**SOME CALCULATIONS PERTAINING
TO CENTRAL CALORIMETER COOLDOWN
RATE AND TEMPERATURE GRADIENTS**

TOM PETERSON

AUGUST 6, 1987

DØ ENGINEERING NOTE # 3740.000EN105

INTRODUCTION

For a first-pass look at how the central calorimeter module assemblies cool down in the CC cryostat, some "hand" calculations of free convection from the end plates were done. A local convection coefficient for heat transfer from the vertically oriented end plates to the argon gas is estimated to be about $1 \times 10^{-3} \text{ W/cm}^2\text{K}$. Even for free convection, flow is turbulent between the end plates and cryostat wall. Total rate of heat removal from both ends of the assembly at 300K to 100K argon gas is about 10KW. The maximum end plate temperature gradient based on convection alone (neglecting end plate conduction) is about 3 degree K per centimeter height.

FREE CONVECTION BETWEEN TWO VERTICAL PLATES

Some nice correlations are presented in the "Handbook of Heat Transfer Fundamentals" for free convection between vertical plates. No information about temperature gradients is obtained, but this probably provides a best estimate of total heat transferred and overall cooldown rate based on free convection from the end plates. Figure 1 illustrates the model for which the results in Table 1 were obtained.

If the gas flowing down circumferentially from the cooling coils over the module skins stays cold, but the end plates are warm, this calculation could give a good estimate of total heat transferred from the end plates to the gas. The overall heat transfer coefficient, h , varies only from $3.0 \times 10^{-4} \text{ W/cm}^2\text{K}$ to $4.2 \times 10^{-4} \text{ W/cm}^2\text{K}$ for the variety of temperature conditions considered; not a large variation. The variation of q_{total} results from the different average wall-to-gas temperature differences.

Once the assumed circulation pattern is established (if ever) the cryostat walls (which contain about 10% of the total heat) have been cooled more than the modules, so a situation like cases 3,4, or 6 applies. Cases 1 and 2 probably never occur since 100K gas would not get to the bottom with

everything at 300K. Therefore, I would use 20KW as a maximum cooldown rate and 10KW as typical.

Since q_{total} depends primarily on the wall-to-gas temperature difference, it could be resolved further by including a model for the heat flow out the modules to the end plates. The end plates themselves are "thin" for heat transfer purposes. The Biot number is ht/k where h is defined above, t is plate thickness, k is solid thermal conductivity = 0.15 W/cmK for stainless steel. It is a measure of the relative importance of internal resistance to heat flow to the convection resistance. For Biot number < 0.1 one may assume the temperature through the thickness of the end plate is uniform. We have $ht/k = (3 \times 10^{-4} \text{ W/cm}^2\text{K}) (1.75 \text{ in.} \times 2.54 \text{ cm/in}) / (0.15 \text{ W/cmK}) = 0.009$. Thus, a model for heat flow out of the modules can assume uniform temperature through the thickness of the endplates. The endplate temperature is a result of the balance between heat stored and heat entering the endplate from inside the module and heat removed by convection. I have not yet started on this problem of internal heat transfer. A finite elements analysis is probably required.

FREE CONVECTION FROM A LONE VERTICAL PLATE

For Rayleigh number greater than 800 in air, two parallel plates (as in the previous model) no longer interfere with heat transfer from one another, i.e., the boundary layers through which heat transfer occurs are separate. Therefore, although this is argon gas, the huge Rayleigh numbers in Table 1 no doubt imply that this is the case for the above model, and correlations for lone vertical plates may also be applied.

The model using a lone vertical plate is just like in Figure 1 with the plate at temperature T_1 completely removed from the picture in the lower left. Table 2 gives results analogous to those of Table 1. This model always ignores the cooling of the cryostat walls.

