SNS Linac Technical Memo

DTL Beam Position Monitor Heat Transfer Analysis

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WBS 1.4.5.2.1.4 Diagnostics – BPM & Phase

Introduction

This analysis was done in order to characterize the heat load the Beam Position Monitor (BPM) would experience encased within the drift tube. One of the considerations of the BPM design is that the feed through being used to take the signal from the electrode, inside the vacuum, out through the BPM body has temperature limitations. The feed through under consideration has a 573 K temperature limitation. This analysis relies on the work done by Lucie Parietti to calculate the expected heat load on all the other drift tubes in the linac. Under normal operating conditions, Lucie determined that the maximum temperature rise above room temperature of any drift tube would be approximately 10 K. It seemed reasonable to assume that the BPM would not be any different, but I felt that as a matter of completeness, checking the temperature distribution in the BPM would be an appropriate thing to do.

This work is based on the work Lucie did for the other drift tubes. The loads are specifically taken from the models she did for tank 2 and tank 6 drift tubes. Lucie was concerned with the worst case loading, or drift tubes having the highest levels of heat flux, so the models she developed were based on the largest drift tubes in each tank. These drift tubes had the most intense power deposition. Lucie obtained the loads for the drift tubes from Jim Billen's SUPERFISH runs. Since she didn't have the loads specific to the drift tubes containing BPMs. I placed the BPMs in the drift tubes for which she did have the heating loads. The models being analyzed are not exactly the same as the BPM drift tubes to be manufactured, but it is believed that the "worst case" loading will provide a more conservative model. The purpose of the analysis was to see if there were any potential problems with the BPM design.

Two different 3-D models were created, one for the tank 2 drift tubes (smallest) and one for the tank 6 drift tubes (largest). The models include the copper BPM cover and copper BPM electrode inside the copper drift tube. The feed through sub-assembly is not modeled. Two different load cases were run for each model. The first load case had the full water flowing through the drift tube. The second load case had one of the water channels blocked off, as if there were no water flowing through it.

Assumptions

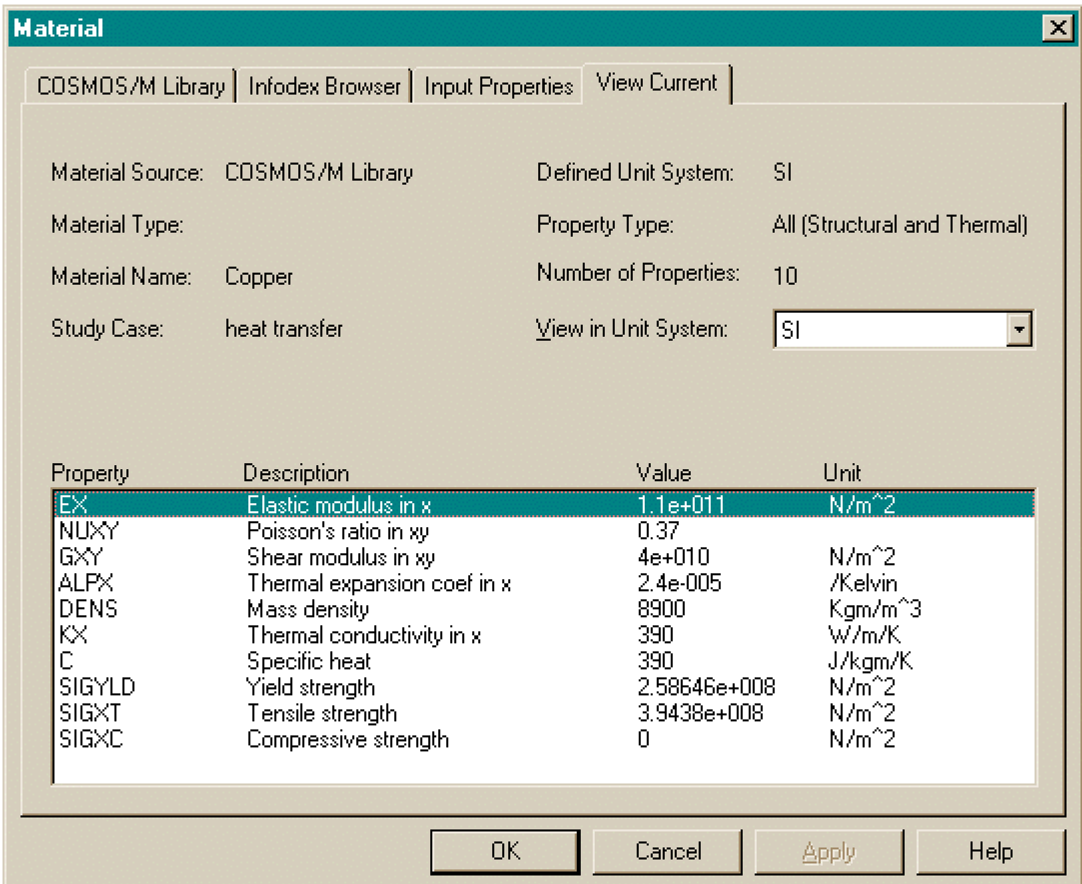
The following assumptions were made for the analysis:

1. Internal, fully developed, flow.
2. Steady state conditions.
3. Constant surface heat flux.

4. Constant properties.
5. Inlet water temperature at 293 K.
6. Initial temperature of drift tube is at 294 K.

Material Properties

The material properties for the copper pieces are listed in figure 1 table below.



