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OPTO-MECHANICAL DESIGN AND ANALYSIS OF MOUNTED OFF-AXIS MIRRORS FOR SPACE APPLICATIONS

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Abstract

Off-Axis mirrors are used in space borne telescopes to minimize the aberrations for improved optical performance. This telescope consists of concave hyperbolic primary and convex hyperbolic secondary mirrors which help in capturing the light from far away sources and obtain the images on to the detector. There is a need to design and analyze the opto-mechanical system to mount the primary mirror of diameter 146 mm and secondary mirror of diameter 49 mm used in this project. The opto-mechanical system needs to withstand the environmental effects during launching of the rocket as well as during space operating conditions. Mainly the system encounters the following types of environmental conditions: inertia loads, vibrations and temperature excursions. The system needs to be designed and analyzed to withstand the above conditions for failure as well as for distortion free wave front error on the optical surfaces at all operating conditions. The objectives of the project are as follows: 1) Prevention of mirror from failure for any operating conditions i.e. during launching of satellite in a rocket and satellite operation in space and 2) Optimization of geometrical parameters of the opto-mechanical components for minimum weight, which helps in improving agility and life of the satellite. In order to achieve the above objectives, two different kinds of mounting configuration have been considered, namely leaf spring ring mount and bipod ring mount. The opto-mechanical system is modeled using Solid Works (CAD Software). Two types of finite element analysis i.e. 1) modal analysis under clamped/free-free condition and 2) linear static analysis under thermal and inertia loads have been carried out using MSC Patran(FEA pre-processor & post-processor software) and MSC Nastran(FEA processor software) to study the stresses induced due to different environmental conditions. Several iterations have been carried out to arrive at the optimum design. Analytical calculations are done and compared with the FEA results in order to ensure validity of the results.

***Keywords**:* Off-Axis Mirrors, Opto-Mechanical system, Mounting Configurations, Optimization for Weight, Finite Element Analysis

1. Introduction

Opto-mechanical design of optical instruments is a tightly integrated process involving many technical disciplines. It begins with a statement of need for a particular hardware item and a definition of goals and firm requirements for the system's configuration, physical characteristics and performance in a given environment where it is intended to function. The design effort proceeds through a logical sequence of major steps and concludes only when the instrument is awarded a pedigree establishing its ability to meet its specifications and to be produced in the required quantity.

The mirrors used in this project have applications in U.V telescopes. Satellite based U.V telescopes are developed for various science and astronomical studies. One of them is the Solar U.V telescope, which is used to view the sun disc in the U.V region and to study the solar dynamics. The Solar U.V telescope consists of an opto-mechanical system to house the primary and secondary mirror. Optical systems are widely used in scientific, defense, and commercial applications. By their nature, optical systems often calls for stringent alignment and geometrical requirements that must be met despite the presence of static, dynamic, and thermal loads. These requirements often couple the optical, structural, and thermal domains and demand a coupled approach to optimize the system's performance.

As an example, an optical instrument inside a spacecraft must meet a number of requirements:

- Be robust enough to withstand the static and dynamic loading encountered during launch and ground handling.
- Function after exposure to extreme temperatures and vacuum conditions.
- Meet optical requirements in the presence of any dynamic, static, or thermal operating loads by minimizing distortions and relative displacements
- Be designed such that its requirements for prelaunch adjustment, calibration, and testing can be met in the context of the larger system
- Be lightweight enough to meet mass requirements

To achieve these objectives, an instrument's design process undergoes through various phases that involve both analysis and testing. Once the results are found to be satisfactory, a prototype will be made and experimental testing will be carried out to assess the performance of the system under simulated conditions. Succeeding which, the actual model will be made and the telescope will be ready to be integrated with the spacecraft for the final testing in integrated mode and then for the launching.

This paper presents brief details of the opto-mechanical design concepts which could be employed for mounting the off-axis mirrors for space applications .Some of the finite element analysis exercises and their results are also presented along with the analytical calculations.

