Thermo-Mechanical Optimization of a Tritium Target

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Abstract

A deep inelastic scattering experiment on tritium and helium mirror nuclei is planned at Thomas Jefferson Lab National Accelerator Facility in order to gain better understanding of the down and up quark distribution in the proton. The experiment will expose a pressurized tritium target cell to a 12 GeV electron beam. Due to interactions between the target and the electron beam, the target will be subject to a variety of heat loads. A design which took these thermal loads into consideration was necessary to insure the target would not mechanically fail during operation. An iterative design process using thermal and mechanical finite element models was used to create a safe design for the target cell assembly.

Research Category: Mechanical Design

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Introduction

A deep inelastic scattering experiment on tritium and helium mirror nuclei is planned at Thomas Jefferson Lab National Accelerator Facility in order to gain better understanding of the down and up quark distribution in the proton. The experiment will expose a tritium target to the upgraded 12 GeV electron beam. A safe design is of primary concern, as tritium is a radiation hazard when inhaled or ingested. The tritium target is designed as a double wall container where the primary canister contains the tritium gas while the secondary container holds helium. A schematic design of the target is included in Fig. 1. The tritium gas will be initially pressurized to 10 atmospheres to ensure sufficient tritium density and favorable scattering cross sections. The secondary canister will be a completely sealed container pressurized to 5 atmospheres. The purpose of the helium is to relieve pressure on primary container, and to provide additional means of cooling.

During the operation, due to the interaction with the electron beam, a number of thermal loads will develop in container walls and in the sealed gasses. The approximately 38.9 W of generated heat is expected to be transmitted to the environment through the combined conductive, convective, and, if the metal walls reach substantial temperatures, radiative heat transfer.

In order to reduce the probability of a mechanical failure during operation, a thermo-mechanical optimization of the proposed tritium target design will be performed. A steady state thermal analysis will be performed to better understand the modes of heat transfer within the tritium target assembly, and to provide the input for a thermal stress analysis. A static mechanical analysis will then be performed using pressure forces in combination with calculated thermal stresses. By optimizing the design and rerunning the thermal and mechanical analyses, it was possible to converge on a safe design solution.

Heat Loads

During operation, the target assembly will be subject to several sources of heat. The primary sources will be interactions where the electron beam passes through the walls of the sealed containers. 5 W of heat will be deposited when the beam passes through each of the four walls in the beams path. The beam is rastered in such a way as to spread out the effective area through which the beam passes. An area of 20 mm^2 was assumed for initial calculations. It is desirable to minimize the raster area, so this area will ideally be reduced in future designs.

The beam interacts with the sealed gasses by the same mechanism as with the walls of the containers. A small fraction of high energy electrons from the beam scatter off of atoms in the gas and release some of their energy as heat. Due to the lower atomic number of tritium and helium, compared to those of the metals in the container walls, the beam absorption and thus the volumetric heat generation will be much lower. However, due to the large gaseous volume, the beam will deposit approximately 18.3 W in the tritium and 0.23 W in both the upstream and downstream sections of helium.

In addition to the beam interactions, there will be at most 3.2×10^{-3} W of heat generated due to the radioactive decay of tritium. This contribution is minor, relative to the 38.9 W of heat generated in the target by interactions with the electron beam.

The heat generated by the previously described mechanisms has to be transported to the exterior of the vacuum chamber. In order for the design to be failsafe, the entire assembly will be passively cooled.

The bulk of the calculations carried out for the following analyses used a temperature boundary condition on the end of the rotating assembly. This assumption was made for a variety of reasons. The primary reason this boundary condition was used is because of the yet undecided manner in which the rotating target assembly attaches to the vacuum container. In current models, the assembly will rotate on a large vacuum ball bearing. The thermal conductivity of this assembly is unknown. If it turns out the bearing assembly is a poor conductor of heat, then alternative methods of removing the 38.9 W of heat will be used. The most likely method will be using a flexible copper braid as a thermal strap. Such thermal straps have been used in vacuum chamber experiments for a similar role. Even if the temperature does not turn out to be 50 °C as it was assumed, the relative temperature distribution will remain largely the same. That is to say, if the maximum temperature of the assembly was 200 °C at steady state with a 50 °C boundary condition, then with a 40 °C boundary condition the maximum temperature would be 190 °C.

Modes of Heat Flow in the System

Heat will be transferred from the target to the exterior of the vacuum chamber by each of the three fundamental modes: radiation, convection, and conduction. In the interest of reducing the complexity of the design problem, each mode of transfer was studied to determine its relative contribution to the flow of heat in the target assembly.

During operation, the sealed target assembly will have a higher temperature than walls of the vacuum chamber. Exchange of electromagnetic radiation between the two will transfer some amount of heat from the target assembly. Due to the nature of radiative heat transfer, the total power radiated by the inner target assembly will be relatively minor provided that maximum temperature in the system does not exceed 250 °C and the temperature differences do not exceed 100-150 °C.