Property	Description	Value	Unit
EX	Elastic modulus in x	1.1e+011	N/m^2
NUXY	Poisson's ratio in xy	0.37	
GXY	Shear modulus in xy	4e+010	N/m^2
ALPX	Thermal expansion coef in x	2.4e-005	/Kelvin
DENS	Mass density	8900	Kgm/m^3
KX	Thermal conductivity in x	390	W/m/K
C	Specific heat	390	J/kgm/K
SIGYLD	Yield strength	2.58646e+008	N/m^2
SIGXT	Tensile strength	3.9438e+008	N/m^2
SIGXC	Compressive strength	0	N/m^2

Figure 1, Copper material properties

Loads and Boundary Conditions for BPM in Tank #2 Drift Tube

As stated earlier the heat flux loads were determined from the SUPERFISH model and translated into COSMOS. The heat flux is applied to the appropriate surfaces of this model. The maximum heat flux value is 27370 W/m^2 and is applied to the center surface of the drift tube. The smallest value of the heat flux is 245.1 W/m^2 and is applied to the surface near the inner diameter of the drift tube. The values applied to the surfaces between these two vary from the maximum to the minimum. Also note that the loads were applied to the drift tube surface symmetrically about the longitudinal centerline.

The water channels provide the cooling to the drift tube. The boundary conditions applied to the cooling channel walls are therefore convection boundary conditions having the following properties:

1. Convection film coefficient – h = 6170 W/(m^2 K)
2. Bulk fluid temperature (K) = 294 K

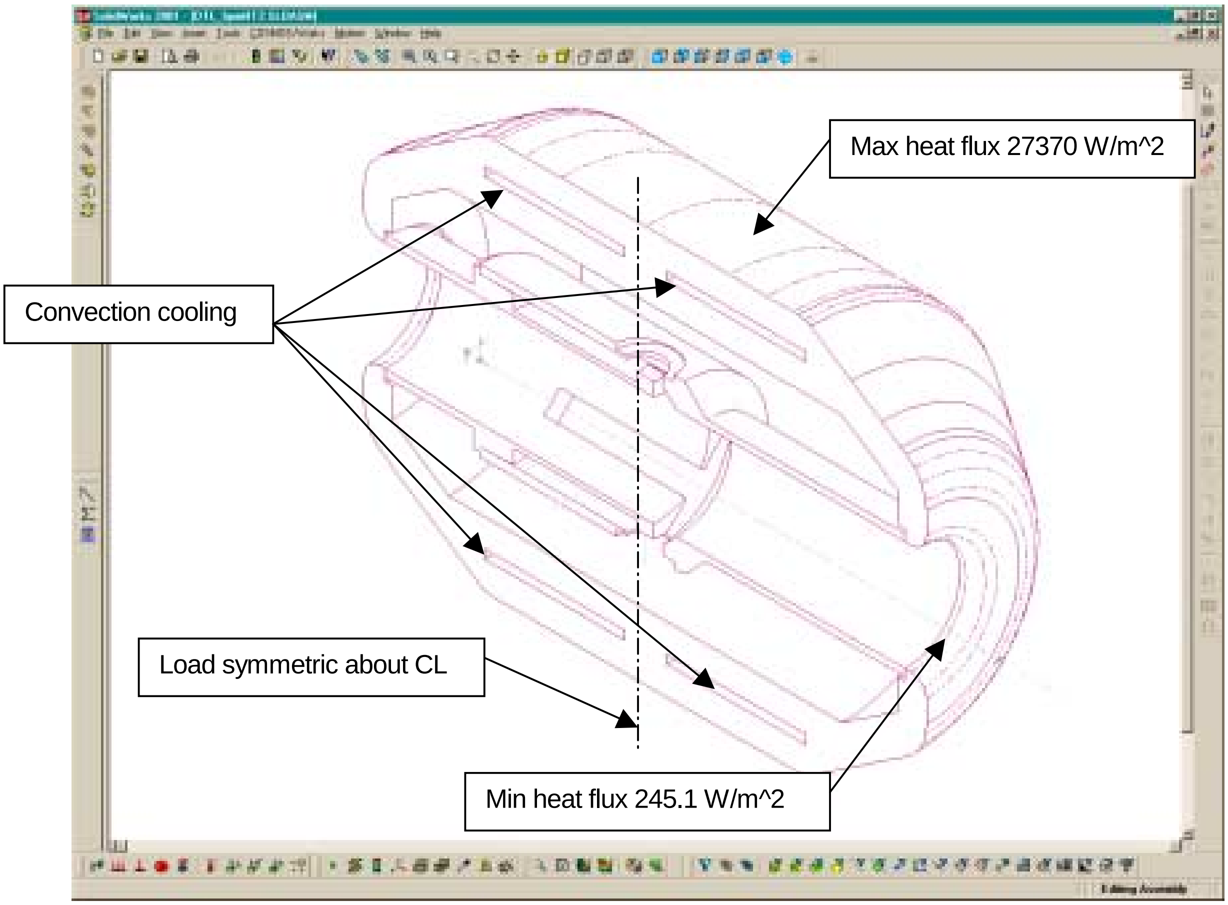


Figure 2, Boundary conditions

Analysis for BPM in Tank #2 Drift Tube

The analysis begins with the determination of the convection film coefficient. The following illustrates the process used to determine this coefficient. Note that I get a slightly higher value for the convection film coefficient compared to Lucie's value. Since the value Lucie calculated is smaller, more conservative, (6170 to 6340) I use her value in the analysis.

Based on the work done by Lucie Parietti for the drift tube heat transfer analysis.
From chapter 8 in Incorpera and Dewitt for an incompressible liquid:

$$q_{conv} := m_{dot} \cdot c_p \cdot \left(T_{mo} - T_{mi} \right)$$

This equation applies "irrespective of the nature of the surface thermal condition or tube flow conditions". From the beam properties defined in file: Injector_screen_1.mcd the heat load to the screen is determined.

$$q_{conv} := 628.04\,\text{watt}$$

This is the number Lucie used for the biggest Drift tube in tank #2, highest heat load.

Some Assumptions about the problem:

1. Fluid is water.
2. The flow rate of the water is 2.2 gpm and there are 2 channels, so for 1 channel you have:

$$\text{flow} := \frac{2.2}{2}\,\frac{\text{gal}}{\text{min}} \qquad \text{flow} = 6.94 \times 10^{-5}\,m^3\,s^{-1}$$

$$v_f := .001002\,\frac{m^3}{kg}$$

$$m_{dot} := \frac{\text{flow}}{v_f} \qquad m_{dot} = 0.069\,kg\,s^{-1}$$

The specific heat of the water is temperature dependent so an assumption must be made about the arithmetic mean temperature, assuming T_{mean} is 300:

$$c_p := 4179\,\frac{J}{kg \cdot K}$$

$$T_{mi} := 293\,K$$

$$T_{mo} := \frac{q_{conv}}{m_{dot} \cdot c_p} + T_{mi} \qquad T_{mo} = 295.17\,K$$

The properties of the fluid should be taken at the arithmetic mean temperature, therefore the above calculation should be iterated until c_p relates to the appropriate mean temperature.

$$T_{mean} := \frac{T_{mo} + T_{mi}}{2} \qquad T_{mean} = 294.085\,K$$

$$c_p := 4181\,\frac{J}{kg \cdot K} \qquad c_p = 0.999\,\frac{BTU}{lb\,R}$$

The water is flowing in two 3.175 cm wide by .2 cm deep channels. Then the velocity of the water can be determined, this velocity should not exceed 15 ft/sec. Velocities above 15 ft/sec cause erosion in copper.

$$\text{area} := 3.175\,cm \cdot .2\,cm$$

$$P := 2 \cdot (3.175 + .2)\,cm$$

$$\text{area} = 0.098\,in^2 \qquad \text{area} = 0.635\,cm^2$$

$$D_h := 4 \cdot \frac{\text{area}}{P}$$

$A_c := \pi \cdot \frac{D_h^{\,2}}{4}$ $\qquad A_c = 0.017\,in^2$

$D_h = 0.148\,in$ $\qquad D_h = 0.376\,cm$

$V := \frac{flow}{area}$ $\qquad V = 3.586\,\frac{ft}{s}$ $\qquad V = 109.29\,\frac{cm}{s}$

Calculating the Reynolds number will allow the determination of the type of flow.

$\rho := \frac{1}{v_f}$

$\mu := .000959\,\frac{N \cdot s}{m^2}$

$Re_D := \frac{\rho \cdot V \cdot D_h}{\mu}$ $\qquad Re_D = 4.28 \times 10^3$

This number should be greater than 4000 for fully turbulent flow. Another important dimensionless characteristic is the Prandtl number.

$k := 0.606\,\frac{watt}{m \cdot K}$

$Pr := \frac{c_p \cdot \mu}{k}$ $\qquad Pr = 6.616$

Using the Dittus-Boelter equation

ReD >= 10,000, 0.7<=Pr<=160, L/D>=10

(there are other experimental correlations) the average Nusselt number can be calculated. Pr is raised to the 0.4 when the water is heated and 0.3 is used when the water is being cooled.

$Nu_D := 0.023 \cdot Re_D^{\frac{4}{5}} \cdot Pr^{0.4}$ $\qquad Nu_D = 39.366$

$k := .606\,\frac{watt}{m \cdot K}$ $\qquad k = 0.029\,\frac{BTU}{hr \cdot in \cdot R}$

With the Nusselt number the average convection coefficient can be found.

