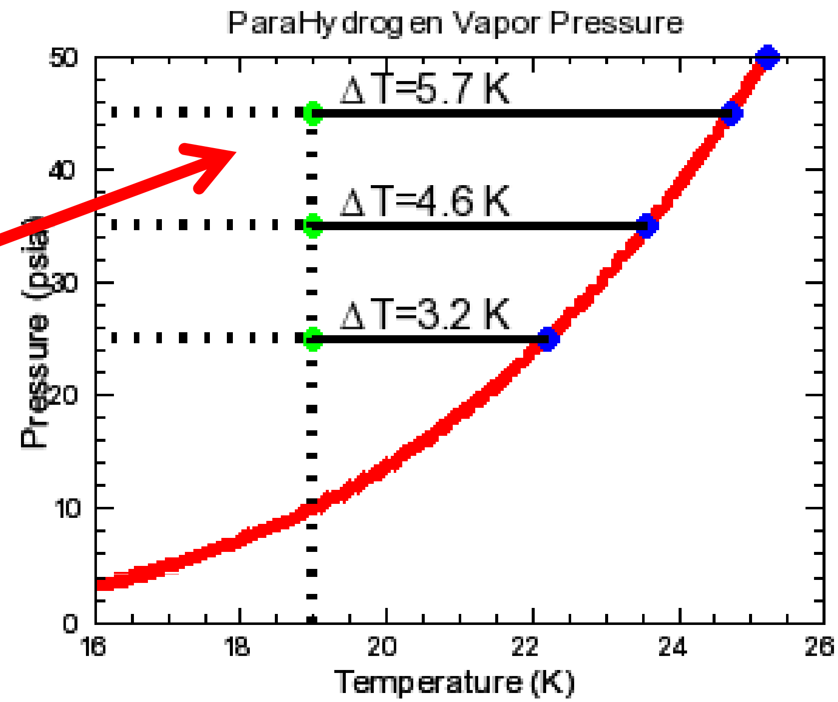
    - OR -
  - **Fix subcooling → Gain more $P_{cooling}$**

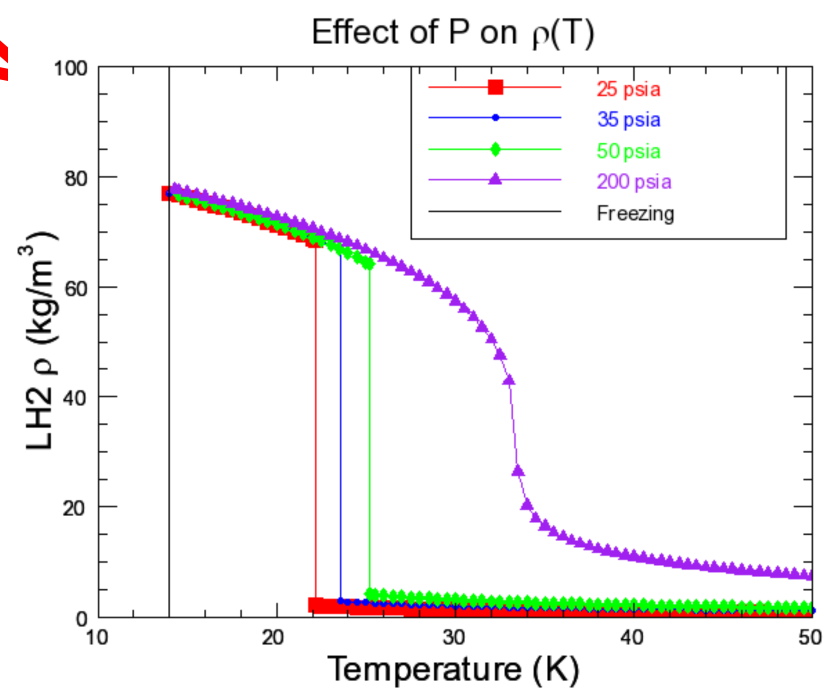
- **Ex:/ 50 psi, 22K**
  - **Win 3K in ΔT**
  - **Still 3K subcooled**

$$P_{cooling} = c_p \dot{m} \Delta T$$

4K coolant → win ~20%

15K coolant → <u>win 75%!</u>





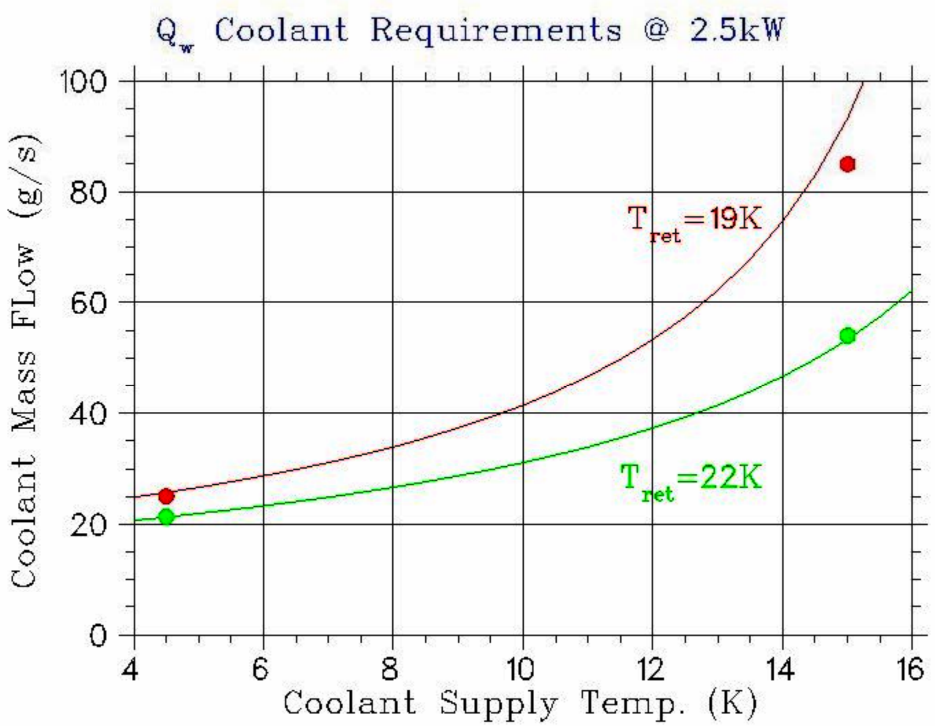
Cooling Power

$$P_b(W) = I_b(\mu A)\ \ \rho(g/cm^3)\ \ t(cm)\ \ dE/dx(MeV/g/cm^2)$$

With: I_b=180 µA, ρ=0.072 g/cm^3, t=35 cm, → **P_b=2.1 kW!**

Cooling Power Budget

Pump efficiency	60%
Flow rate (liters/s)	15
Pump Head (psi)	1.3
Pump Power (hp)	0.5
Beam Current (uA)	180
Beam Power (W)	2120
PID reserve (W)	150
Pump heat (W)	75
Viscouse heating (W)	224
Conductive Losses (W)	50
Total Load (W)	2618

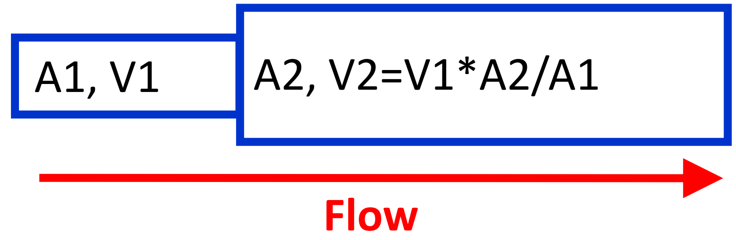


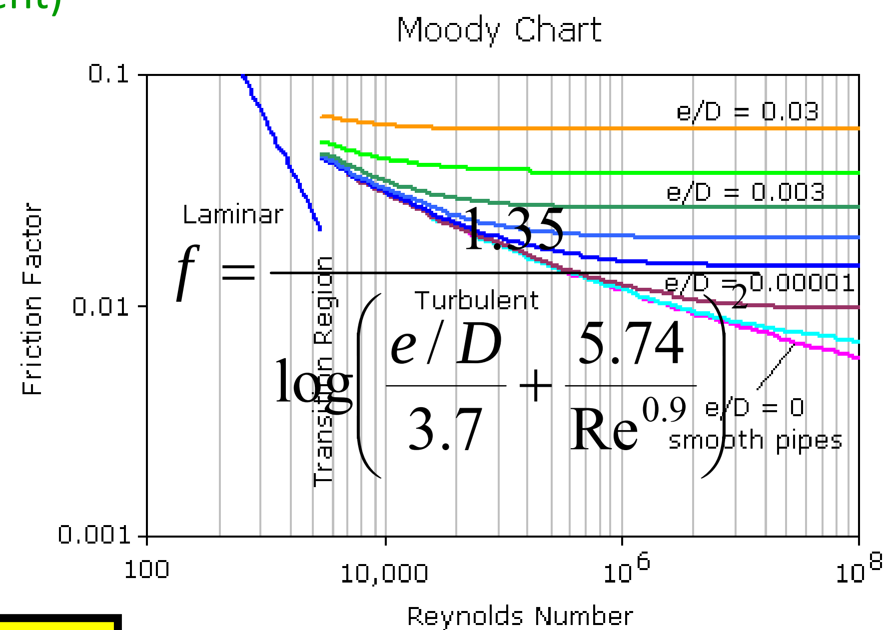
Viscous Heating

$$h_L = \frac{(v_1 - v_2)^2}{2g}$$ (Abrupt Enlargement)