The concentric tubes of the target assembly form an annular geometry which has been thermally well characterized for a range of Prandtl numbers in the literature [1], [2], [3]. Correlations from the literature typically report the effects of convection in terms of an effective thermal conductivity. Two such correlations were applied to the helium in the concentric annulus of the tritium targets to determine the effect of convection on heat transfer from the inner cylinder.

Several assumptions were made for the physical values that were used in the following experimental correlations. The first is that helium at 5 atm behaves as an ideal gas. It was also assumed that the average temperature drop between the inner and outer cylinder was 10 °C. This is was a rough value based on initial models. Both models reported the ratio of effective thermal conductivity to static thermal conductivity, k_{eff}/k as being proportional to the quarter power of the temperature difference. Consequently, even a 5 fold increase in the temperature difference only increases the thermal conductivity ratio by 1.5 times. Each of the models were applied in the temperature range of 300 K to 1000 K, which corresponds to the material limits of AISI 316L and AL 6061.

The first experimental correlation tested was reported by Berkengeim [1]. The correlation reported thermal conductivity ratios in the range $0.264 < k_{eff}/k < 0.539$. According to [1], this indicates that convection plays a negligible role, and that the static thermal conductivity can be used. The second correlation was reported by Raithby and Hollands [2]. Similar to [1], the correlation reported a ratio between effective and static thermal conductivity. The model was accurate for a Prandtl number range of $0.7 < Pr < 6000$. Helium at 5 atm and the temperatures tested has a Prandtl number in the range $0.65 < Pr < 0.68$. While slightly out of bounds, the model should roughly determine if convection plays any role. The correlation reported slightly higher ratios of thermal conductivities, in the range $0.321 < k_{eff}/k < 0.659$. Still, according to the model this indicates convection plays a negligible role.

Since both models reported that convection does not contribute to heat transfer, it is safe to say the convection can be ignored in future models. Therefore, heat transfer in the target cell is dominated by conduction. This is an important result, as it allows for the construction of a model with static gasses.

An assumption was made that the maximum temperature and temperature difference in the model will be well below the above stated values, so radiative heat transfer was not taken into account initially. Based on these considerations, it was concluded that conduction is the primary mode of heat transfer in and through the target assembly. The thermal conductivity of metals, such as AISI 316L or AL 6061-T6, is 100 to 1000 times greater than that of gasses at low pressures. Thus, the most of the heat will be conducted away by the metal structure of the target assembly, rather than by the contained gasses.

General Design Method

An iterative design approach was used to create a satisfactory design for the target assembly. Starting with a best guess structure based on the bare design requirements, several phases of modifications were made and tested. The best features of each generations design were carried through to the next generation, while parts that needed attention were modified.

A temperature profile for each design was calculated using ANSYS 12.0, starting with a thermal model based on the heat loads and boundary conditions previously described. If the temperatures in the model were below the melting temperature of the construction material, then a static mechanical analysis was performed. Average temperatures for the helium and tritium were calculated using the results of the thermal model. Using the assumptions that tritium and helium behave as an ideal gas, and that the canisters were filled at room temperature, the pressures of each gas were calculated for thermal equilibrium conditions. Mechanical boundary conditions were generated for a static mechanical analysis using the calculated pressures, along with symmetry constraints which came about from the modeling method. ANSYS calculated resultant stresses based on these combined loads.

The resultant stresses in the model were evaluated against the yield stress of the construction material at the highest temperature in the assembly. If the material was below the yield stress, then the design was considered feasible. For the final design, it will be necessary to determine a minimum acceptable factor of safety for the maximum calculated stresses. This factor of safety will insure that most potential material and construction flaws do not contribute to a mechanical failure of the tritium target.

1st to 2nd generation

Thermal modeling based on the initial design showed parts of the target would get too hot. As can be seen in Fig. 2, one end cap of the target is at approximately 1400 °C at steady state. This temperature is above the melting temperature of AISI 316L steel, which is around 1370 °C [4].

The hottest parts of the target were where the electron beam interacted with the walls of the gas cells. The downstream end of the target was at a lower temperature than the upstream end. This was due to the proximity of that end to an aluminum heat sink running along the bottom of the gas cells. In the 2nd generation design, the heat sink was extended to reach closer to each end of the cell and ensure more efficient heat transfer. Additional design changes were made to reduce the thermal resistance through the walls of the containers wherever possible. The walls of the initial design were thin for the purpose of ease of construction. It was possible to selectively increase the thickness of the end caps of the target cells, which granted significantly better thermal performance. These design changes were predicted to reduce stresses in the end caps. Since the highest thermal gradients in the initial target design were on the end caps, this was where the highest thermal stresses would be localized. The thickened parts of the end caps will be subject to much lower stresses, due to lowering of the temperature gradients by increasing the cross sectional area of the end caps. The last major improvement to the design of the target was the manner in which the inner tritium cell was mechanically constrained to the outer helium cell. In the initial design, these positioning rings were stepped off from the ends of the tritium cells. By moving the rings closer to the end caps, it was possible to eliminate the need for the bulk of the generated heat to pass through the thin walls of the cylindrical part of each container. Instead, heat would just travel from the end caps to the positioning rings to the aluminum heat sink. The rings were also updated with gas flow paths so that they could be used to fill the target. This removed the need for additional valves to connect to each of the tubes, and thereby decreased the complexity of construction.