$h := \frac{Nu_D \cdot k}{D_h}$ $\qquad h = 0.634\,\frac{watt}{cm^2 \cdot K}$ $\qquad h = 7.753\,\frac{BTU}{hr \cdot in^2 \cdot R}$

Once the boundary conditions are established the finite element model can be run. This model is a 3-D, steady state, heat transfer model. The type of element used is a 10 node, second order, solid parabolic element. The model consists of 11, 647 total elements and 22,605 nodes. The meshed model is shown below in figure 3.

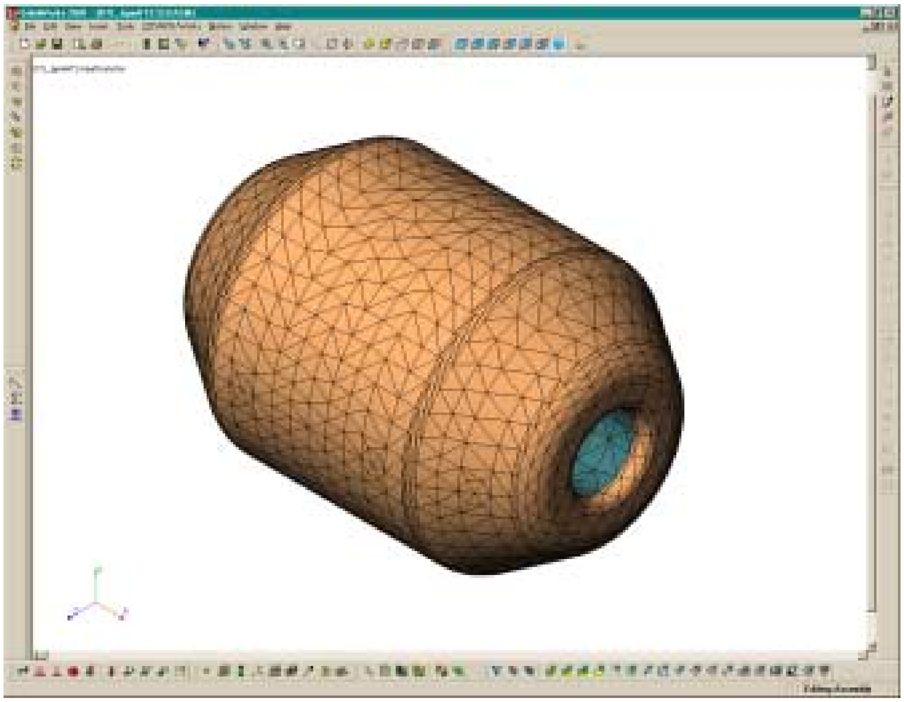


Figure 3, Meshed model

Results for BPM in Tank #2 Drift Tube

The resulting temperature distribution for the load case where water is flowing in both cooling channels is shown below. The maximum temperature in the BPM is 300.1 K.

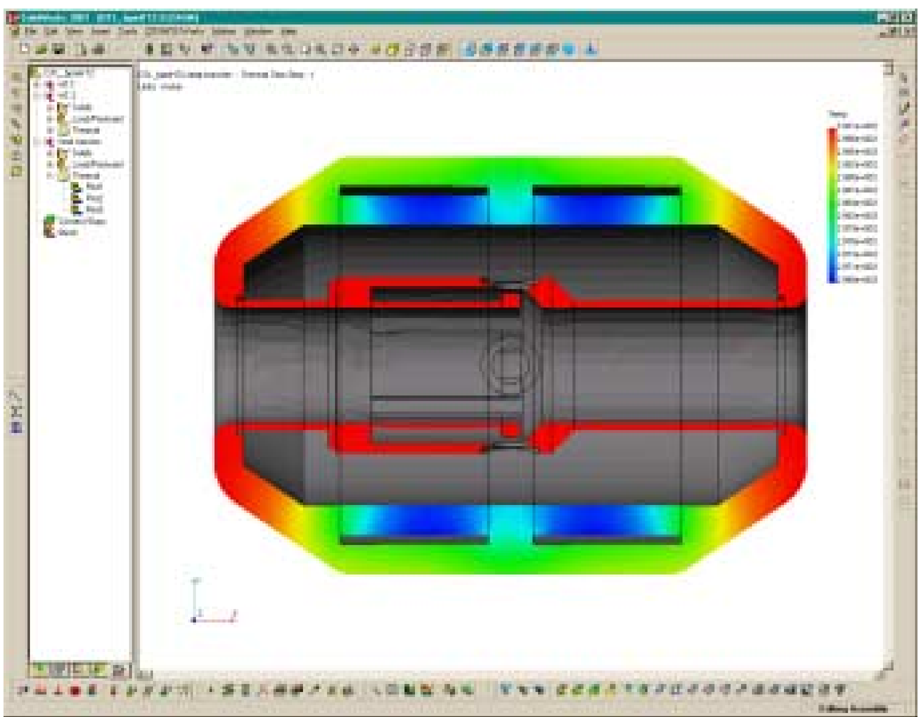


Figure 4, Water flowing in both cooling channels, Maximum temperature is 300.1 K

The second load case has one cooling channel blocked. The result of the second load case is shown in figure 5, the maximum temperature in the BPM is 314.4 K.

