Modeling of Highly Stable Thermal Environment Using COMSOL

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Abstract: Precision physics experiments in space flight require highly stable thermal environments, especially if the experiments are targeted for earth orbits where eclipses will cause large temperature variation. We have been designing and modeling a thermal system with sub micro-Kelvin stability using COMSOL. Our design is comprised of multiple alternative layers of conductive materials and vacuum isolations, with supporting spacers in between. To reduce the computation time, we have used 2-dimensional model, and found that it produced more stringent design requirements than the 3-dimensional model. We combine multi-physics model such as heat transfer and structure mechanics to simulate the effects of both at the same time. We create different boundary conditions with range of temperature to assure the design capability to be in the range of 0.1 micro-Kelvin stability and uniformity. Our model runs with different types of meshing methods and numerical solvers to compare their accuracy and computing time.

Keywords: Thermal, Heat, Multilayer, Radiation, Multi-Physics, Microkelvin

1. Introduction

Precision physics experiments on length metrology requires extremely stable thermal environment. The Modular Gravitational Reference Sensor (MGRS) [1] targets measurement precision of 10^{-12} m or 1 picometer, requiring a temperature stability of 10 microkelvins. The Space Time Asymmetry Research (STAR) [2] targets measurement precision of 10^{-15} m, or 1 femtometer, requiring a temperature stability of 0.1 microkelvin. In addition, temperature homogeneity over a volume of several liters is required at the same level. In a harsh space environment, very large temperature variations may occur, when a spacecraft orbits through sun shine and earth shadowing regions.

Experimental measurement of 0.1 microkelvin temperature resolution is not yet commonly available. Proper design modeling is important is critical in this work. We carried out COSMOL modeling in 2007 [2, 3]. Our idealized, initial model predicted that a multi layer structure could reach the 0.1 microkelvin temperature stability and homogeneity requirements. However, the structure was highly simplified and did not include the supporting mechanism. We have further developed a more realistic COMSOL model with actual supporting structures in the current projects. To reduce the computational time, we have used 2-dimensional model, and verified that the 2-dimemsional model actually produced more stringent requirements than the 3-dimensional model.

2. Geometry and physical model

The thermal enclosure consists of six alternating layers of aluminum shields and vacuum spacing. The aluminum layers will spread the heat concentration to improve the temperature uniformity, and the vacuum spacing will reduce heat conduction to the same order of magnitude as radiation transfer. Radiative heat transfer can be further reduced by coating low emissivity materials on the aluminum as shown in Figure 1. Shiny gold coating can reduce the emissivity to ~0.03, approximately a factor of 10 reductions from a rough surface. Supporting spacers are used between any two layers, and heat conduction through spacers can be significant.

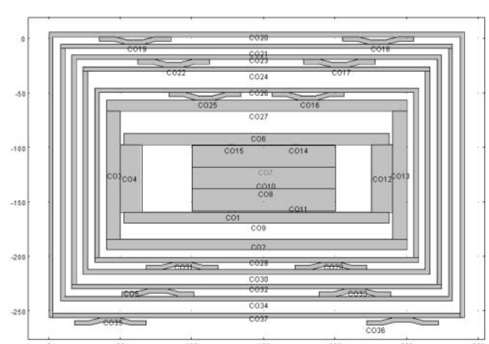


Figure 1. Geometric model for multilayer thermal enclosure

Inside the enclosure is the optical bench and the core science optical cavities made of Ultra Low Expansion glass (ULE), with a nominal thermal expansion coefficient of 10^{-9} at an optimal temperature around 15^0 C. The optical bench will have several optical components. The cavities are further sandwiched between the two optical benches. The cavities are orthogonally bored in the ULE block. Other materials, such as zerodur and fused silica, will be considered if thermal enclosure can provide better performance than microkelvin scale.

Laser light will be coupled using optical fibers. The number of electronics components will be minimized to reduce heating effects.