The rotating target switching part was modified to simplify construction of the final device. The updated design consisted of a single cylindrical piece, to which each gas cell would be bolted via an intermediary: the aluminum heat sink. While bolted connections have higher thermal resistances than welds, using a bolted connection was seen as an acceptable tradeoff when considering welding something so close to a pressurized vessel. It would be possible to place a sheet of indium or silver in between the bolted faces to facilitate better thermal contact. This would help to mitigate the effects of the connection.

2^{nd} to 3^{rd} Generation

Thermal modeling of the 2^{nd} generation model showed that many of the design updates worked to reduce the maximum temperature in the target in steady state. Fig. 3 shows that the maximum temperature dropped from 1400 °C to 464 °C. This temperature is well below the melting temperature of AISI 316L stainless steel, so from the standpoint of the thermal model the design was acceptable.

The results of this static mechanical analysis showed that the design needs further optimization from a mechanical standpoint. As can be seen from Fig. 4, there are highly localized stresses on the ends of each cylinder. In order to determine whether the cause of these stresses was primarily due to thermal gradients or static pressure load, another mechanical analysis was performed using only the thermal loads. The results of this test show that the stresses are primarily caused by thermal gradients in the end caps. Fig. 5 shows that stresses in the end caps even increase slightly when the static pressure loads are ignored. This is because the stresses in the end caps are primarily compressive, and are relieved slightly by the pressure of the gasses.

Although it turned out to be infeasible due to the calculated von Mises stress values being higher than the yield stress value for SIA 316L, the design performed significantly better than the 1st generation design. Several possible solutions for this problem were proposed.

The compressive stresses were caused by expanding material being surrounded by material which was not expanding. One way to relieve this compression was to make the heated part more flexible. An example of this is depicted in the end cap in Fig. 6. The idea is that when the center is heated, it will be freer to expand against the thin walls than if it had to push against the solid end caps. The downside to this is that the thermal resistance of the part will be increased by the long thin step. This is a tradeoff that needs to be considered with this design.

Another method to reduce the stresses on the end cap is to reduce the temperature gradients. There are two methods by which the temperature gradients can be decreased. The first is by decreasing the amount of heat that is deposited in the first place. Decreasing the thickness of the wall will reduce the amount of heat that is deposited by beam interaction with the metal in the walls. This has the side effect of decreasing the mechanical strength of the wall.

The other method to reduce the temperature gradient in the wall is to increase the thermal conductivity of the wall material. This can typically only be done by changing the material that is used. Stainless steels have low thermal conductivities relative to other materials, such as aluminum or copper. However, the stainless steels are typically corrosion resistant, simple to weld and machine, have high melting temperatures, and maintain high yield stresses at high temperatures. AISI 316L was chosen for the 1st and 2nd generation models because of these characteristics. Switching to another material means that all of these points must be considered. The DOE Handbook: Tritium Handling and Safe Storage [5] lists several materials which have been successfully used in tritium canisters. The handbook recommends using AISI 316L because of the above properties; however others, including aluminum, are acceptable.

Aluminum 6061-T6 was chosen for the 3rd generation design despite its mechanical inferiority to AISI 316L. It was suspected that the ten fold increase in thermal conductivity would reduce the temperature gradient to such a great degree that high mechanical strength would not be needed. Another advantage of using AL 6061-T6 was that since aluminum has a lower Z number than iron, the electron beam will scatter less frequently off its atoms. Nearly transparent walls are necessary because scattering products from the walls of the target are picked up by the surrounding detectors, contributing noise to the tritium scattering signal. Minimizing the thickness of the walls minimizes the amount of scattering off the walls of the experiment, reducing this source of error. Calculations showed that an aluminum wall approximately 3 times thicker than stainless steel wall would equally transparent to the beam. From the thermal design standpoint, the increased wall thickness will decrease the thermal gradients in the walls.

3rd to 4th Generation

Thermal modeling of the 3rd generation showed that using aluminum reduced the maximum temperature in the cell from 464 °C to 164 °C. As can be seen from Fig. 7, the highest temperature is located in the center of tritium. The maximum temperature of the aluminum in the model is 112 °C. This is below the 150 °C limit over which many aluminum alloys loose heat treatment and temper properties. From the perspective of the thermal model, the design is feasible.

A static mechanical analysis of the 3rd generation was performed, using boundary conditions and loads very similar to previous models. In contrast to the 2nd generation design, the maximum equivalent stress was located on the end of the inner cylinder, as depicted in Fig. 8. This localization underlines the need for further refinement of this model. The high stress gradient at the end of the inner cylinder is caused in part by the sharp corner. In reality that part would be welded. This weld would act as a fillet. It is likely that if this fillet were included in the model, the magnitude of the stress at that location would change. This goes to show that the model does not always reflect the reality of the physical situation. Despite the likelihood that the calculated maximum stresses in this model are higher than the stresses in the actual situation, none of them went over the yield stress of AL 6061-T6 at 200 °C [6]. This means that the 3rd generation model is a usable design. By using this model as a basis of design, it will be possible to detail design a 4th generation model to maximize the capabilities of the design to certain factors of safety.