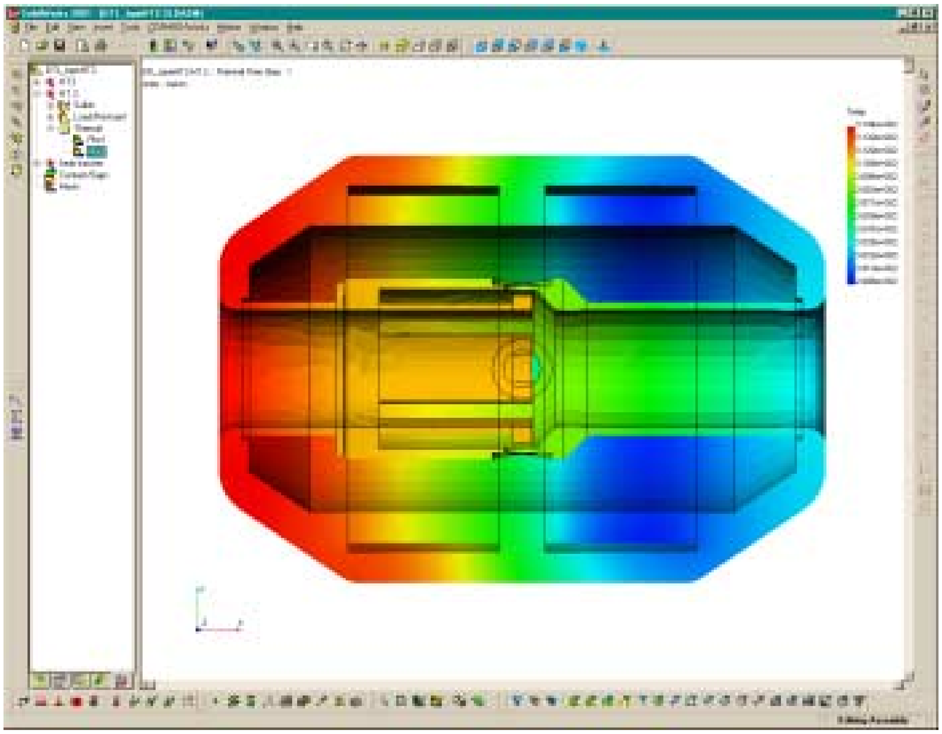


Figure 5, Water flow blocked in one cooling channel, Maximum temperaure is 314.4 K

Conclusions for BPM in Tank #2 Drift Tube

The first model shows that with everything working fine the expected temperature increase is only about 6 K. The second model shows that if only half of cooling water was flowing the temperature would increase 20.4 K. These models indicate that under normal operating conditions the BPM feed throughs will not be exposed to temperatures anywhere near their advertised limits. Even if there is an off normal condition and only half the cooling water is available the feed throughs should be fine. Of course if there were no cooling water at all, there would be a serious problem.

Loads and Boundary Conditions for BPM in Tank #6 Drift Tube

As stated earlier the heat flux loads were determined from the SUPERFISH model and translated into COSMOS. The heat flux is applied to the appropriate surfaces of this model. The maximum heat flux value is 27370 W/m^2 and is applied to the center surface of the drift tube. The smallest value of the heat flux is 245.1 W/m^2 and is applied to the surface near the inner diameter of the drift tube. The values applied to the surfaces between these two vary from the maximum to the minimum. Also note that the loads were applied to the drift tube surface symmetrically about the longitudinal centerline.

The water channels provide the cooling to the drift tube. The boundary conditions applied to the cooling channel walls are therefore convection boundary conditions having the following properties:

3. Convection film coefficient – h = 12100 W/(m^2 K)

4. Bulk fluid temperature (K) = 293 K

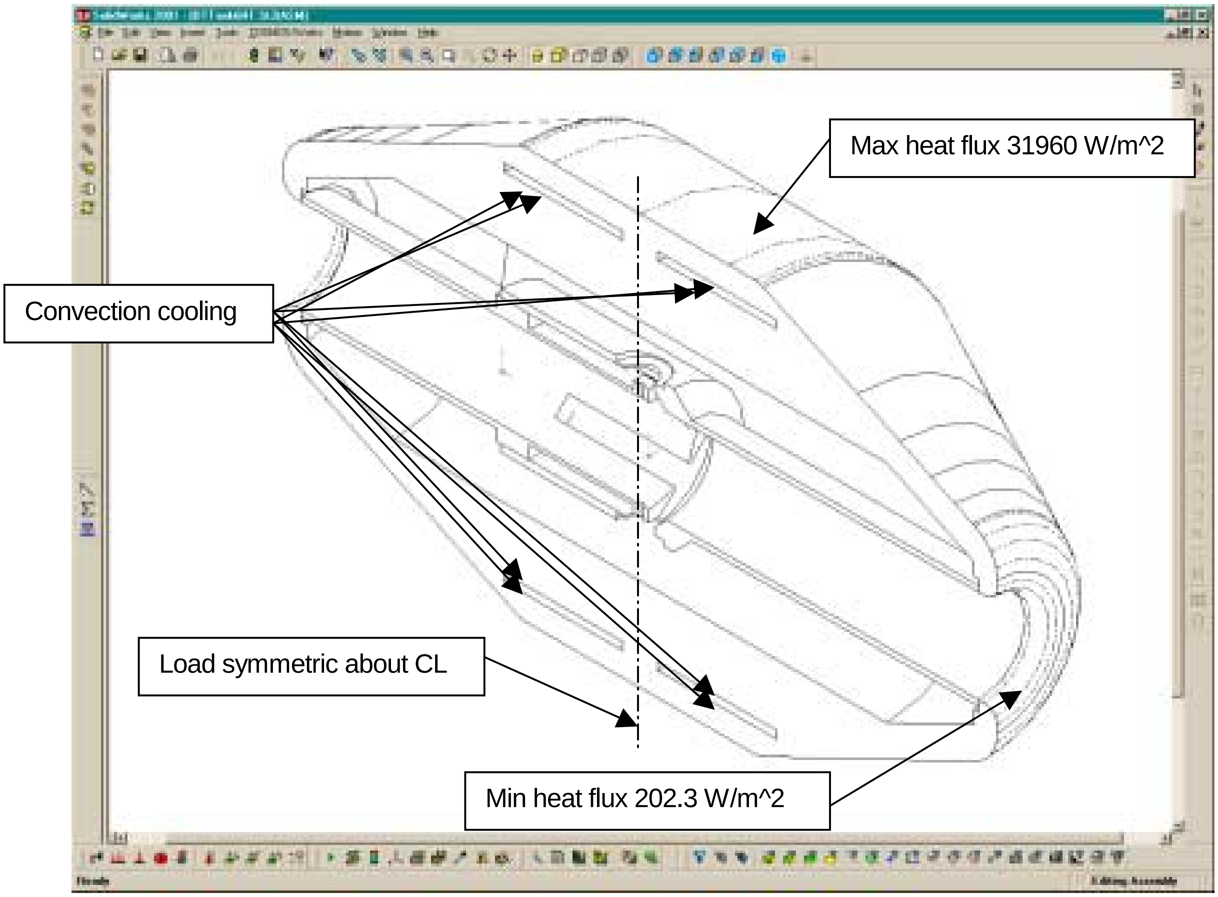


Figure 6, Boundary conditions Tank 6 drift tube BPM

Analysis for BPM in Tank #6 Drift Tube

The analysis begins with the determination of the convection film coefficient. The following illustrates the process used to determine this coefficient. Note that I get a slightly higher value for the convection film coefficient compared to Lucie's value. Since the value Lucie calculated is smaller, more conservative, (12400 to 12100) I use her value in the analysis.

Based on the work done by Lucie Parietti for the drift tube heat transfer analysis.
From chapter 8 in Incorpera and Dewitt for an incompressible liquid:

$$q_{conv} := m_{dot} \cdot c_p \cdot \left(T_{mo} - T_{mi}\right)$$

This equation applies "irrespective of the nature of the surface thermal condition or tube flow conditions". From the beam properties defined in file: Injector_screen_1.mcd the heat load to the screen is determined.
This is the number Lucie used for the biggest Drift tube in tank #6, highest heat load.

$$q_{conv} := 1159.69 \text{watt}$$

Some Assumptions about the problem:
1. Fluid is water.
2. The flow rate of the water is 5.1 gpm and there are 2 channels, so for 1 channel you have:

$$flow := \frac{5.1}{2} \frac{gal}{min} \qquad flow = 1.609 \times 10^{-4}\, m^3\, s^{-1}$$

$$v_f := .001002 \frac{m^3}{kg}$$

$$m_{dot} := \frac{flow}{v_f} \qquad m_{dot} = 0.161\, kg\, s^{-1}$$

The specific heat of the water is temperature dependent so an assumption must be made about the arithmetic mean temperature, assuming T_{mean} is 300:

$$c_p := 4179 \frac{J}{kg \cdot K}$$

$$T_{mi} := 293\, K$$

$$T_{mo} := \frac{q_{conv}}{m_{dot} \cdot c_p} + T_{mi} \qquad T_{mo} = 294.728\, K$$

The properties of the fluid should be taken at the arithmetic mean temperature, therefore the above calculation should be iterated until c_p relates to the appropriate mean temperature.

$$T_{mean} := \frac{T_{mo} + T_{mi}}{2} \qquad T_{mean} = 293.864\, K$$

$$c_p := 4181 \frac{J}{kg \cdot K} \qquad c_p = 0.999 \frac{BTU}{lb\, R}$$

The water is flowing in two 3.175 cm wide by .2 cm deep channels. Then the velocity of the water can be determined; this velocity should not exceed 15 ft/sec. Velocities above 15 ft/sec cause erosion in copper.