2. Design Constraints

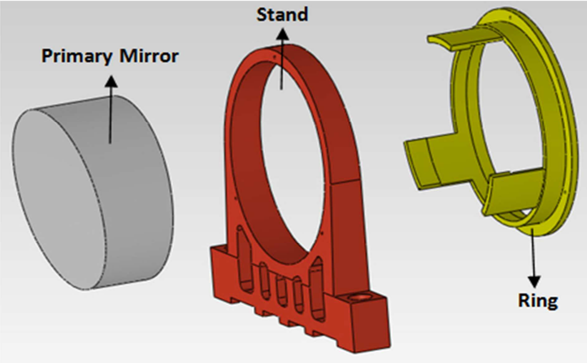
- Since the diameter of the primary mirror is 146 mm, a gap of 57 mm from the bottom part of the mirror to the base of the mount is maintained based on the preliminary design. The base to mirror centre distance amounts to 130mm. The base to mirror centre distance (130 mm) must be maintained in the secondary mirror mount so that the light rays falls appropriately on the reflective surfaces to obtain the images.

- The mass of the mirror mount is dependent upon the total load carrying capacity and the number of payloads being carried by the rocket. Along with this telescope, many payloads are being carried by the rocket and the weight of this telescope is governed by the weight of the other payloads. It has been decide that the total mass of the telescope (Primary mirror mount + Secondary mirror mount) should be less than 6.5 Kg.

3. Opto-Mechanical Design

3.1. Primary Mirror Opto-Mechanical Design

A. Leaf Spring Ring Mount: The components of primary mirror opto-mechanical system shown in figure 1. It can be employed as a design concept to mount the primary mirror having a diameter of 146 mm. Leaf spring ring mounting configuration is considered as a first option.



Component	Mass (grams)
Mirror	2354.89
Ring	623.08
Stand	1695.43
	Total mass = 4673.39

Figure 1: Exploded view of the primary mirror opto-mechanical system

The major contribution of the overall mass of the assembly is the mass of the stand and there is scope for reducing the mass of the mounting structure. However, the mass of the mirror is fixed and it cannot be altered once the mirror design is finalized. The opto-mechanical system must be optimized for weight. It is always desirable that the mass shall be as minimum as possible without sacrificing the strength and stiffness. The initial mass before optimization is found to be 5.345 kgs. The final mass after optimization is found to be 4.673 kgs. We find that a

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percentage reduction of about 12.6 % is possible. To obtain the final design, several iterations are carried out. Figure 2 and 3 shows the preliminary and final design of the primary mirror opto-mechanical system respectively.

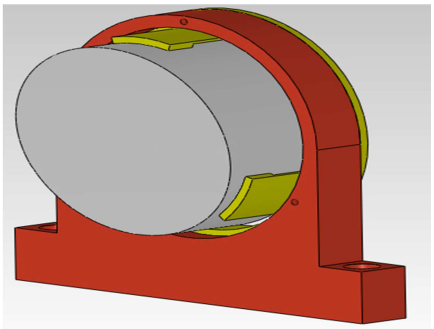

Figure 2: Preliminary design of the primary mirror opto-mechanical system

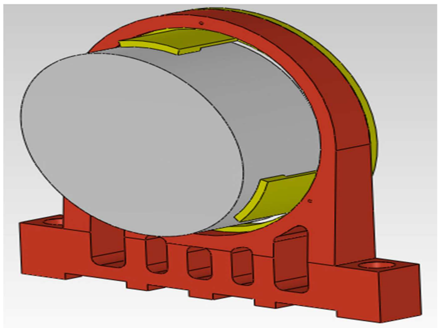

Figure 3: Final design of the primary mirror opto-mechanical system

B. Bipod –Mirror Mount: The leaf spring ring mount is replaced with a bipod type ring mount having the same contact area with the mirror. Bipod type ring mounting configuration is considered as the second option. The performance of both the type of mounts is compared. Figure 4 shows the bipod ring for the primary mirror.

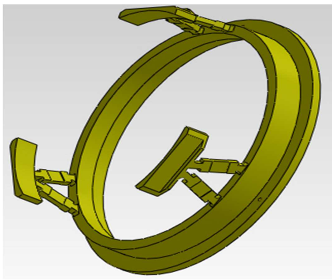

Figure 4: Bipod Ring for primary mirror

The mass of the stand is slightly modified to accommodate the bipod ring. Figure 5 shows the bipod ring mount assembly for primary mirror.