$$h_L = \frac{K_L v_2^{\,2}}{2g}$$ (Abrupt Contraction, Commercial Fittings)

$$h_L = f\,\frac{L}{d}\,\frac{v^2}{2g}$$ (Circular Pipe)





$$f = \frac{1.35}{\left(\log\left(\frac{e/D}{3.7} + \frac{5.74}{\mathrm{Re}^{0.9}}\right)\right)^2}$$

$$\text{Viscous Heating (W)} = \frac{\text{Flow (l/s) Head(psi)}}{\text{pump efficiency}} \times 6.89$$

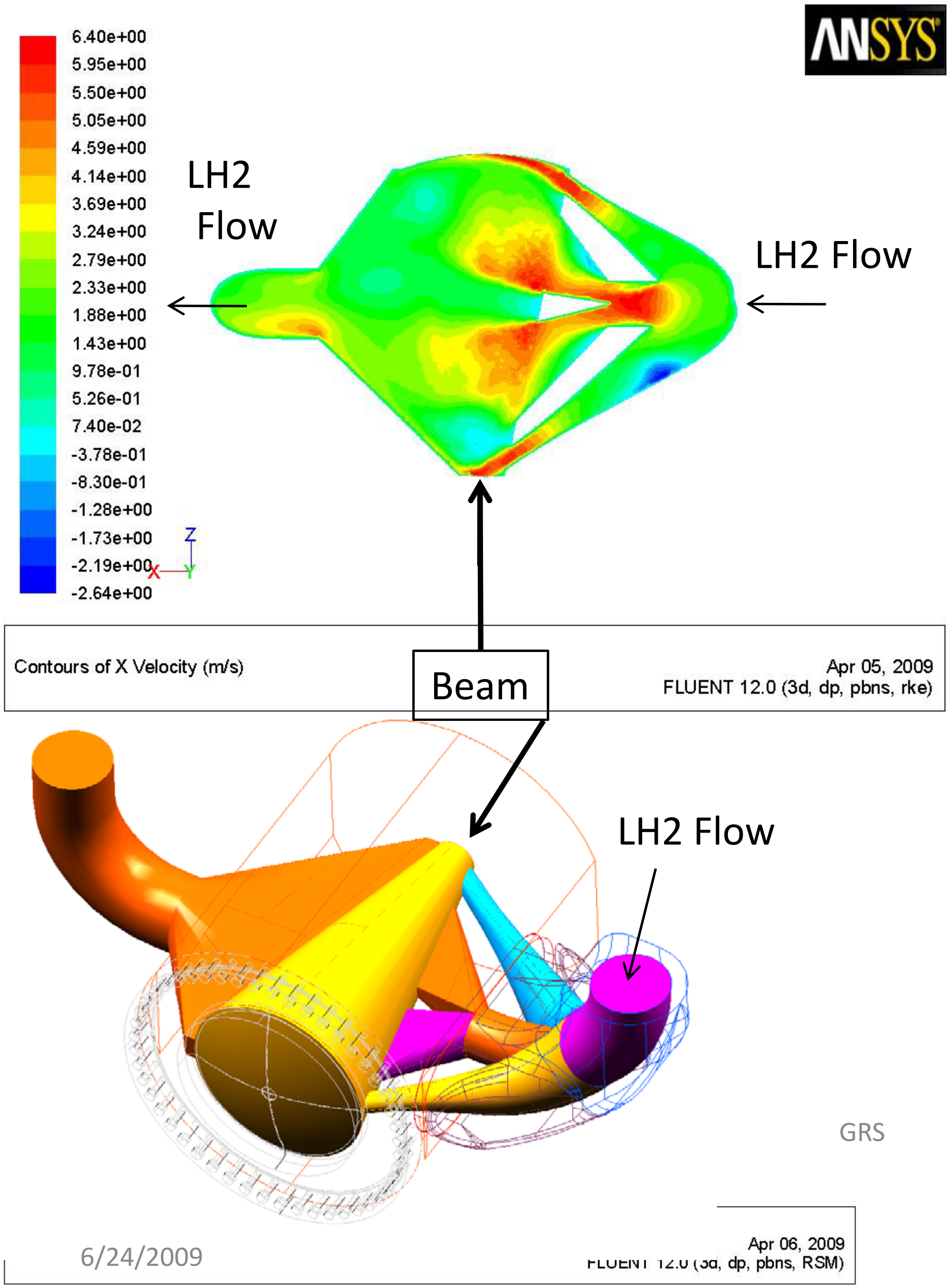
Ex: 15 l/s, 2 psi, 80% → 250 W
30 l/s → 2000 W!

So, viscous heating $\propto v^3$!

GRS

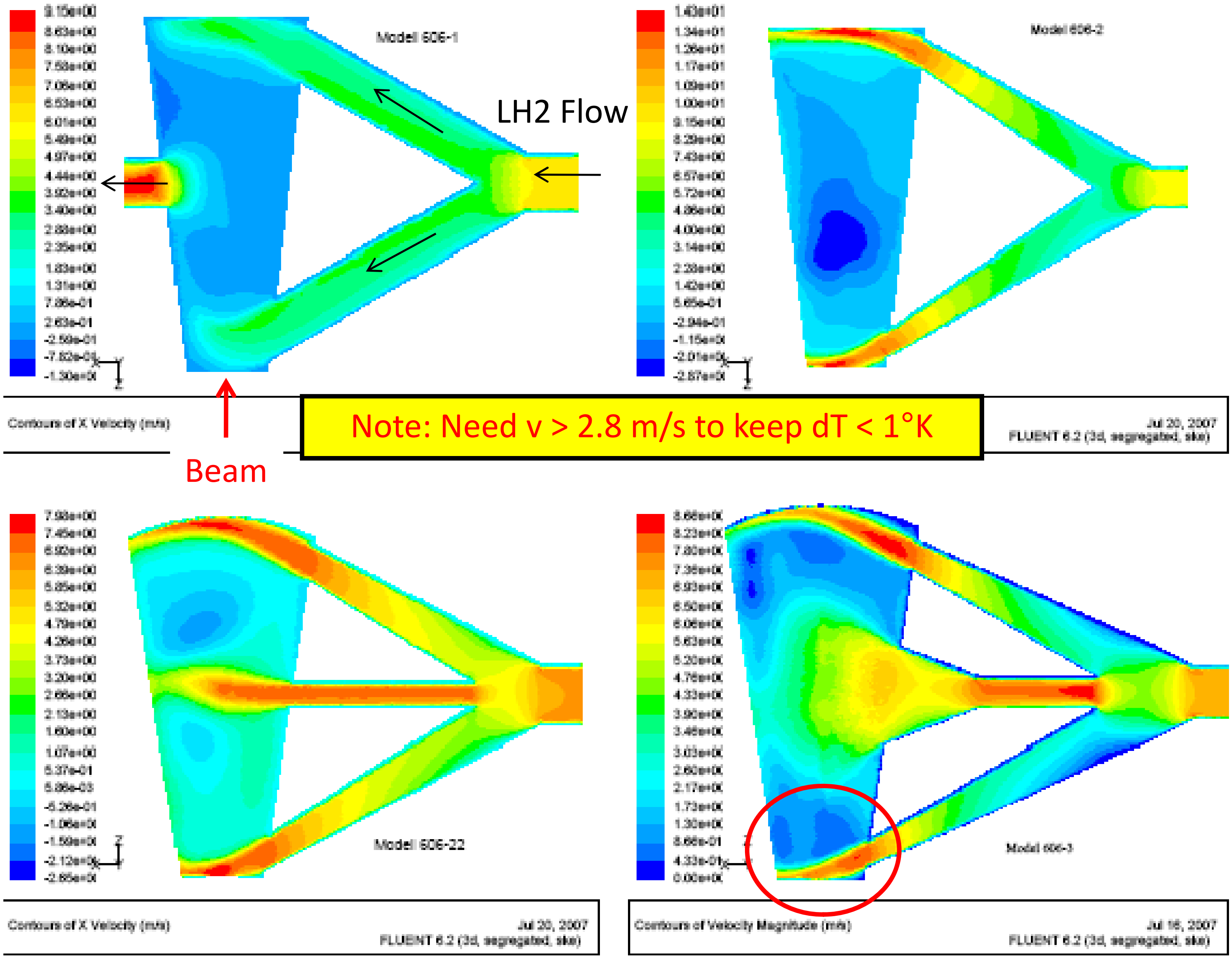
Note: $\Delta P = h_L\, \rho\, g$, $Re = v\, d\, \rho / \mu$,
e ~ 0.0015 mm for Al pipes