The goal of the modeling is to investigate the performance of the thermal enclosure with sufficient realism, and thus guide thermal design. We paid particular attention to the supporting spacers of the multilayer structure. Spacer shape, location, and material needed to be optimized via modeling process.

Meshing has important consequences for modeling accuracy. In the regions with large temperature gradient, finer meshes are used. A finer mesh example for the model is shown in Figure 2. The trade-off is increased computational time, especially when solving for 3-dimensional models. We have attempted several meshing schemes to balance accuracy and computational time.

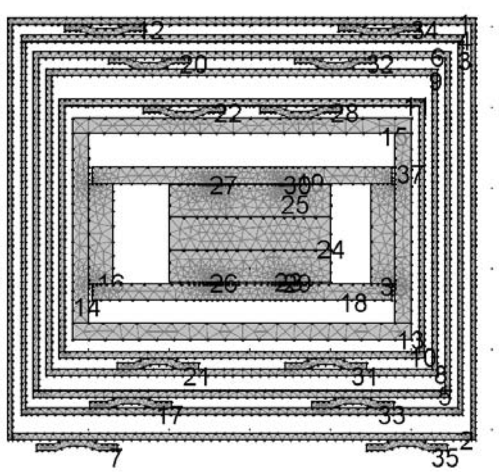


Figure 2. Geometric model with finer mesh.

3. Heat Transfer and Material Properties

The model mainly involves radiation transfer in vacuum between two adjacent layers and conduction through the spacers.

Radiation is the main mode of heat transfer in vacuum/space. For the simplest case of two parallel plates with temperature T_1 and T_2 and emissivity ε_1 and ε_2, the radiative heat transfer is the given by

$$q_{12} = \frac{A_1 \sigma \left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

where σ is the Stefan-Boltzmann constant given by $\sigma = 5.67 \cdot 10{-}8$ W/(m$^2 \cdot$T^4), A_1 is the facing area of the two plates.

The bulk heat conduction through a spacer between two plates is given by

$$Q_{12} = \frac{kA}{l}\left(T_1 - T_2\right)$$

where k is the thermal conductivity, A is the spacer contact area, and l the spacer length.

The COMSOL model has the ability to solve for more general cases, involving space distribution of heat flows, and for complex boundary conditions.

The material properties from COMSOL Heat Transfer Module [6-9] are listed in Table 1.

Thermal Properties	**Aluminum 7075**	**Ti – 6Al-5Zr-o.5Mo-o.25Si**	**ULE**
Thermal Conductivity (W/(m*K))	129	4.1	1.31
Density (kg/m3)	2810	4450	2210
Heat Capacity at Constant Pressure (J/(Kg*K))	960	552	767

Table 1. Material thermal properties used in COMSOL Model [6-9]

4. Stationary Solutions

Given a set of time invariant boundary condition, COMSOL solves for stationary temperature distribution of the thermal enclosure. Some important information, such as temperature spatial distribution, spacer thermal effects, and material property impacts, can all be viewed from stationary analysis.

Figure 3 shows the temperature distribution throughout the enclosure for an external temperature difference ~50K between the top and bottom. To reduce numerical noise at minute temperature scale, this temperature difference is taken about ~5000 times larger than 1 mK, a temperature difference achievable in spacecraft design. The resulting temperature distribution can be scaled back to allow actual temperature reading under normal boundary conditions. As shown in Figure 3, the most of temperature gradient is reduced at first 2 layers. The most of temperature inhomogeneity is reduced at the first 3 layers. After 3 layers of isolation, the temperature distribution becomes uniformly stable.

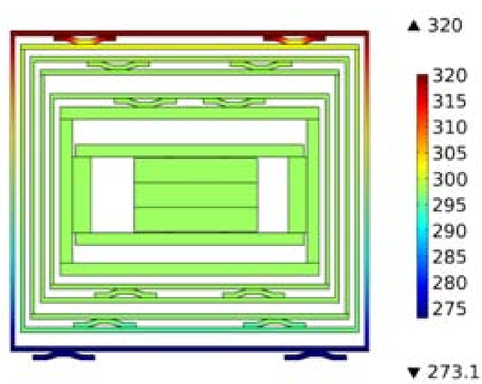


Figure 3. Solution of temperature distribution using COMSOL Heat Transfer Module.