Component	Mass (grams)
Mirror	2354.89
Ring	665.70
Stand	1668.07
	Total mass = 4688.66

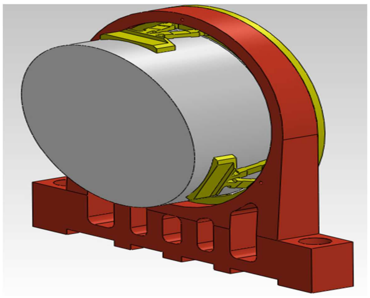

Figure 5: Bipod ring mount assembly for primary mirror

3.2. Secondary Mirror Opto-Mechanical Design

The components of secondary mirror opto-mechanical system, shown in figure 6, could be employed as a design concept to mount the secondary mirror having a diameter of 49 mm. Leaf spring ring mounting configuration is considered. The opto-mechanical system is optimized for weight by designing a mount having mass as minimum as possible. Figure 7 shows the secondary mirror opto-mechanical assembly. For secondary mirror, analysis is carried out only for leaf spring ring mount since the weight of secondary mirror is less compared to primary mirror.

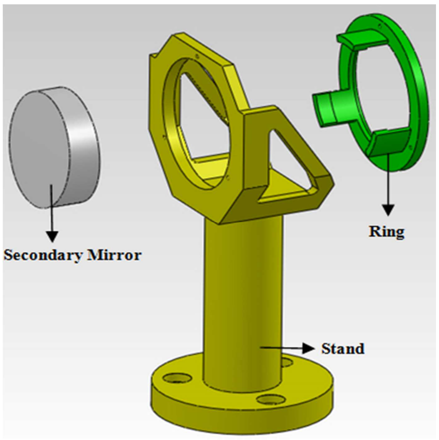

Figure 6: Exploded view of secondary mirror opto-mechanical

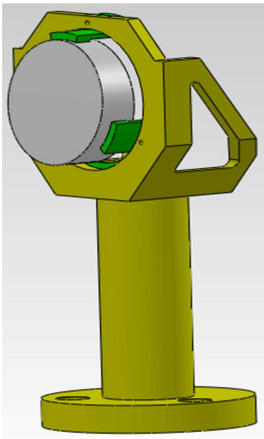

Figure 7: Secondary mirror opto-mechanical assembly

Component	Mass (grams)
Mirror	65.08
Ring	53.59
Stand	382.82
	Total mass = 501.49

4. Finite Element Analysis of the Opto-Mechanical System

4.1. FEA Procedure Employed

The CAD model is imported into MSC Patran (FEA Pre-Processor) where the components are meshed using CQUAD 4 elements by extracting the surfaces. The CQUAD 4 elements are then transformed into CHEXA 8 elements using the sweep/transform- extrude, arc, loft, etc tools provided. Since accuracy is of the highest concern, hexahedral mesh is the most preferable one. Rigid body elements can be used to connect different elements if required. Material properties, loads /boundary conditions are applied and the bulk data file (bdf file) is generated. The bulk data file is imported into MSC Nastran (Processor) and MSC Access database (xdb file) is generated. The MSC Access database (xdb file) is imported back to MSc patran for post-processing purposes. The following material properties of the components are used in the finite element analysis and is given in table 1

Table 1: Material properties of components

Component	Mirror	Ring	Stand	Glue	Screws
Material	Zerodur	Invar	Titanium	EC2216 B/A	S.S
Young's Modulus(E) (Gpa)	90.3	148	108	0.689	193
Poisson's Ratio(υ)	0.243	0.29	0.31	0.43	0.3
Density ρ (gm/cm^3)	2.53	8	4.5	1.25	8
CTE (x 10^{-6}/°C)	0.05	1.3	8.6	102	16
Yield Strength (Mpa)	5-10	300	800	17.2 at 24°C	900

4.2. FE Model Assumptions

Following assumptions are made while carrying out the finite element analysis.

1. Mirror is assumed to be linearly elastic even though it is brittle, because any brittle material posses elastic property when the deformations are in nano scale.

2. Glue layer is not modeled in the FE Models, as the glue layers are very thin, very compliant and low modulus (E < 1000 psi) rubber like and nearly incompressible ($\nu > 0.49$). Each of the above features causes some difficulty in modeling the adhesive layer. The sudden change in element size required to describe very thin layers can cause geometrical modeling problems. Due to computational limitations, the optics and mount cannot be modeled with such a fine resolution.