The location of the maximum stress was not close to any localized heat generation. This means that the stresses found in the target were primarily caused by pressure loads, in contrast to the 2nd generation design. This means that the rest of the geometry, not just the end caps, becomes important for the 4th generation design.

The two simplest ways to modify the overall geometry of the cell is to change the thickness of the cylinder walls and windows, and to change the pressure of the helium. The pressure of tritium was also varied in further test, in order to verify that the vessel would withstand static pressures after nearly all of the tritium radioactively decayed. The results of these varied thicknesses and pressures can be found in Table 1. Note that the calculated factors of safety are relative to the yield stress of AL 6061 [6], not the ultimate stress. Also, the maximum computed stresses occur in a region at 120 °C, while the yield stress values were obtained at 200 °C. As it is known that yield stress values decrease with the increase in temperature, actual factors of safety are higher than reported.

The results of the temperature calculations indicate that neglecting the radiative heat transfer was valid approximation. Taking radiation in account would slightly improve the calculated safety factors.

Welding considerations for the 4th generation design indicate it is likely AL 6063 will be used instead of AL 6061. While the thermal conductivity of AL 6063 is better than AL 6061, the yield and ultimate stresses are lower. The relatively high factors of safety calculated for each of the various thickness configuration shows that the design can most likely handle the material change to AL 6063.