Cell Design



- **Factors:**
  - Bulk boiling
  - Window boiling
  - Density reduction
  - Head
  - Volume
  - Longitudinal vs transverse flow
- **Conclusions:**
  - For Qweak cell, CFD predicts TF better than LF by ~ *2 in Δρ/ρ, & ΔT (bulk).
  - TF flow also better at windows if properly directed

Transverse Velocity Model Dependence



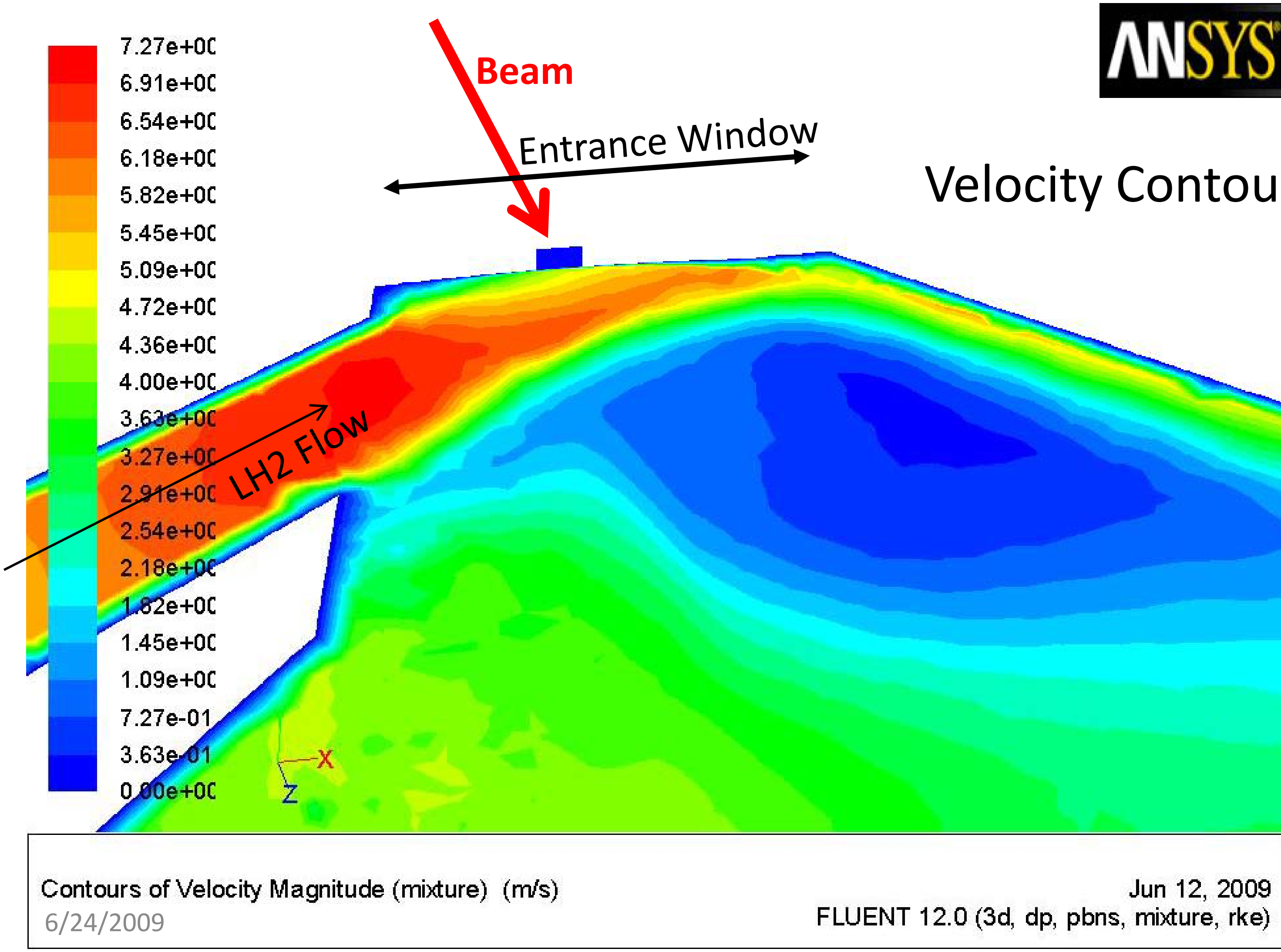
Two Phase CFD (window boiling)