3. Screws are replaced with rigid body elements.

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4.3. Acceptance Criteria

The following are the stress levels within which the individual components of the assembly can be subjected to while applying all types of loading conditions.

- Stress in mirror < 5 Mpa
- Stress in ring < 300 Mpa
- Stress in stand < 800 Mpa
- Stress in glue < 5 Mpa
- Natural Frequency of assembly > 200 Hz

Environmental noise due to the launching of the rocket is considered while calculating the natural frequency. The frequency range of the noise is between 10-100 Hz. Hence, the component must be designed in such a way that its natural frequency is more than 100 Hz. The opto-mechanical mount is designed to obtain a natural frequency of above 200 Hz since the analysis does not consider the material non-homogeneity, voids, cracks, etc. The validity of the results is checked by testing the components after its assembly. Since the experimental results will be closer to the FEA results, a margin of more than 100 Hz is considered as the design target. (i.e.> 200 Hz)

4.4. Analysis of Primary Mirror Mounts

4.4.1. Leaf Spring Ring Mount

4.4.1.1. Modal Analysis

Boundary condition: The base is fixed with 6 degrees of freedom. Figure 8 shows the finite element model of primary mirror with leaf spring ring mount. Figure 9 shows the Modal analysis result. Natural frequency obtained is 242.02 Hz

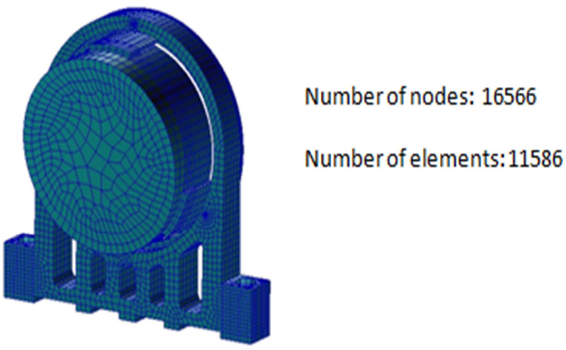

Figure 8: Finite element model of primary mirror with leaf spring ring mount

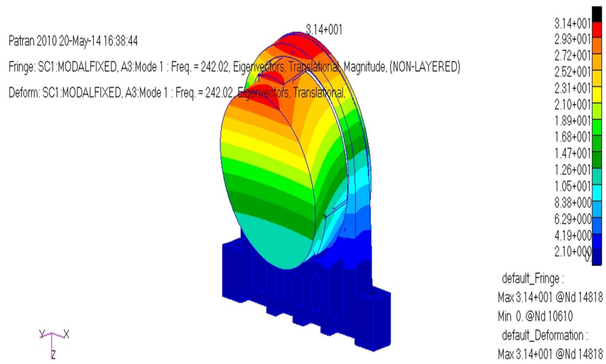

Figure 9: Modal analysis result of primary mirror with leaf spring ring mount

Several iterations are carried out to optimize the opto-mechanical system for weight. The variation of natural frequency with reduction in mass is presented in the table 2.

Table 2: Variation of natural frequency with reduction in mass

Iteration	Mass (Grams)	Natural Frequency (Hz)	Generalized Stiffness Values
1	5345.22	295.84	3.45E+06
2	4978.10	274.48	2.97E+06
3	4837.87	271.92	2.92E+06
4	4723.18	254.47	2.56E+06
5	4673.39	242.02	2.31E+06

From table 2, we can conclude that the results comply with the fact that natural frequency is directly proportional to the stiffness of the opto-mechanical system. Further reduction in mass of the stand will lead to reduction of natural frequency. After five iterations, the design is considered optimum keeping in view that the natural frequency target to be above 200Hz.

4.4.1.2. Linear Static Analysis

Boundary condition for quasi-static loads: The random vibration/dynamic load that arises during the launch is converted to an equivalent Quasi-Static load using Miles' equation [1]. It gives an approximate method to find whether the component would withstand the dynamic loads. This eliminates in carrying out the dynamic analysis. However, for accurate prediction of stress distribution, dynamic analysis may be conducted.