For $Gr > 10^{10}$ flow is fully turbulent, so flow is turbulent for these conditions. In turbulent flow the "local" convection coefficient is nearly constant over the surface. Thus, the values of h_c in Table 2 represent typical heat transfer coefficients for any location over the endplate. The local heat flux would be $q/\text{area} = h_c (T_w - T_{\text{gas}})$ where T_w is wall temperature at that elevation and T_{gas} is argon temperature at that elevation outside of the boundary layer. Using the q in Table 1 and the h_c in Table 2 the average difference between wall and gas temperature was found (listed in Table 2).

To clarify the difference between h , the convection coefficient for the parallel plates, and h_c for the lone vertical plate; they are defined differently. For the parallel plates the Nusselt number, $Nu = hs/k$, (see Table 1 for variable definitions) is correlated experimentally with Rayleigh number. h is defined as $q/2HW(T_w - T_{oo})$ where T_{oo} is the entering gas temperature. For the lone vertical plate Nusselt is correlated with $GrPr$ (which is the same as the Rayleigh number but with different characteristic lengths since there is no spacing, S). In that correlation $Nu = h_c H/k$ where h_c is defined as (q/area) locally on the plate $= h_c (T_w - T_{\text{gas}})$ local. Since flow is turbulent h_c is constant. Since the gas flow is induced by the overall buoyancy difference, $T_w - T_{oo}$ is used in the Gashof number.

COOLDOWN SUMMARY

A central calorimeter cooldown which occurred at an average rate of 10 KW would take a total of about 13 days. Total argon circulating mass flow at 10KW cooling rate, warming from 100K to 200K, would be $10000W / (0.53J/gK)(100K) = 190g/s$ or about 95g/s per side. Nitrogen consumption at 3atm would be at least $10000W / (183J/g) = 55g/s$ or 260 liters of liquid per hour.

MAXIMUM VERTICAL TEMPERATURE GRADIENT IN AN END PLATE

Figure 2 shows a model of one end plate for consideration of temperature gradients and a derivation of an expression for the maximum vertical temperature gradient neglecting conduction in the stainless steel. Using the largest convection coefficient, h_c , from Table 2, $h_c = 1.4 \times 10^{-3} \text{ W/cm}^2\text{K}$. Mass flow is at least 50 g/s per side (arbitrarily took half of previously calculated flow to conservatively estimate a maximum gradient), and wall temperature minus gas temperature is at most 200K. These numbers result in a maximum predicted gradient of 3K/cm.

CONCLUSIONS

Convection calculations from central calorimeter end plates predict convection coefficients of about $1 \times 10^{-3} \text{ W/cm}^2\text{K}$ from the end plates. Cooling rates are on the order of 10KW, resulting in about a 2 week cooldown, based on free convection to argon gas from the end plates. Temperature gradients for this type of gaseous cooldown are estimated to be less than 3K/cm neglecting conduction.

A finite elements heat transfer analysis could be done using some of these estimates to define boundary temperatures or heat fluxes. Such an analysis could much better estimate cooldown times and thermal gradients than I have here. These calculations should be viewed as order-of-magnitude at this time. Perhaps conduction could be studied using an existing ANSYS model. I will pursue these next steps as time allows unless I am directed otherwise.

Figure 1. Model for Free Convection in the Gap between End Plates and Cryostat

Sketch of assembly showing possible circulation pattern.

Actual gap on ends = 10 cm to 14 cm
Use 12.7 cm (5") avg

Actual end plate area $\approx 13.7 \text{ m}^2$

This at each end.

Model area = $4.06 \times 2.8 = 11.4 \text{ m}^2$

Table 1. Results for Free Convection Between Plates and Cryostat Wall.