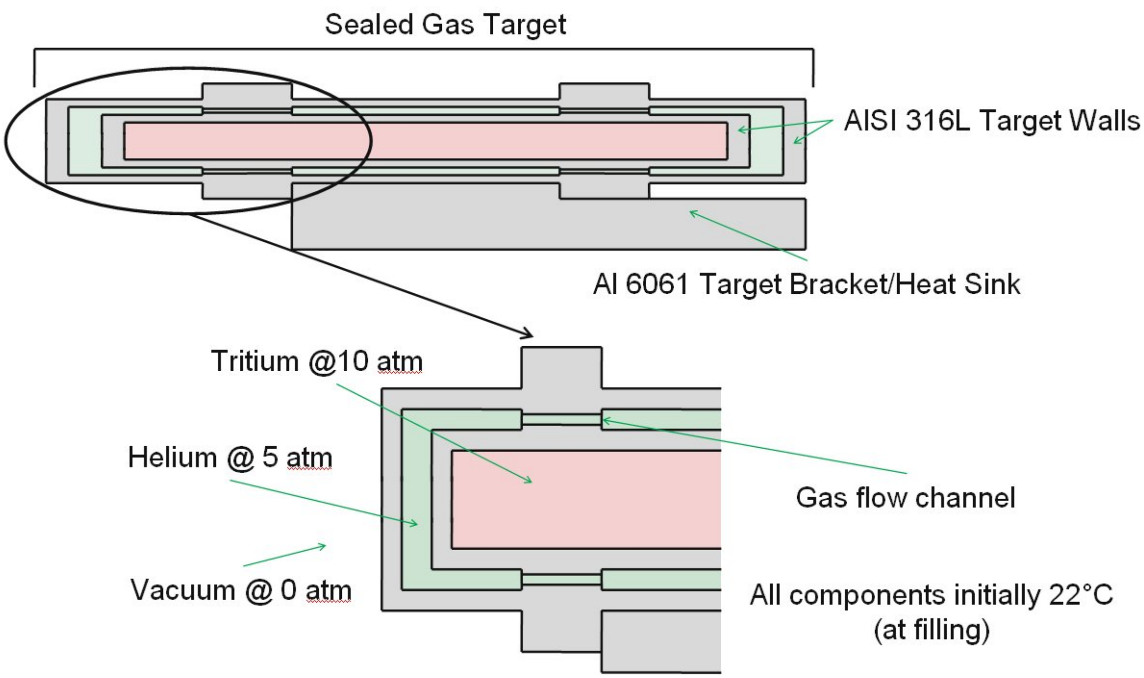

Figure 1 – Schematic design of the double wall tritium target

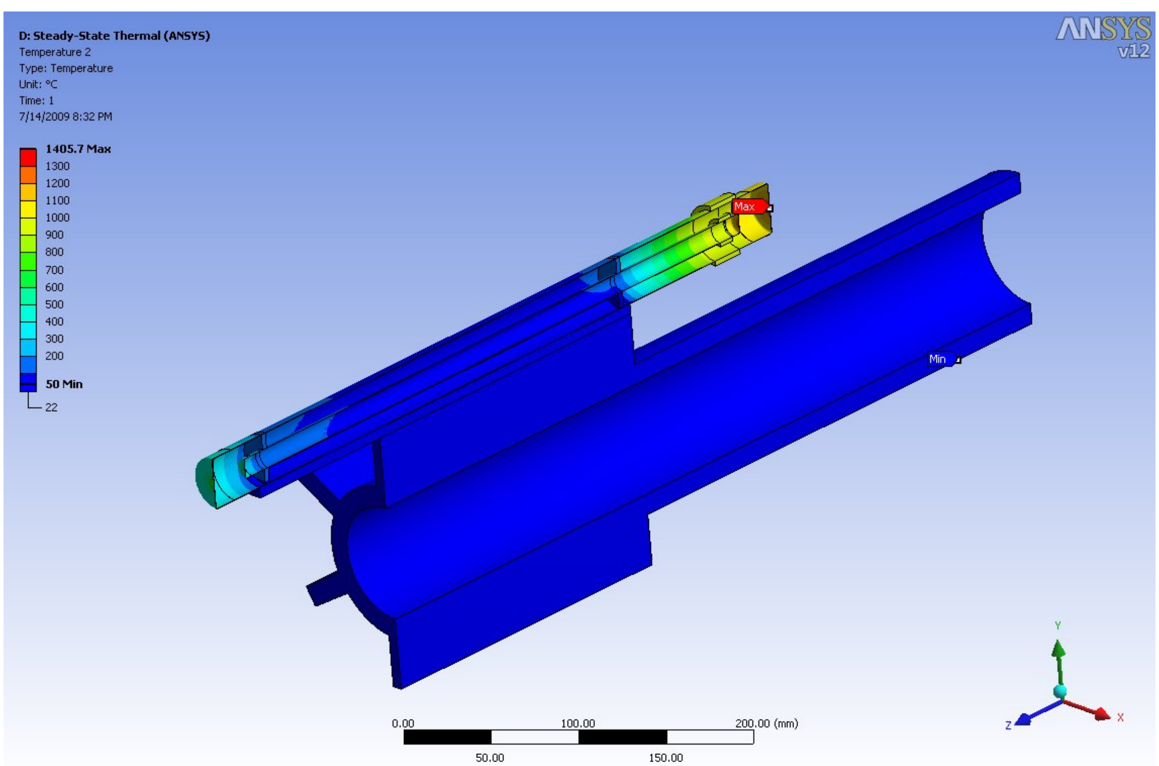

Figure 2 – Temperature distribution of the initial (1st generation) design (AISI 316L)

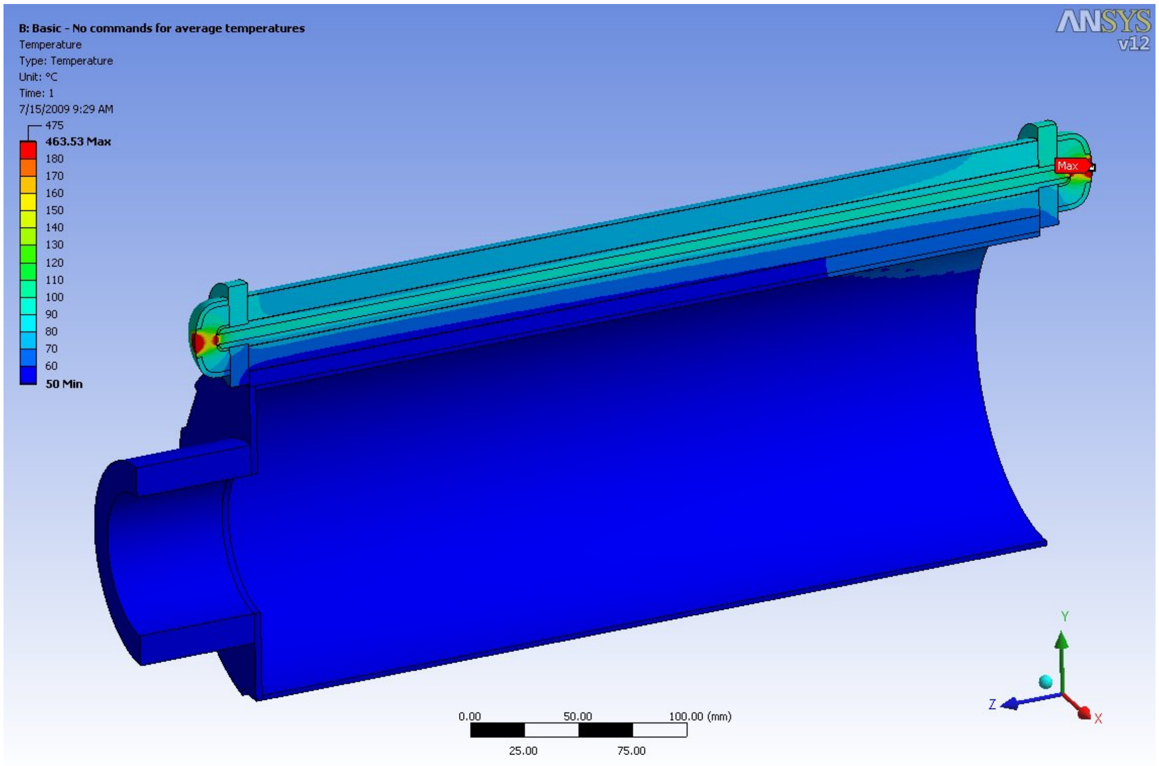

Figure 3 – Temperature distribution of the 2nd generation design (AISI 316L)