CFD simulation by S. Covrig

Rastered Beam profile on 0.005” Al cell entrance window

ANSYS

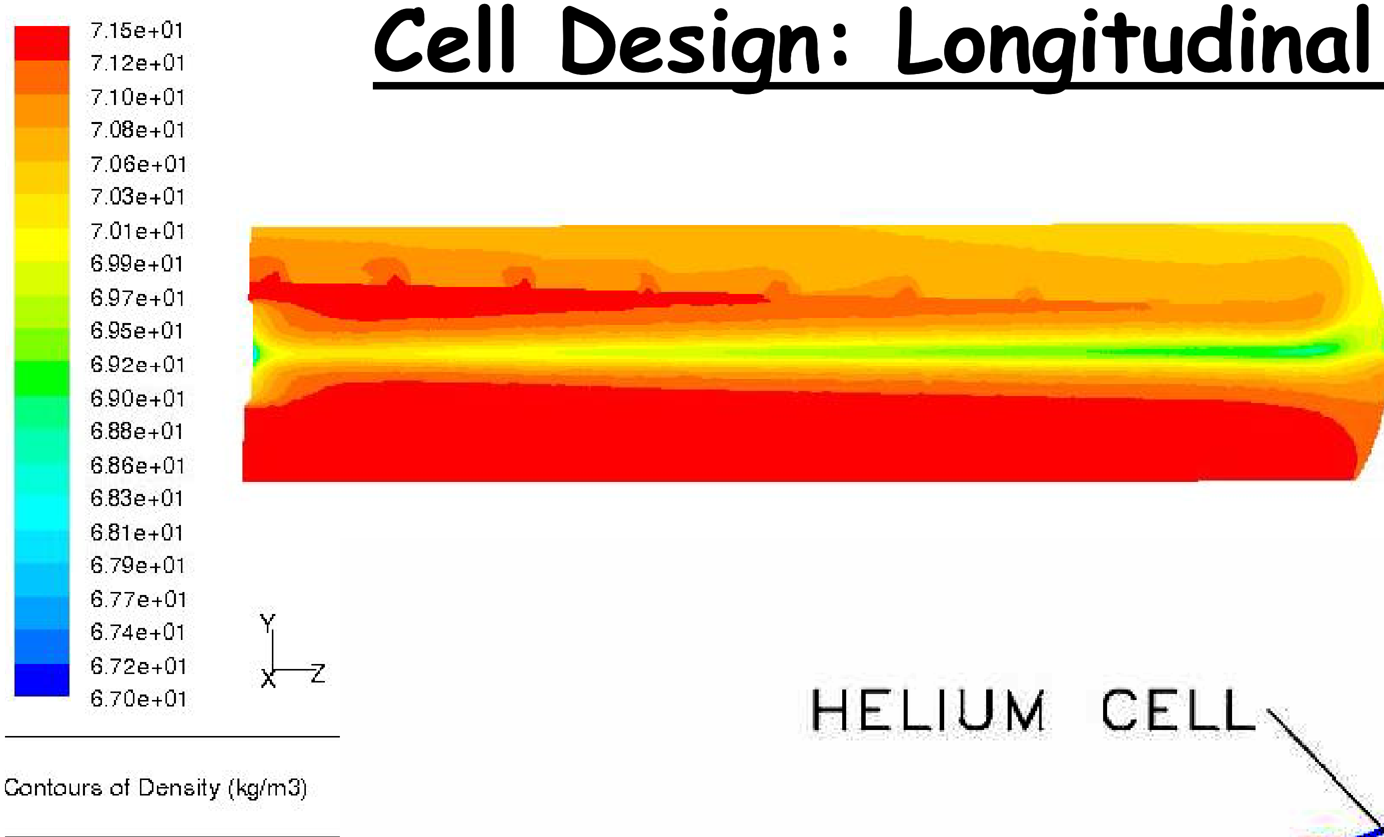
Velocity Contours

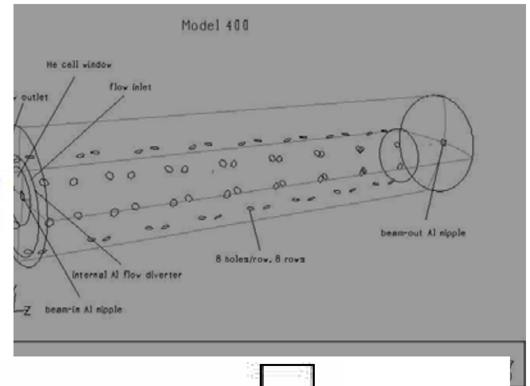


Both Phases

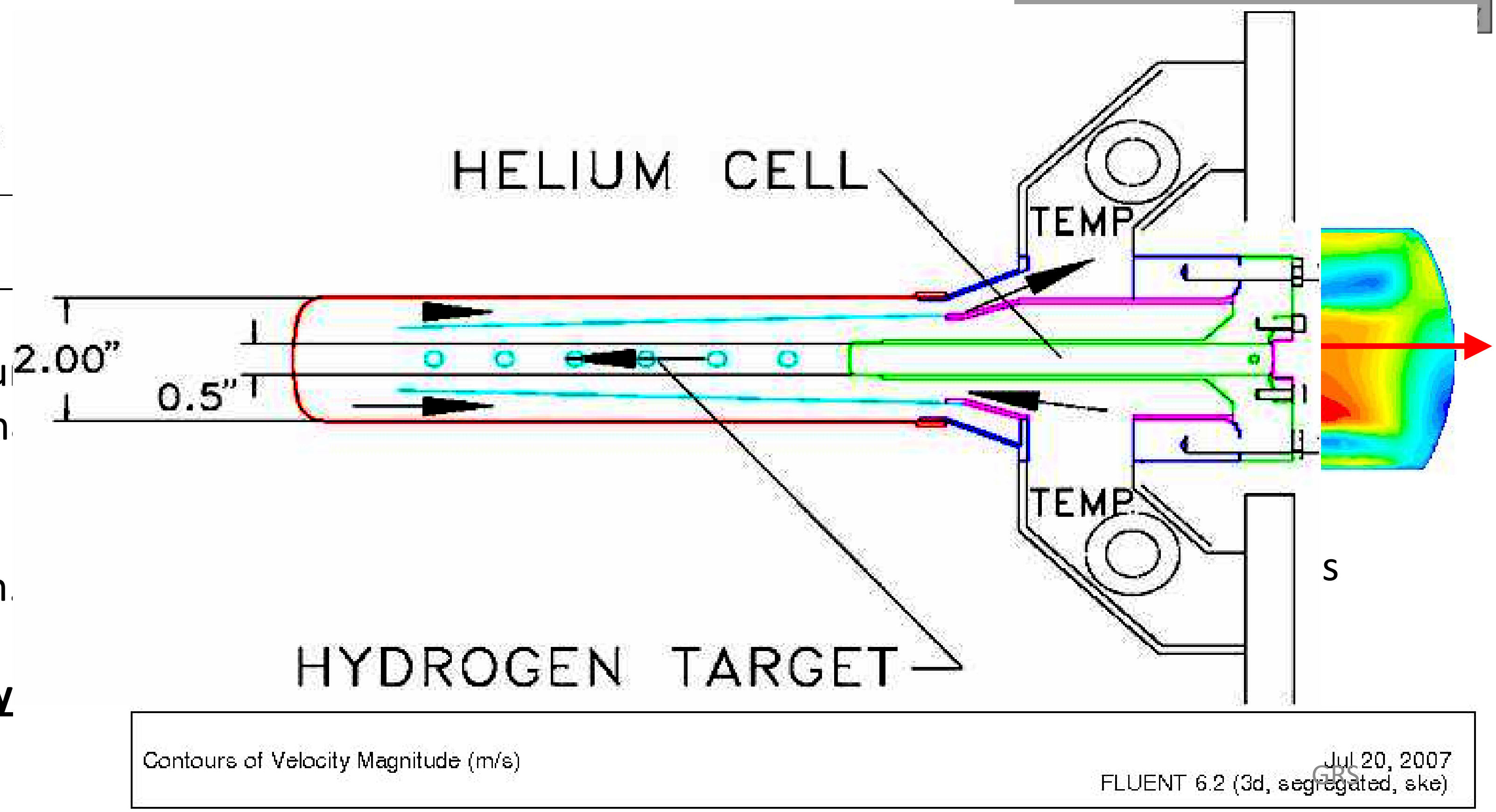
<u>Vapor Only</u>
(BLUE means no vapor there, ie just liquid).

Cell Design: Longitudinal Skewed





Mainz/A4 fou
empirically th
boiling was
reduced with
off-axis beam.



<u>Confirmed by CFD model.</u>

s

Prototype for 11 GeV Møller Target Cell

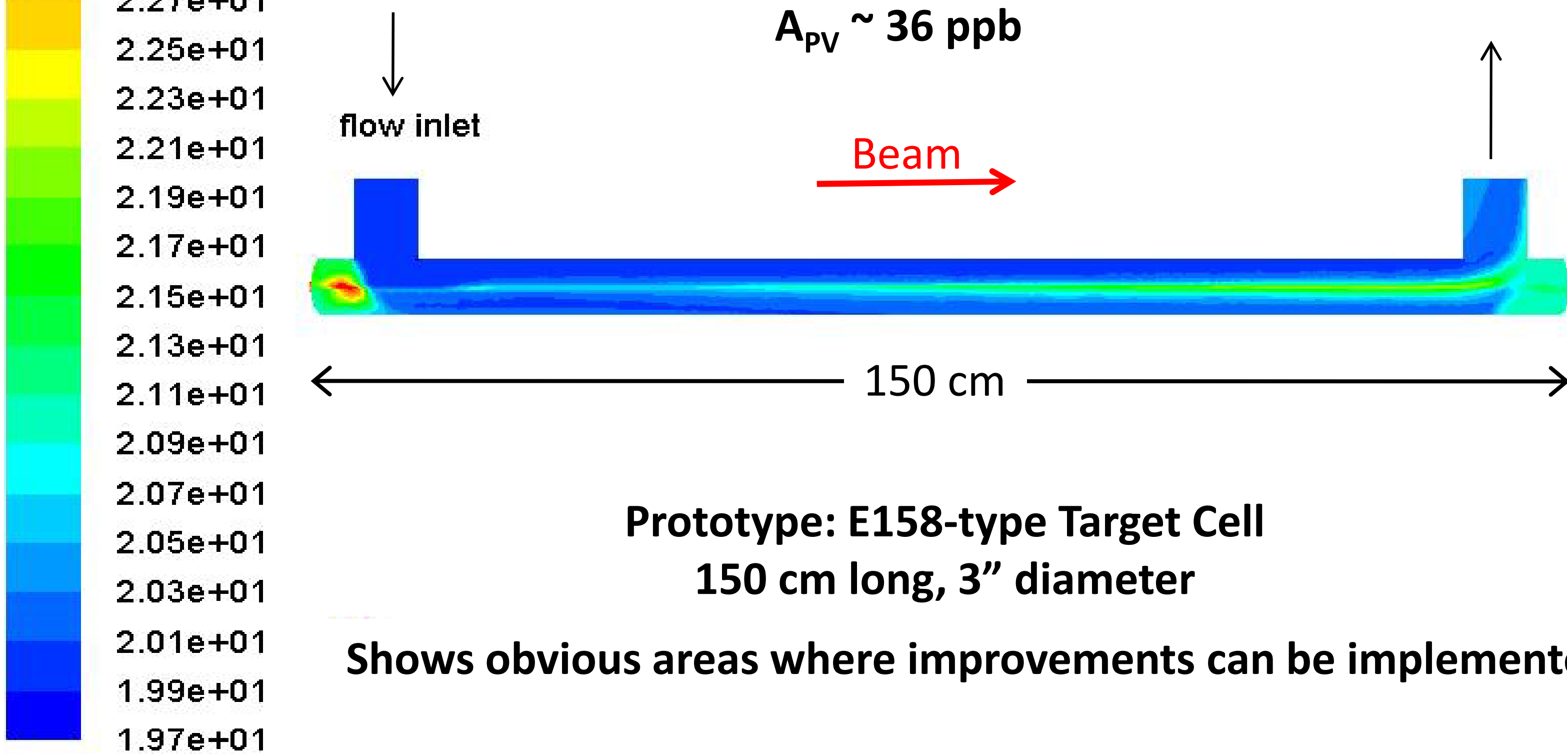
Beam heating 4600 W @85 μA

Need δρ/ρ < 26 ppm @ 2000 Hz

(δρ/ρ < 9 ppm @ 250 Hz)

S. Covrig, JLab

A_{PV} ~ 36 ppb



Prototype: E158-type Target Cell
150 cm long, 3" diameter

Shows obvious areas where improvements can be implemented.