Number	T_{∞} (K)	T_1 (K)	T_2 (K)	Ra	Nu	$\bar{h}$ (W/cm ² K)	q_{total} for both ends (KW)	Notes
1	100	300	300	1.9×10^7	40.7	4.2×10^{-4}	38.2	includes cooling cryo-walls
2	150	300	300	6.4×10^6	31.1	3.5×10^{-4}	23.9	includes cooling cryo-walls
3	100	300	100	2.2×10^7	42.6	3.4×10^{-4}	15.5	cryostat walls cold
4	150	300	150	5.9×10^6	30.6	3.0×10^{-4}	10.3	cryostat walls cold
5	100	200	200	2.2×10^7	42.6	3.4×10^{-4}	15.5	includes cooling cryo-walls
6	100	200	100	2.7×10^7	44.8	3.0×10^{-4}	6.8	cryostat walls cold

T_{∞} = gas temperature entering the channel at the bottom

T_1 = module end plate temperature

T_2 = cryostat wall temperature

Ra = Rayleigh number = $g \beta (\bar{T}_w - T_{\infty}) S^3 / \nu \alpha H$

where g = gravitational acceleration

β = gas thermal expansion coefficient

$\bar{T}_w = (T_1 + T_2) / 2$

S = spacing = 12.7 cm

ν = kinematic viscosity (viscosity $\div$ density)

α = thermal diffusivity

H = height = 406 cm

Table 1. (continued)

Nu = Nusselt number = $0.62 Ra^{1/4}$ (from experiment)

$\bar{h}$ = Overall convection coefficient = $Nu \times k / S$

where k = fluid thermal conductivity

S = spacing = 12.7 cm

q = Rate of heat transfer per flow channel =
= $2 H W \bar{h} (\bar{T}_w - T_\infty)$ (actually this defines $\bar{h}$)

where W = width = 280 cm

and the other variables are as previously defined.

$q_{total} = 2q$

Table 2. Results for Free Convection from a Lone Vertical Plate Representing the Assembly of Module End Plates.

T_{∞} (K)	T_w (K)	Gr	Pr	Nu	h_c (W/cm ² K)	average $T_w - T_{gas}$ (K)
100	300	3.07×10^{13}	0.639	4.36×10^3	1.41×10^{-3}	48
150	300	1.04×10^{13}	0.636	2.82×10^3	1.00×10^{-3}	45
100	200	3.06×10^{13}	0.656	4.40×10^3	1.10×10^{-3}	27

T_{∞} = gas temperature approaching the bottom of the plate

T_w = plate temperature

Gr = Grashof number = $g \beta (T_w - T_{\infty}) H^3 / \nu^2$

where g = gravitational acceleration

β = gas thermal expansion coefficient

H = plate height = 406 cm

ν = kinematic viscosity

Pr = Prandtl number = $c_p \mu / k$

where c_p = heat capacity of gas

μ = viscosity

k = gas thermal conductivity

Nu = Nusselt number = $0.021 (Gr Pr)^{2/5}$ (from experiment)

h_c = Average convection coefficient = $Nu \times k / H$

Average $T_w - T_{gas} = q / (HW h_c)$

Table 2. (continued)

where q = heat transfer from end plates at one end only from Table 1.

H = height = 406 cm

W = width = 280 cm

h_c = convection coefficient in Table 2

T_{gas} = gas temperature outside of boundary layer

Figure 2.

Estimate of Upper Limit for Wall Temperature Gradient for Convection Alone

argon gas flowing past surface
with flowrate $\dot{m}$
heat capacity C_p
convection coefficient h_c

Energy balance on gas gives

$$\dot{E}_{in} = \dot{m} C_p T_g(x) + h_c W \Delta x (T_w(x) - T_g(x))$$

$$\dot{E}_{out} = \dot{m} C_p T_g(x + \Delta x)$$

Setting $\dot{E}_{in} = \dot{E}_{out}$, rearranging, letting $\Delta x \rightarrow 0$

gives

$$\frac{\partial T_g}{\partial x} = \frac{h_c W}{\dot{m} C_p} (T_w(x) - T_g(x)) .$$

Since the steepest change in wall temperature would occur with very good heat transfer, I will postulate that for the most extreme rate of change of wall temperature with respect to height, the gas temperature closely follows the wall temperature.