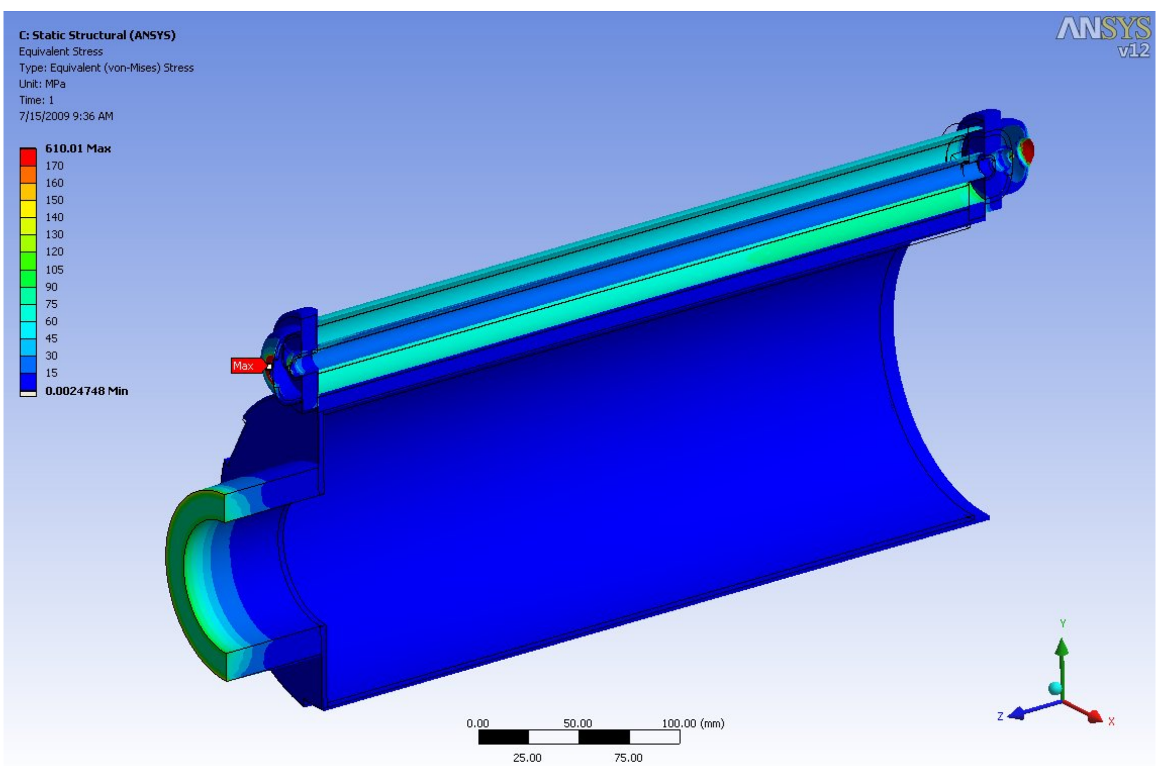

Figure 4 – Equivalent stress distribution of the 2nd generation design (AISI 316L)