Figure 4 shows the temperature distribution along the X axis after the first layer of shielding. There is significant reduction in temperature gradient after the first layer already. However, temperature humps are observed near the spacer contacting points. In our model, we have in general observed that the spacer design, including shape, contact areas, location, material thermal properties all have impacts on the temperature distribution. Therefore all these aspects must be properly addressed for the enclosure to achieve the targeted performance. We have varied many parameters in modeling to achieve both thermal isolation performance and mechanical strength.

Figure 5 shows the temperature distribution along X-axis for the surfaces innermost of the enclosure near the instrument cavity block. The temperature distribution in the central core instrument is flat within a few microkelvin.

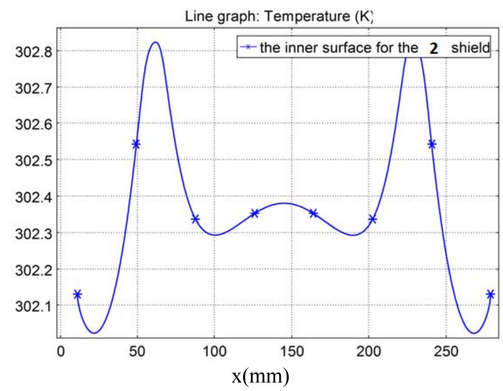


Figure 4. Temperature distribution after the second layer of shield

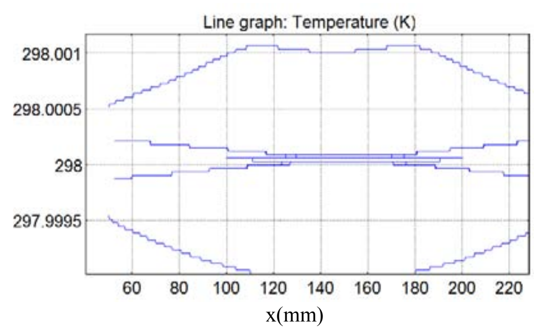


Figure 5. Temperature distribution after sixth layer of shield

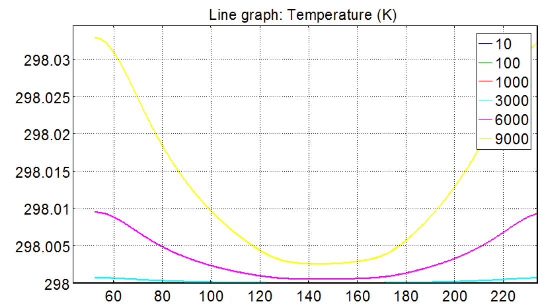
5. Dynamic Temperature Distribution

There are time-dependent thermal disturbances in earth orbit, such as earth shadowing of the sun light, and spacecraft attitude change and the resulted sunlight absorption change. The thermal model will necessarily solve for temperature evolution of the payload environment. To fully characterize the spatial-temporal behavior of the thermal enclosure and the payload instruments, We used the COMSOL heat transfer module to solve entire spatial temperature distribution at each time value for extended time horizon.

Figure 6 shows the temperature distribution change at inner shield when a 50K step temperature rise function is applied to the top and bottom of the thermal enclosure at $t = 0$. For an orbital period of 6,000 seconds, with a shadow period 2,000 seconds, there is a temperature rise of ~2 mK at the enclosure edge, and ~50 μK at the center. As such, an active temperature control loop may be required, though the control effort is substantially reduced, compared with the case without using the thermal enclosure. Similarly, Figure 7 shows the temperature distribution change at inner cavity over time.