Contours of Static Temperature (k)

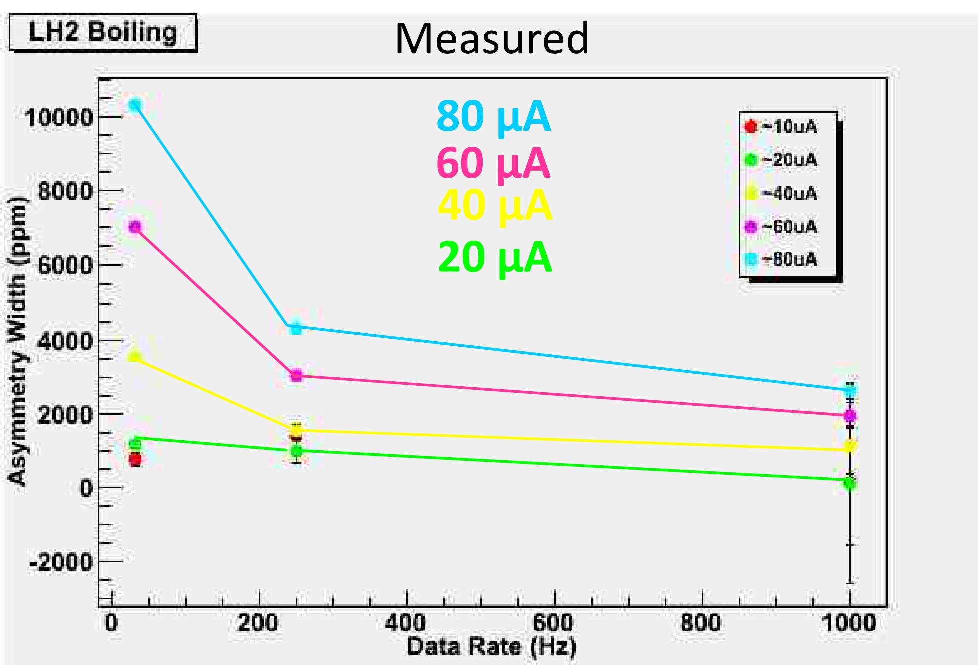
Nov 14, 2008
FLUENT 12.0 (3d, dp, pbns, rke)

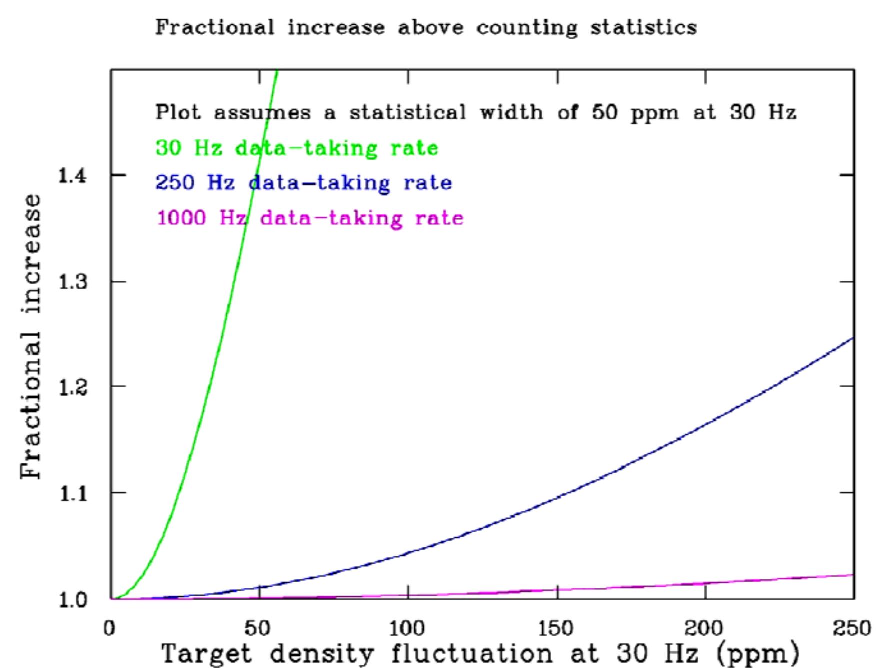
Higher helicity reversal rates

The statistical width is given by: $\Gamma_{stat} = \sqrt{\Gamma^2_{counting} + \Gamma^2_{target}}$

1. We can reduce the relative contribution of the target boiling term by going to higher helicity reversal frequencies (**increased $\Gamma_{counting}$**).

2. Tests (**VPI**/Jlab/OU, June 2008) with a Hall C standard tgt indicate that **the boiling term drops with frequency** as: $\Gamma_{targ} = \Gamma_{30\,Hz}\left(\frac{30\text{ Hz}}{f}\right)^{0.4}$



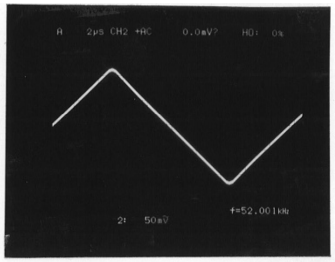


First Dynamic CFD Model

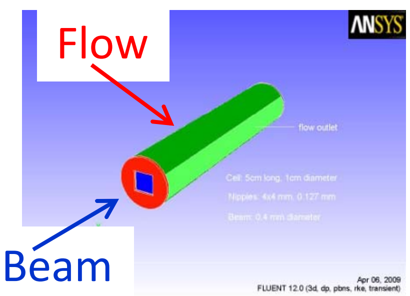
Toy Cell ϕ = 1 cm, 25 kHz (Δ=120 Hz) raster 4x4 mm²