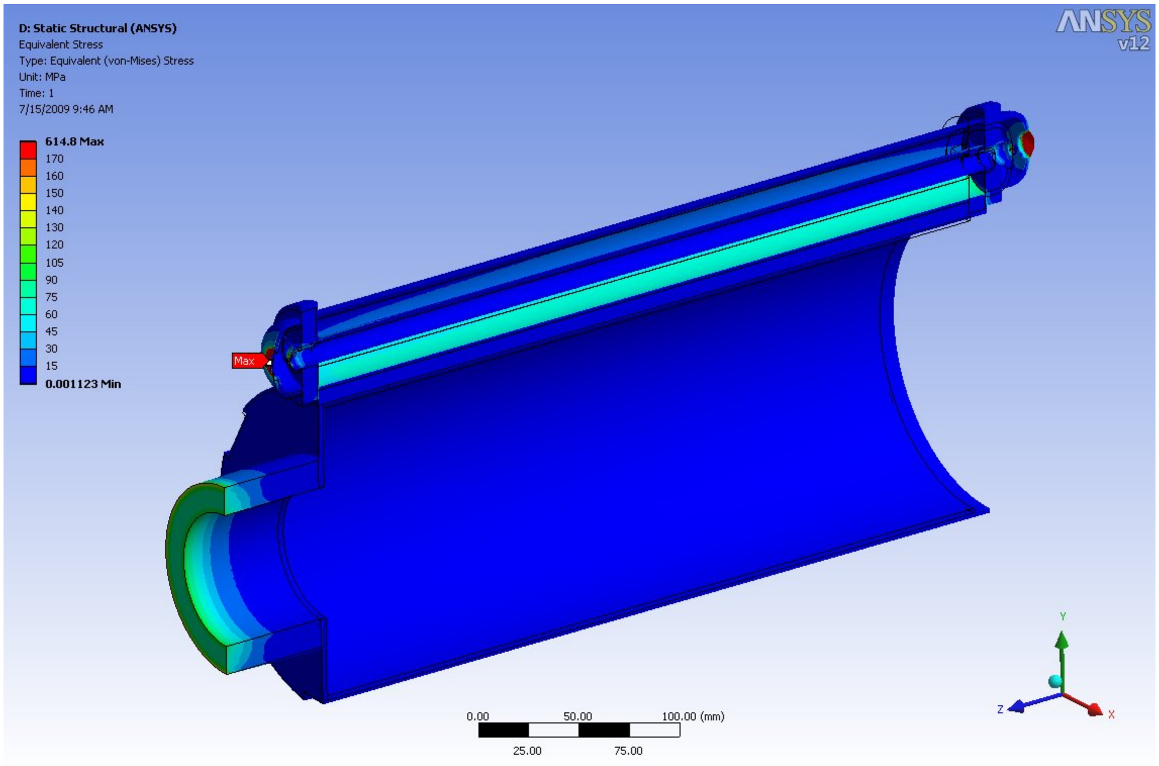

Figure 5 – Equivalent stress distribution of the 2nd generation design (AISI 316L), neglecting gas pressure. Note the slight increase in maximum stress