From Miles' equation we have,

$$G_{RMS} = \sqrt{\frac{\pi \, f_n \, Q \, [ASD_{input}]}{2}} \quad (1)$$

Where,

G_{RMS} = Root Mean Square Acceleration in G's

f_n= Natural frequency = 242 Hz

$Q = \frac{1}{2\zeta}$= Transmissibility (or amplification factor) at f_n

where ζ is the critical damping ratio.

A critical damping ratio of 0.05 is considered since the transmissibility / amplification is high at this ratio.

$[ASD_{input}]$ = Input Acceleration Spectral Density at f_n in units of $\frac{g2}{Hz}$ = 0.1

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$$G_{RMS} = \sqrt{\frac{\pi * 242 * 0.1}{2 * 2 * 0.05}} = 19.5$$

The mirror should be designed to withstand a minimum random vibration level of 3 x 19.5 = 58.49 times gravity. The primary mirror mount is designed to withstand an inertia load of 70 g.

Boundary condition for thermal loads: Taking 20°C as the ambient temperature. The system is subjected to a temperature variation of +30°C and -30°C. (ΔT=30°C). Figure 10 shows some of the FEA results of the primary mirror (with leaf spring ring mount) opto-mechanical components. Figure 10 (a) shows stress in the mirror due to 70g inertia load in X direction. Figure 10 (b) shows stress in the ring due to 70g inertia load in X direction. Figure 10 (c) shows stress in the stand due to 70g inertia load in X direction. Figure 10 (d) shows stress in the mirror due to plus 30°C thermal load. The stresses in mirror, leaf spring ring mount and stand with 70g inertia load imposed in X, Y and Z directions and due to variation in temperature are presented in section 4.4.1.3, summary of results.