⊙ Beam

Simulation by
S. Covrig, JLab



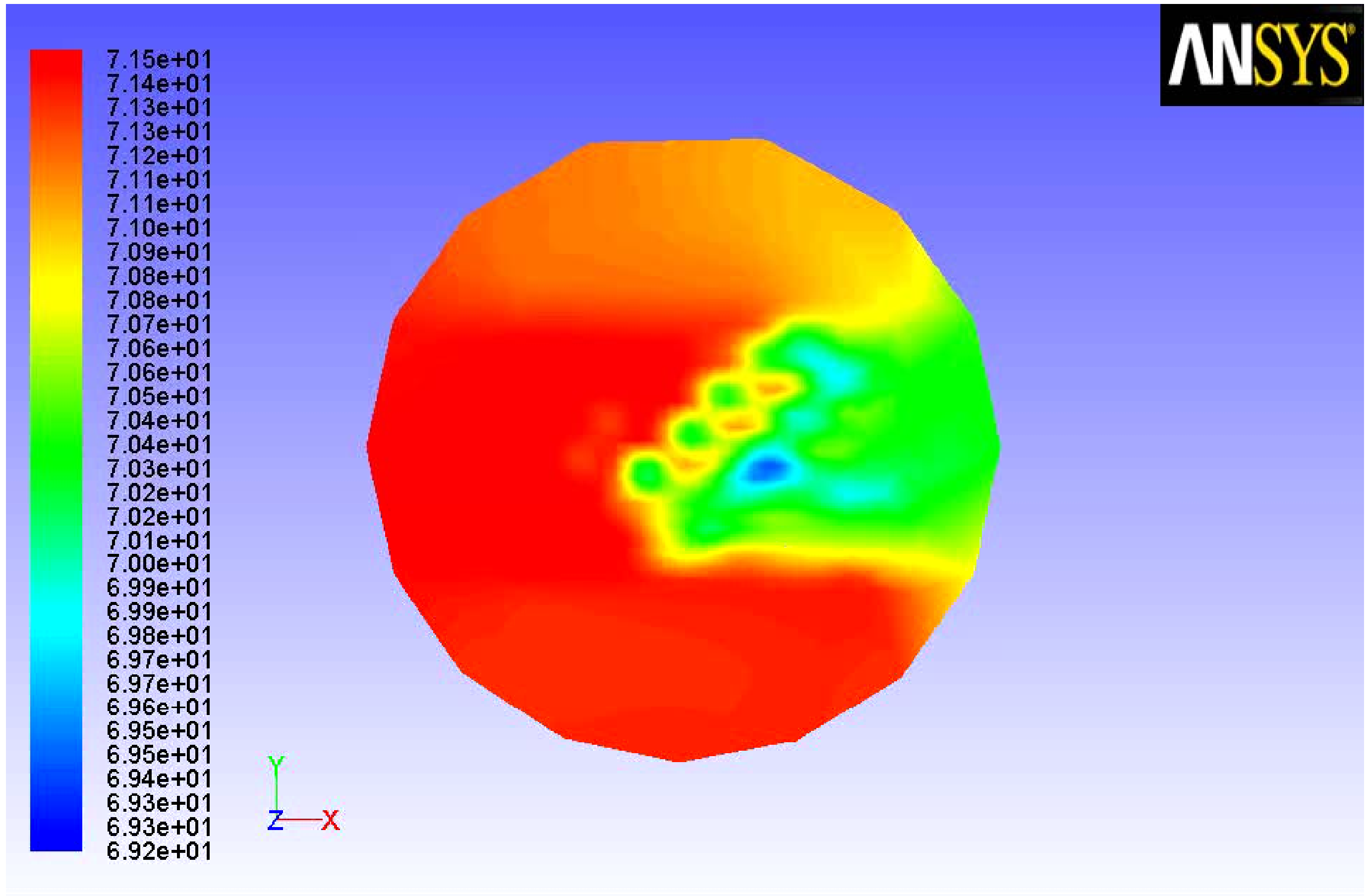
LH2
Flow →



Cold

Warm

Hot



Contours of Density (kg/m3) (Time=2.0201e-01)

Feb 19, 2009
FLUENT 12.0 (3d, dp, pbns, rke, transient)

Q_{weak} Accident Scenarios

4 kg of LH_2 in 2 metal boundaries

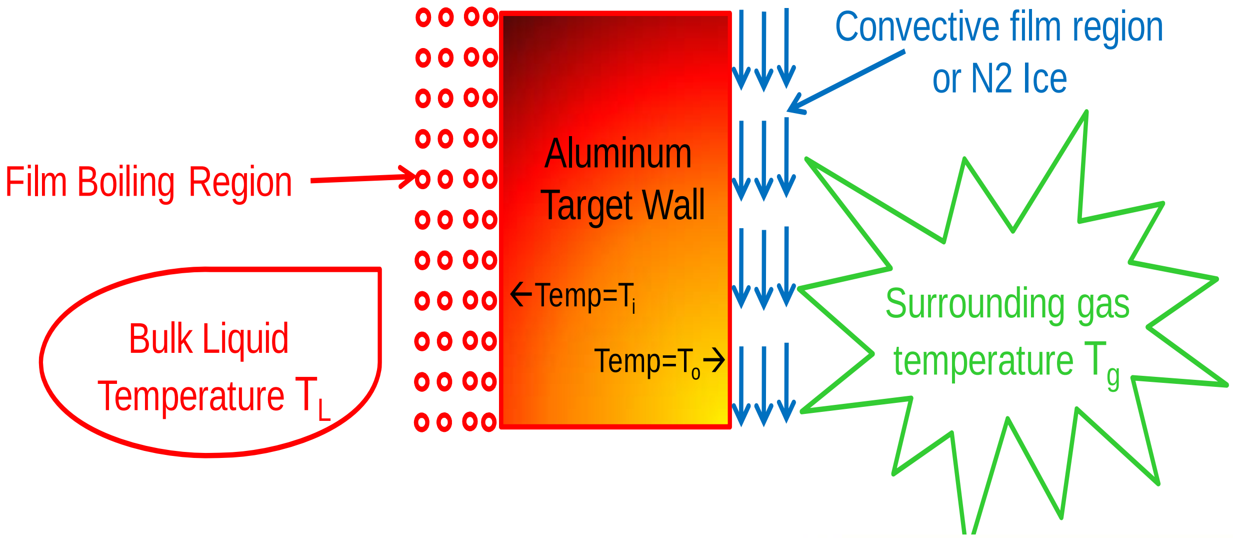
- **Relief (Sudden Loss of Vacuum): (SC breaks, but not loop)**
  - 105 g/s, $\Delta p = 1$ atm (ø pipe > 2")
- **Vent (cryo-loop breaks), with *CFD*: (loop breaks, but not SC)**
  - 210 g/s, $\Delta p < 1$ atm (ø pipe > 4")
- **Release: H2 escapes into Hall C (26×10^6 liters) (both break)**
  - ODH: none
  - Flammability: possible (556 MJ from burning 4 kg of hydrogen)
  - If it does burn: ΔT (avg) is only 17 C

Hydrogen concentration in normal air:

$4 < c_V < 74$ % : deflagration ⟶ Sub-sonic waves

$18 < c_V < 54$ % : detonation ⟶ Shock Waves

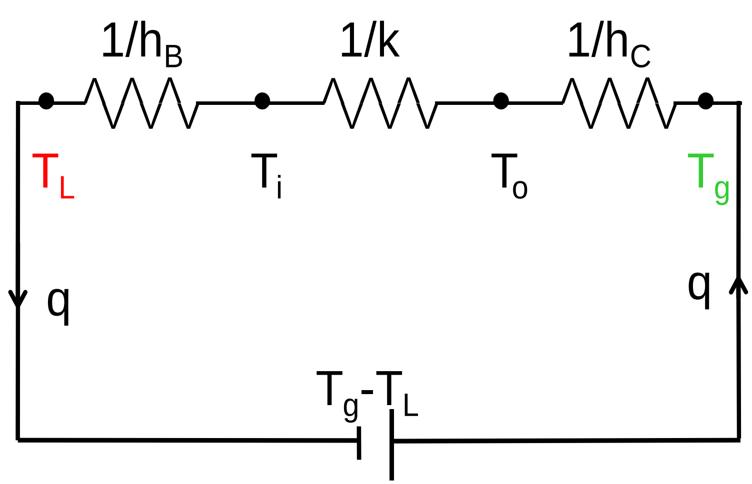
Safety Calculations



2 Approaches:

- **Schmitt, 1990.**
  Full calc. Conservative.
  Numerical solution.
    - **Bates, G0 @JLab**

- **Seely, 2004.**
  More realistic assumptions, less conservative.
  Analytic solution.
    - **JLab standard tgts**



Thermal	Electrical
q(power/area)	I (current)
ΔT (Temperature)	V (Voltage)
1/U (heat xfer coeff)	R (resistance)
$q=U(T_g - T_L)$	I=V/R (Ohm's Law)

Safety on the Back of an Envelope

Boiloff Mass Flow & Duration :

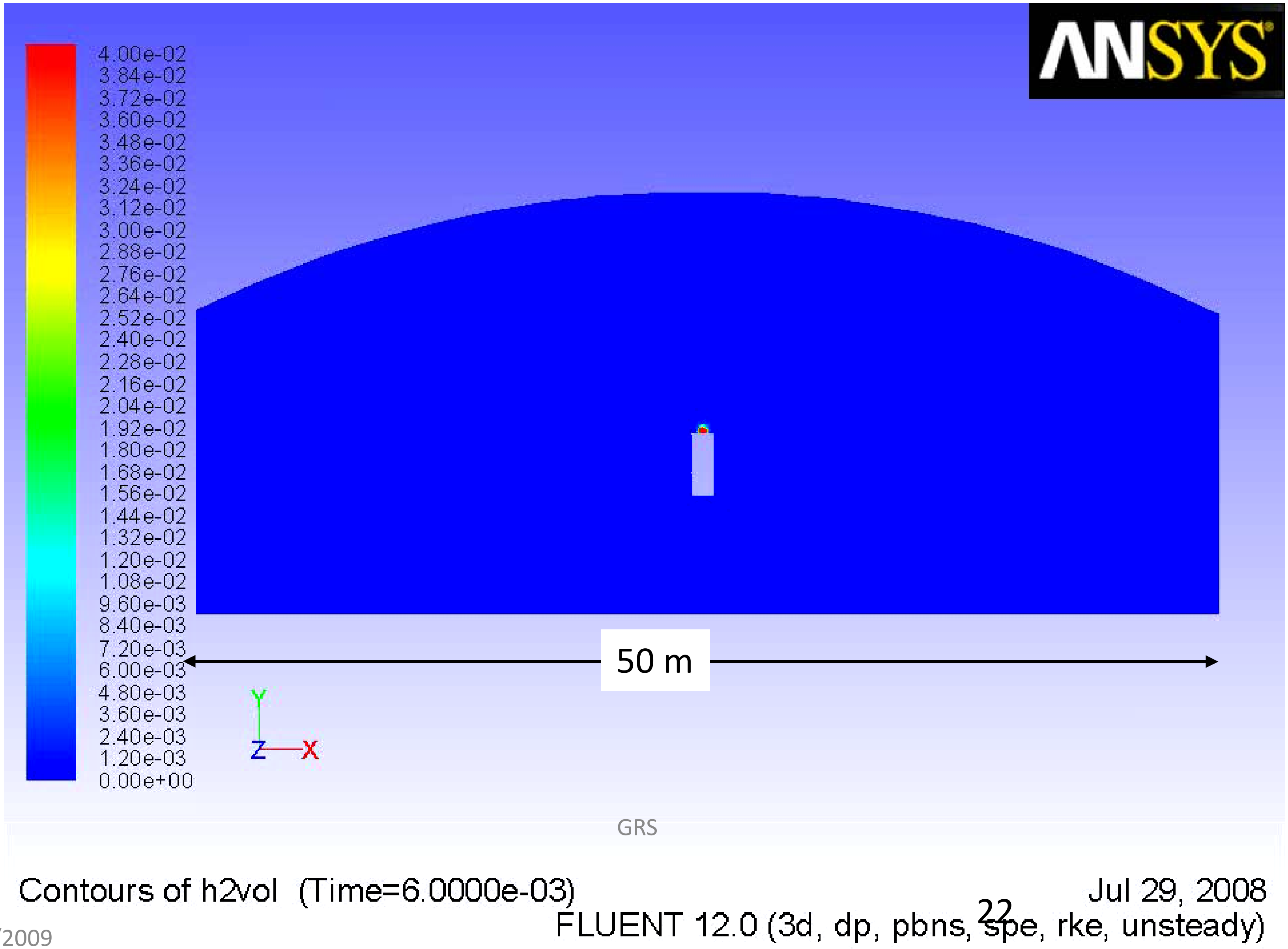
- Heat flux h ~ 8 to 15 kW/m^2, depending on target design (so take h ~ 15 kW/m^2)
- With A ~ 2 m^2, → heat xfer rate Q=30 kW
- Latent heat of vaporization h_v ~ 4.3x10^5 J/kg
- M ~ 3.7 kg (Qweak target)
- Then:
  - mass flow $\dot{m}$ = Q / h_v ~ 70 g/s
  - Boiloff time Δt =M/$\dot{m}$ ~ 53 s