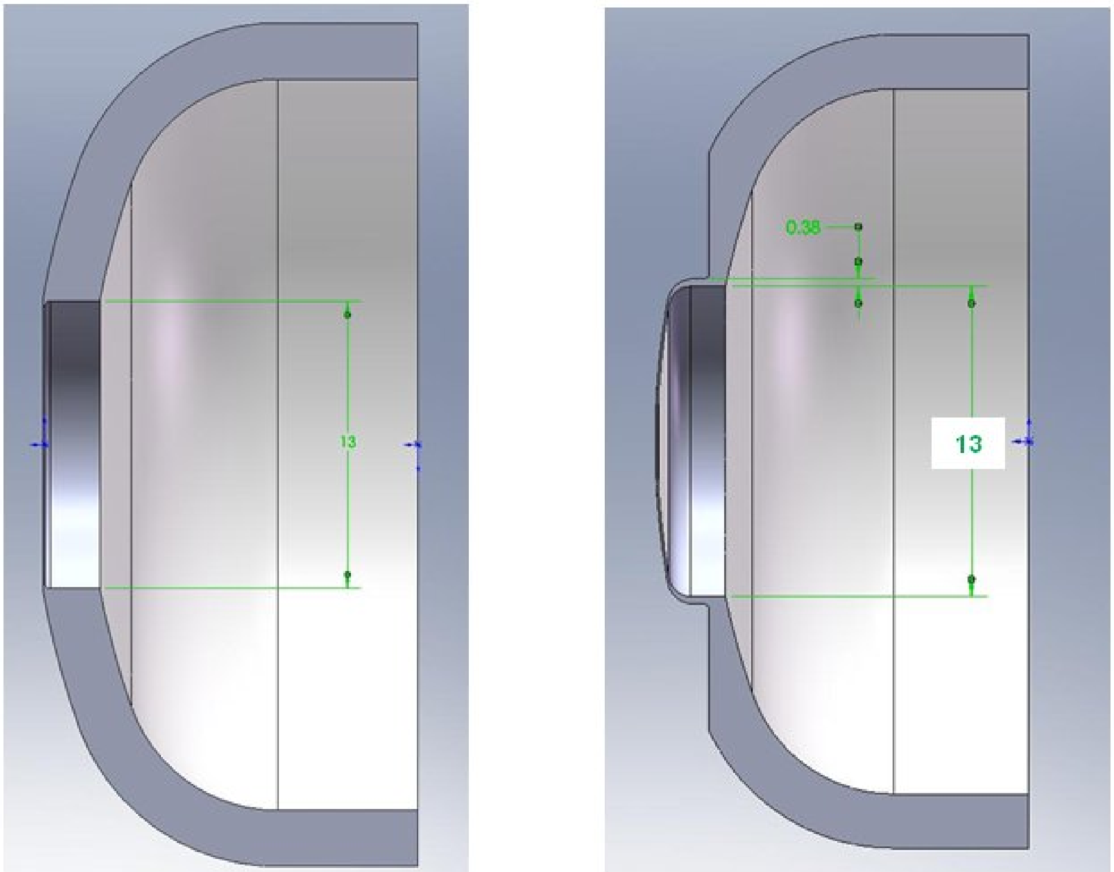

Figure 6 – (Left) Old end cap design, (Right) New stepped end cap design

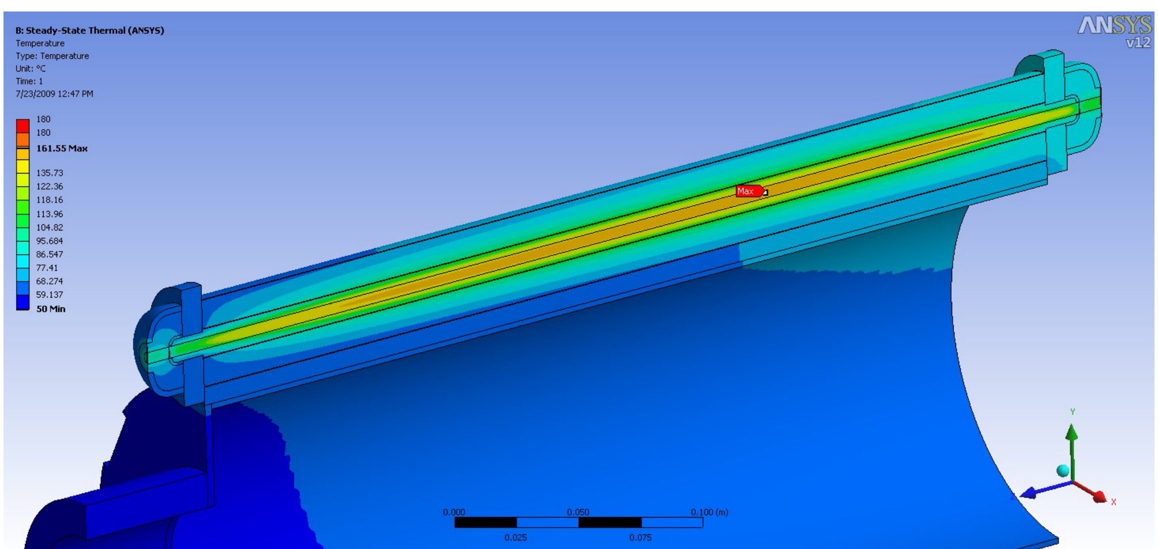

Figure 7 – Temperature distribution of the 3rd generation design (AL 6061)

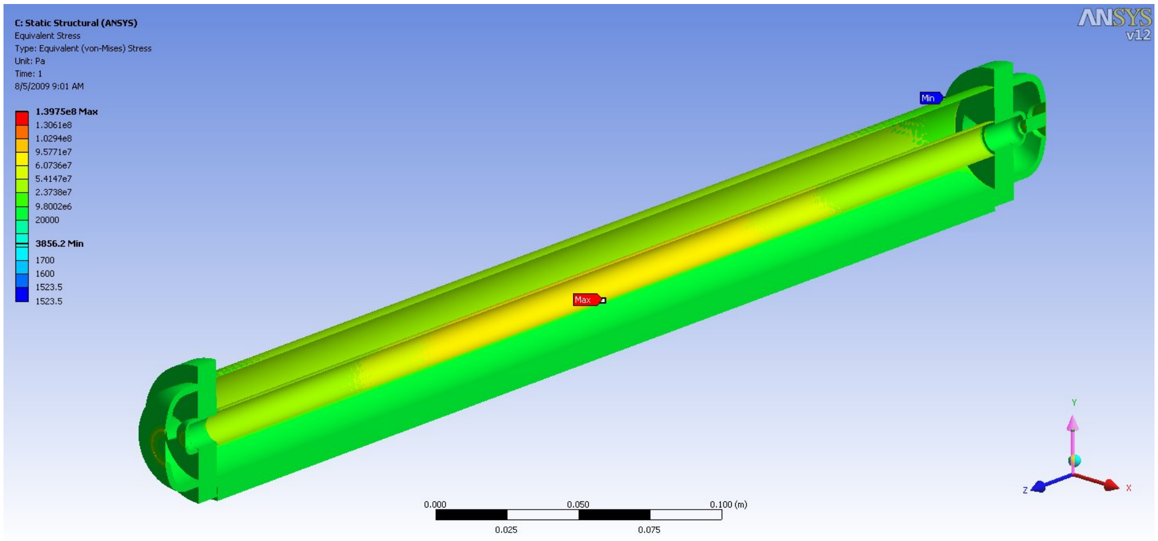

Figure 8 – Equivalent stress distribution of the 3rd generation design (AL 6061)

Factor of Safety = (Max Allowable)/(Calculated)			T_{max} (°C)	Equivalent Stress FOS (σ_{max} = 221 MPa)			
				Inner		Outer	
			AL 6061	End Caps	Cylinder	End Caps	Cylinder
Walls - 0.25 mm, Windows - 0.25 mm	Tritium - 10 atm	Helium - 2 atm	111.9	4.800	2.519	7.045	7.895
		Helium - 5 atm		5.742	3.013	4.495	4.693
	Tritium - 20 atm	Helium - 2 atm		2.235	2.011	7.159	7.300
		Helium - 5 atm		2.504	2.393	4.407	4.556
Walls - 0.30 mm, Windows - 0.30 mm	Tritium - 10 atm	Helium - 2 atm	112.5	5.814	2.736	7.131	8.331
		Helium - 5 atm		6.963	3.173	4.972	5.248
	Tritium - 20 atm	Helium - 2 atm		2.893	2.246	7.245	7.774
		Helium - 5 atm		3.264	2.604	4.883	5.118
Walls - 0.381 mm, Windows - 0.381 mm	Tritium - 10 atm	Helium - 2 atm	118.4	7.665	2.733	7.275	8.786
		Helium - 5 atm		8.413	3.182	6.299	5.715
	Tritium - 20 atm	Helium - 2 atm		4.048	2.310	7.385	8.306
		Helium - 5 atm		4.615	2.624	6.201	5.514

Table 1 – Factors of safety of the aluminum parts in target cells of various dimensions and gas pressures.

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