$$area := 3.175\, cm \cdot .2\, cm \qquad P := 2 \cdot (3.175 + .2)\, cm$$

$$area = 0.098\, in^2 \qquad area = 0.635\, cm^2$$

$$D_h := 4 \cdot \frac{area}{P}$$

$$A_c := \pi \cdot \frac{D_h^2}{4} \qquad A_c = 0.017\, in^2$$

$$D_h = 0.148\, in \qquad D_h = 0.376\, cm$$

$$V := \frac{flow}{area} \qquad V = 8.312 \frac{ft}{s} \qquad V = 253.354 \frac{cm}{s}$$

Calculating the Reynolds number will allow the determination of the type of flow.

$$\rho := \frac{1}{v_f}$$

$$\mu := .000959\ \frac{N \cdot s}{m^2}$$

$$Re_D := \frac{\rho \cdot V \cdot D_h}{\mu} \qquad Re_D = 9.921 \times 10^3$$

This number should be greater than 4000 for fully turbulent flow. Another important dimensionless characteristic is the Prandtl number.

$$k := 0.606\ \frac{watt}{m \cdot K}$$

$$Pr := \frac{c_p \cdot \mu}{k} \qquad Pr = 6.616$$

Using the Dittus-Boelter equation

ReD >= 10,000, 0.7<=Pr<=160, L/D>=10

(There are other experimental correlations) the average Nusselt number can be calculated.
Pr is raised to the 0.4 when the water is heated and 0.3 is used when the water is being cooled.

$$Nu_D := 0.023 \cdot Re_D^{\frac{4}{5}} \cdot Pr^{0.4} \qquad Nu_D = 77.132$$

$$k := .606\ \frac{watt}{m \cdot K} \qquad k = 0.029\ \frac{BTU}{hr \cdot in \cdot R}$$

With the Nusselt number the average convection coefficient can be found.

$$h := \frac{Nu_D \cdot k}{D_h} \qquad h = 1.242\frac{watt}{cm^2 \cdot K}$$

Once the boundary conditions are established the finite element model can be run. This model is a 3-D, steady state, heat transfer model. The type of element used is a 10 node, second order, solid parabolic element. The model consists of 30,590 total elements and 52,405 nodes. The meshed model is shown below in figure 7.

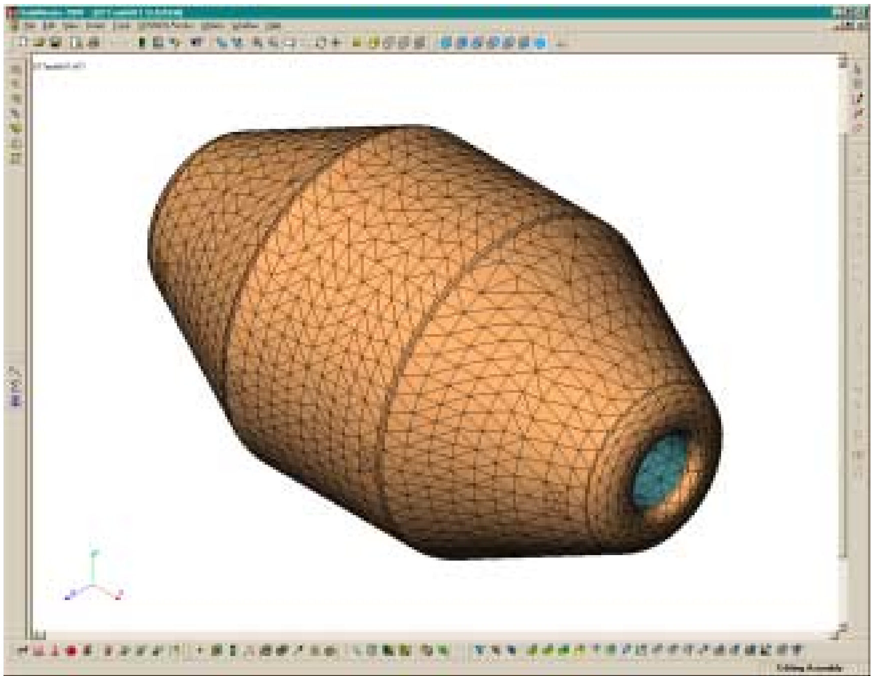


Figure 7, Meshed model, tank #6 drift tube BPM

Results for BPM in Tank #6 Drift Tube

The resulting temperature distribution for the load case where water is flowing in both cooling channels is shown below. The maximum temperature in the BPM is 304.1 K.