Pipe Sizing & ΔP:

- Must stay subsonic 400 m/s < v_s < 1300 m/s
- V = $\dot{m}/(\rho * A_{pipe})$
  - Fastest warm, at 300K, where ρ=0.2 kg/m^3
    - 1" pipe → 700 m/s supersonic, NOT OK
    - 2" pipe → 170 m/s Subsonic → OK
- ΔP=f L/d v^2 ρ/2.
  - Take L=350', & f=0.023 (typ):
  - 1" pipe → ΔP= 650 psi (NOT OK)
  - 2" pipe → ΔP= 20 psi (OK)

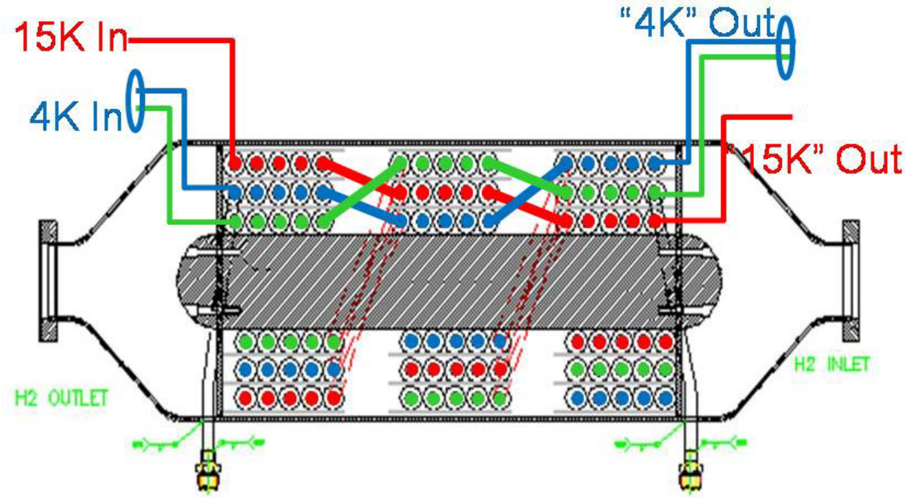
Deflagration Boundary (4%), Top Release

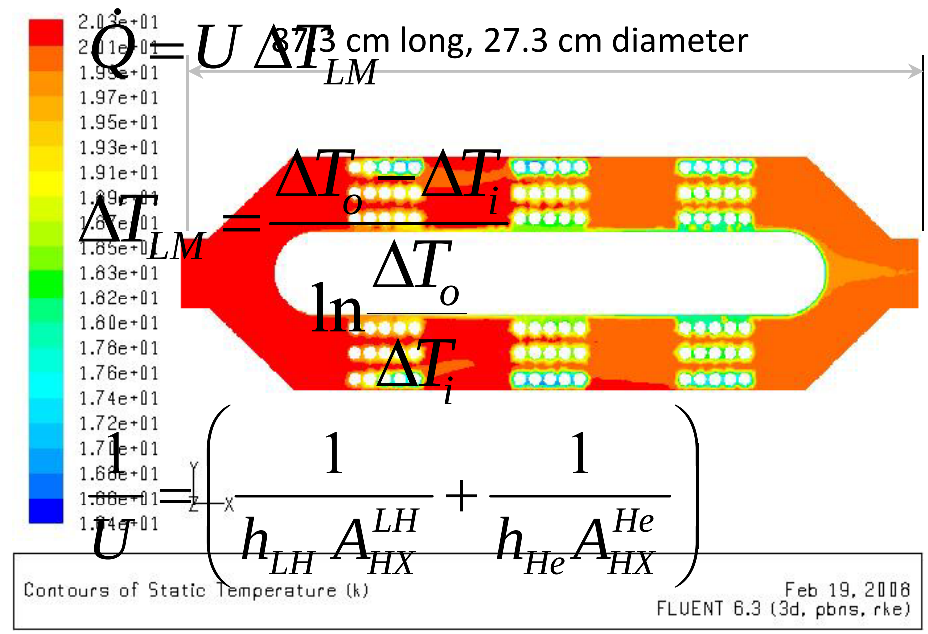
CFD simulation by S. Covrig, Jlab



3 kW Heat Exchanger

- *Cooling Power 3000 W!*
- **Combine capabilities of both 4K and 15K refrigerators → hybrid HX**
- **4 K: 2 layers, 2.5 kW @25 g/s**
- **15 K: 1 layer, 500W @17g/s**
- **24 liters of LH_2.**
- **CFD: head & freezing.**
  - **Head: 0.6 psi @ 1 kg/s**
  - **Doesn't freeze despite 4K coolant**
- **Basic design performance calculated analytically (counterflow HX):**