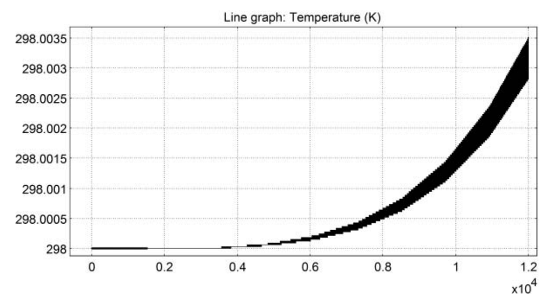
To be more quantitative, Table 2 lists the temperature rise at different layers at different times. For the initial 100s, the typical spacecraft spin period, the temperature rise is around $\sim 10^{-12}$ Kelvin, well within the requirement. In more detail, ΔT for the cavity at 2000 sec. is 4.7943200343070E-07 K, or 0.47 microkelvin. For 6000s, the typical spacecraft orbital period, the temperature rise may become significant, and thus may call for further active thermal control.

In our previous work [2, 3], a simple feedback loop was added. A thin film heater between the 2nd and 3rd layers of the thermal enclosure was used to compensate the temperature change at 10-100 microkelvin level. However, the physical model did not include the spacers and their thermal conducting properties. In the renewed effort, we will develop a more complete model, to develop the control law, the actuator, and the sensor configurations.



x(mm)

Figure 6. Temperature along the X-axis for the inner Aluminum shield for different times



Time (sec.)

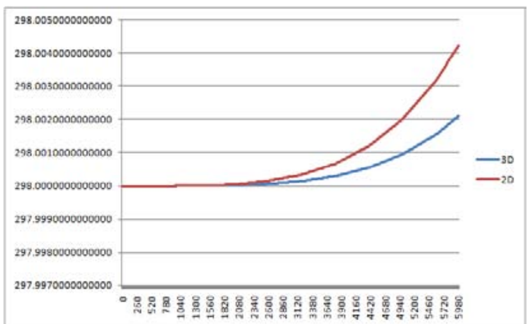
Figure 7. Temperature along the X-axis for the inner cavity over time.

Time	10	100	1000	6000
ΔT1	1.51E+00	4.93E-01	3.77E-01	3.78E-01
ΔT2	5.53E-02	1.58E+00	1.96E+00	1.41E+00
ΔT3	3.45E-06	9.25E-05	1.57E-02	3.42E-01
ΔT4	2.79E-07	2.87E-05	3.88E-03	1.21E-01
ΔT5	1.47E-07	8.84E-08	5.53E-05	1.07E-02
ΔT6	1.37E-08	1.38E-08	1.34E-05	6.22E-03
ΔT7	2.34E-08	4.85E-08	4.24E-06	2.99E-03
ΔT cavity	1.02E-12	1.02E-12	2.77E-08	7.54E-05

Table 2. ΔT for The Aluminum shields, ULE layer and Cavity.

6. 2-D versus 3-D Model

We have attempted 3-dimensional (3-D) model, but the computational time was quite substantial. However, we have realized that the 2D model can actually produce more stringent thermal design criterion than 3D model. Physically, the 2D model can be viewed as a 3D model with infinite third spatial dimension, thus does not have the fringe effects that 3D model accounts for. The temperature variations in 2D model can be more pronounced compared with that in 3D model. Figure 8 shows the actual computational comparison. Indeed, the 2D model predicts faster temperature rise than 3D model does, and thus requires more stringent thermal design.



Time (sec.)

Figure 8. Temperature for the node from the inner cavity over time for both 2D and 3D.

7. Conclusions

We have analyzed the design of a multilayer thermal isolation enclosure using COMSOL and Heat Transfer Module. We show that the microkelvin temperature stability is possible for six layers of alternating conducting material and vacuum.

The temperature fluctuation over spacecraft spin period is reduced by a factor of 10^{-12}. Similarly, the temperature gradient is reduced at similar degree, within the core cavity structure. At orbital period, there is a need for active thermal control, with the benefit of reduced control effort, thanks to the multilayer thermal isolation structure.

In the future, we will try to do more detailed modeling involving a thermal control loop, more detailed 3 D modeling, and further optimization of material, and include more accurate orbital thermal effects, and thus perfect the thermal design.

8. References

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