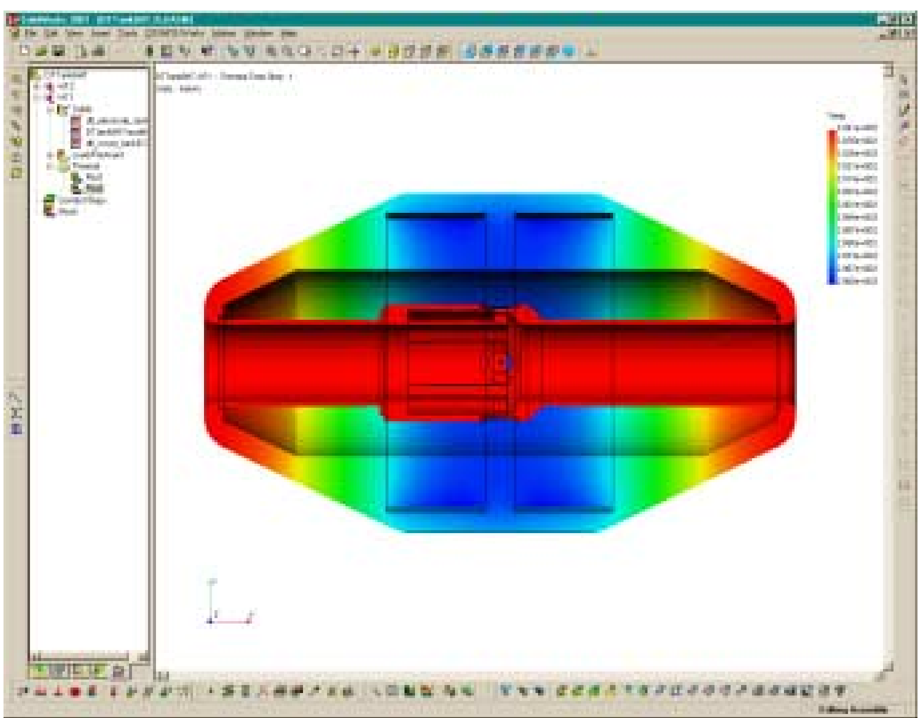


Figure 8, Water flowing in both cooling channels, Maximum temperature is 304.1 K

The second load case has one cooling channel blocked. The result of the second load case is shown in figure 5, the maximum temperature in the BPM is 317.3 K.

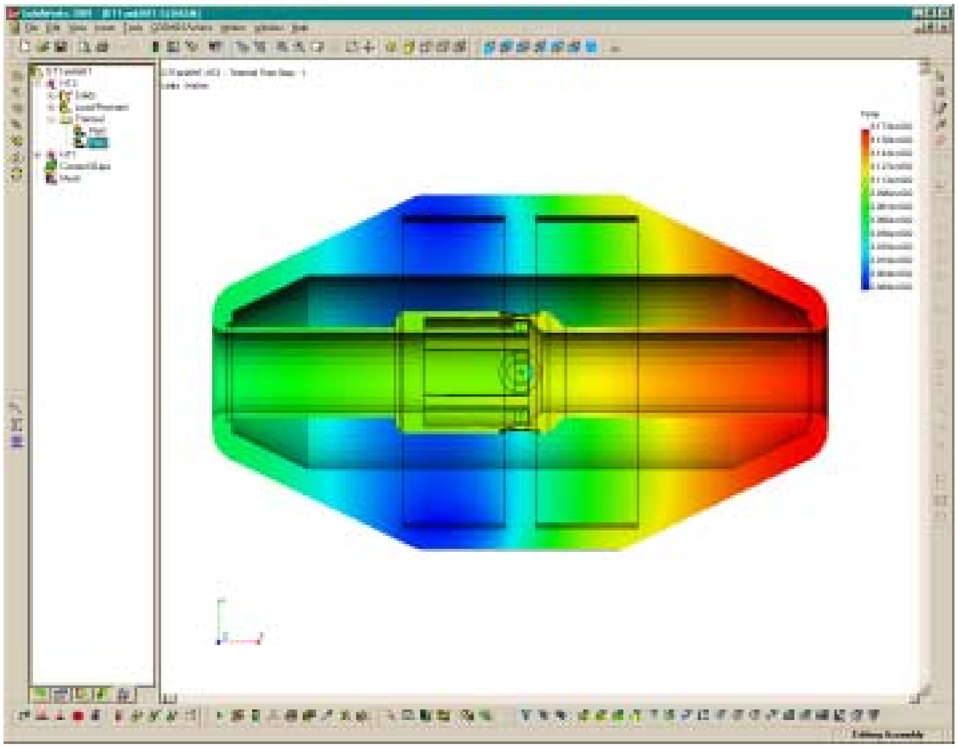


Figure 9, Water flow blocked in one cooling channel, Maximum temperaure is 317.3 K

Conclusions for BPM in Tank #6 Drift Tube

The first model shows that with everything working fine the expected temperature increase is only about 10.1 K. The second model shows that if only half of cooling water was flowing the temperature would increase 23.3 K. These models indicate that under normal operating conditions the BPM feed throughs will not be exposed to temperatures anywhere near their advertised limits. Even if there is an off normal condition and only half the cooling water is available the feed throughs should be fine.