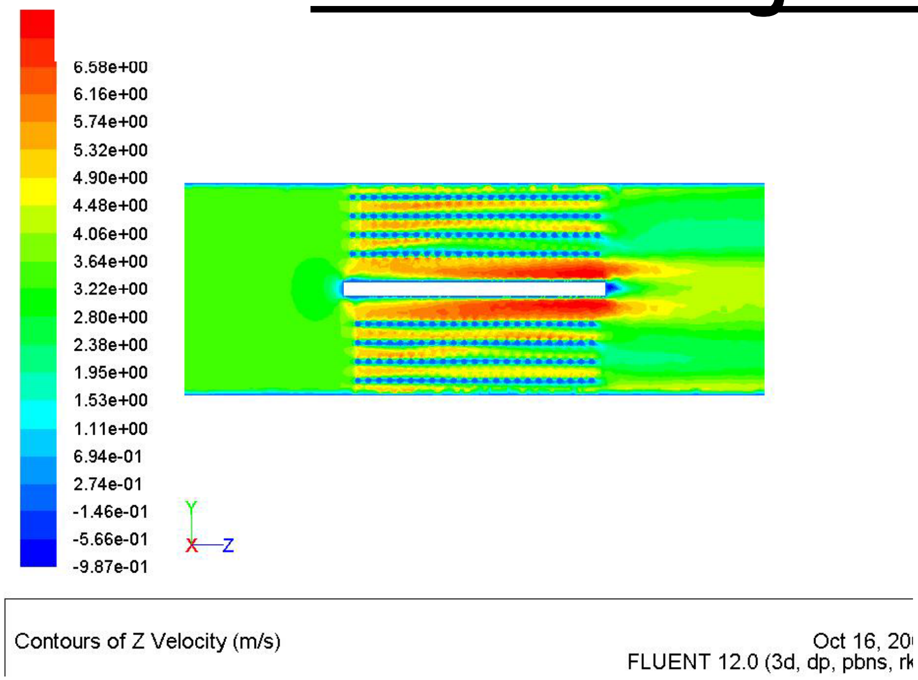


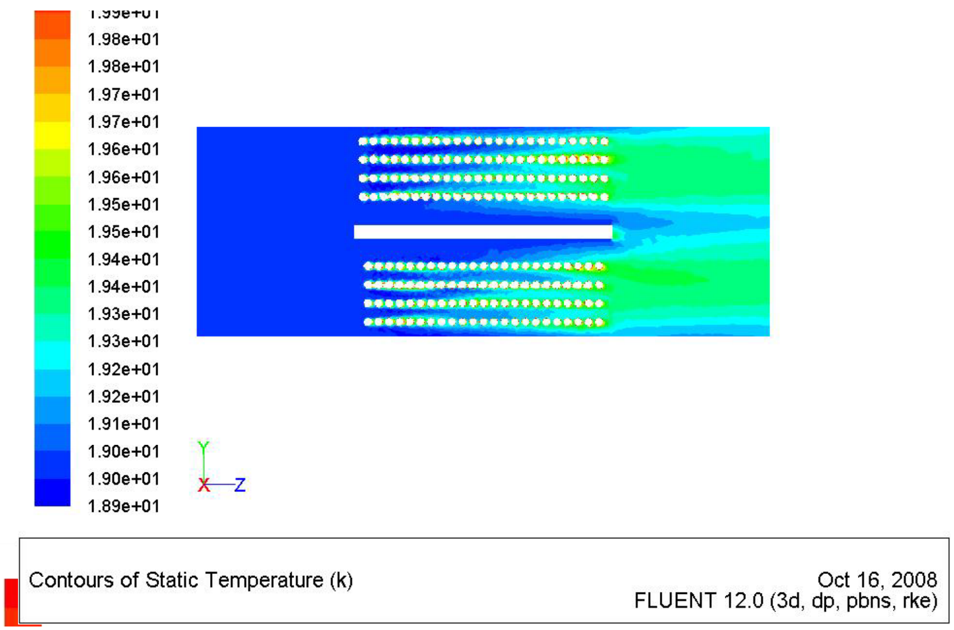
$$\dot{Q} = U\,\Delta T_{LM}$$

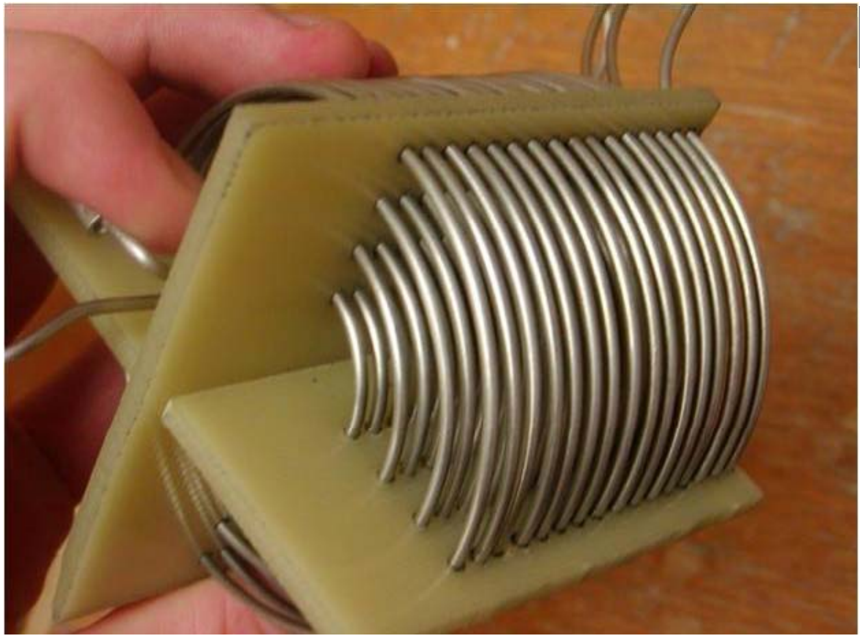
$$\Delta T_{LM} = \frac{\Delta T_o - \Delta T_i}{\ln \frac{\Delta T_o}{\Delta T_i}}$$

$$\frac{1}{U} = \left(\frac{1}{h_{LH} A_{HX}^{LH}} + \frac{1}{h_{He} A_{HX}^{He}} \right)$$

Heater Design: 2 vs 4 layer Heater







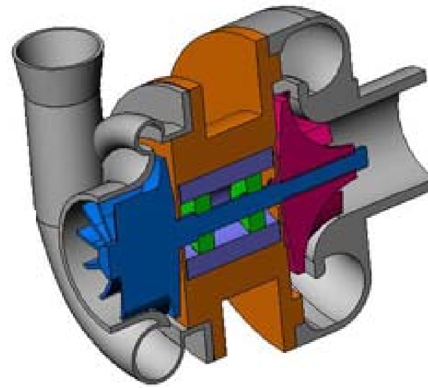
No boiling

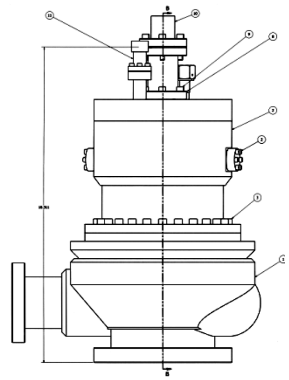
ΔP=0.3 psi

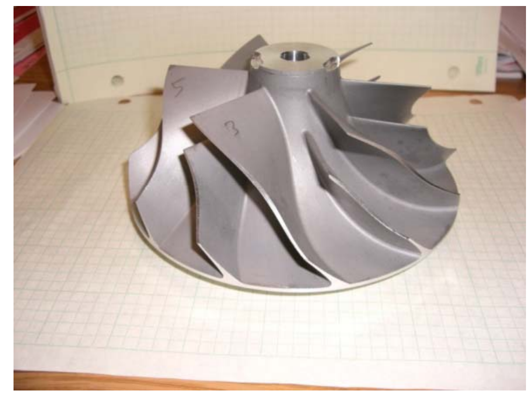
2500 W

1.2 Ω

Pump

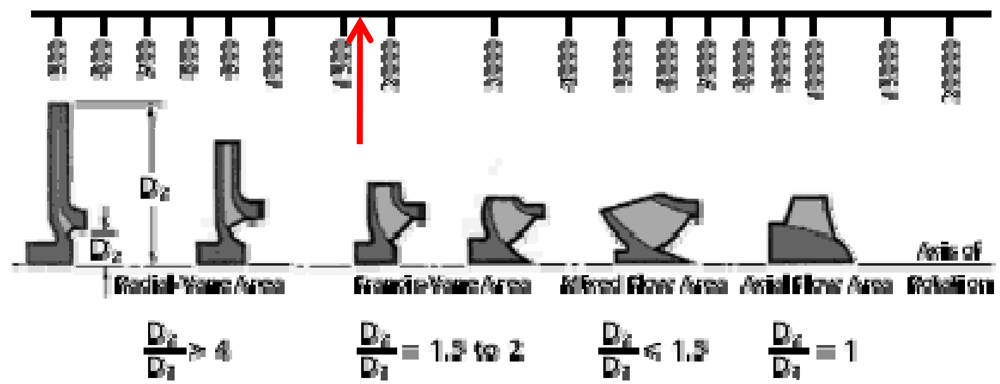






- Two Crucial design parameters:
  - Pressure head & mass flow
    - Together these also fix viscous heating loss!
- Mass flow:
  - Direct impact on boiling
- Head:
  - From CFD, and/or analytically
- Specific Speed dictates impeller design:

$$N_s = \frac{N(\mathrm{rpm})\sqrt{Q(\mathrm{gpm})}}{g\left[H(\mathrm{feet})\right]^{0.75}}$$



- Commercial pump options (Barber-Nichols)

Backgrounds

- Cell window backgrounds
  - Mitigate with thin nipple over beam area
    - Improves heat conduction too
  - Thinnest Al ~ 0.003" (76 μ)
  - Be never used before… plan to try it for Qweak
    - Nominally 2* Al thickness
    - A(Al) ~ A(Be)
- Ice - can be a potential problem
  - Possible explanation for a discrepancy in msrd/simulated Al bkg yield in G0-bkwrd

Summary

- **Next generation targets: design with CFD**
  - **next challenge: 5 kW, 25 ppm Moller @ JLab**
- **There are some hard boundaries**
  - **Flowrate & head tradeoffs (viscous heating)**
  - **P & T can only be pushed so far**
  - **Flowrate, cooling power limitations**
  - **Window heating**
  - **Subcooling only works so far**
- **Present R&D:**
  - **Dynamic CFD → ΔA predictions**
    - **Impact of raster & helicity reversal frequencies**
    - **Dependence on flowrate in a fixed cell design**
  - **Two phase CFD**
  - **Qweak Target !**