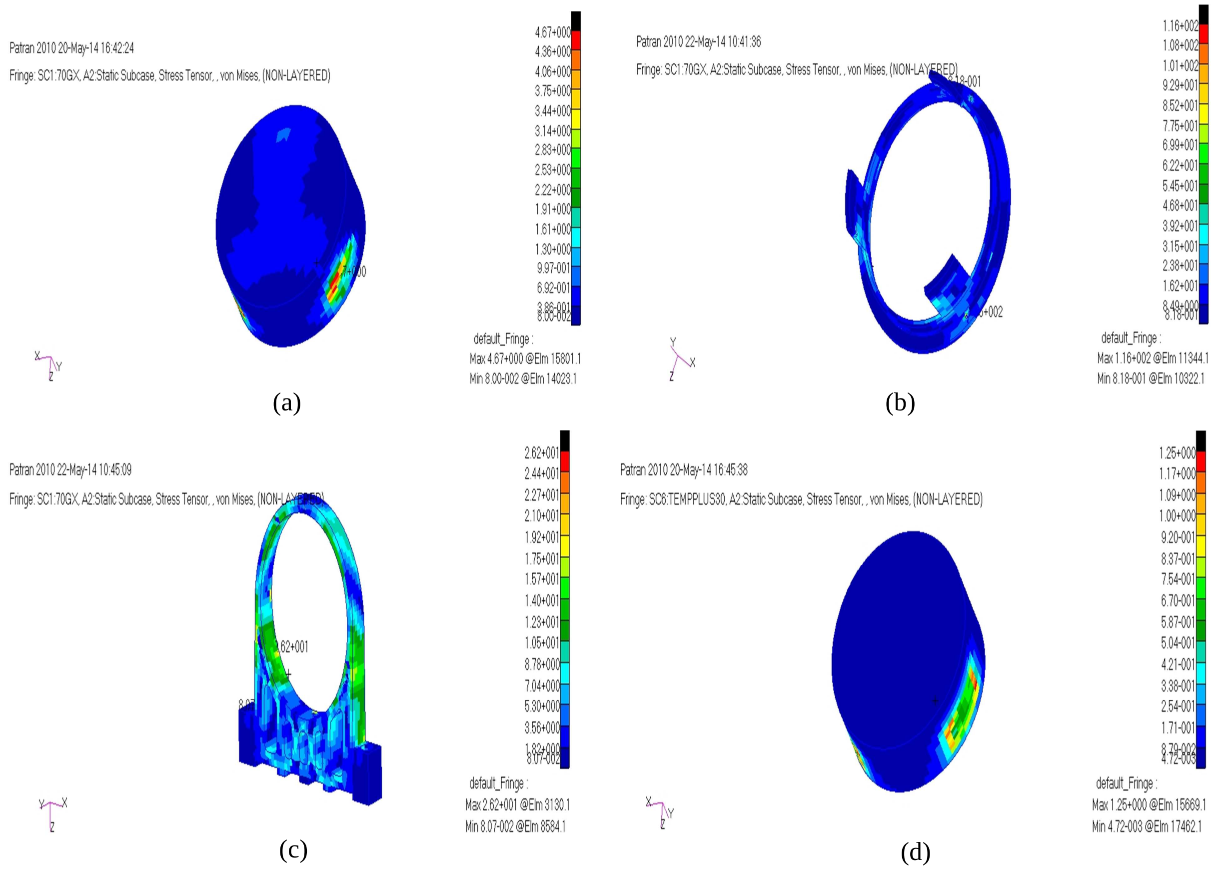

Figure 10: FEA results of the primary mirror (with leaf spring ring mount) opto-mechcanical components

4.4.1.3. Summary of results

Table 3 and 4 shows the summary of results for the components of primary mirror with leaf spring ring mount subjected to quasi-static and thermal loads respectively. The stresses in the individual components are given.

Table 3: Quasi-Static Analysis results for components of primary mirror with leaf spring ring mount

Sl. No.	Component	Von Mises Stress(Mpa) in component for 70g load			Allowable stress(Mpa)
		X	Y	Z	
1	Mirror	4.67	3.89	4.82	5
2	Ring	116	98.5	120	300
3	Stand	26.2	28	72.6	800

Table 4: Thermal stress analysis for components of primary mirror with leaf spring ring mount

Sl. No.	Component	Von Mises Stress(Mpa) in component due to thermal loads		Allowable stress(Mpa)
		+30°C	-30°C	
1	Mirror	1.3	1.3	5
2	Ring	33.2	33.2	300
3	Stand	84.9	84.9	800

4.4.2. Bipod Ring Mount

4.4.2.1. Modal Analysis

Boundary condition: The base is fixed with 6 degrees of freedom. Figure 11 shows the finite element model of primary mirror with bipod ring mount. Figure 12 shows the Modal analysis result. Natural Frequency obtained is 230.64 Hz

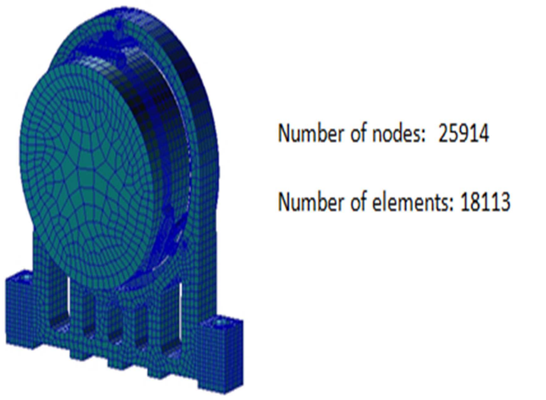

Figure 11: Finite element model of primary mirror with bipod ring mount

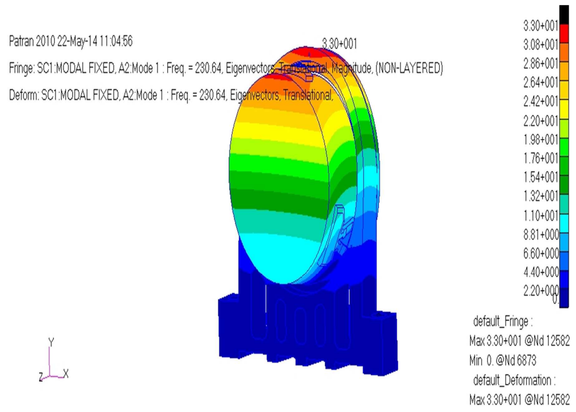

Figure 12: Modal analysis result of primary mirror with bipod ring mount

4.4.2.2. Linear Static Analysis

Boundary condition for quasi-static loads: Applying Miles' equation (equation 1) for the primary mirror with bipod ring mount configuration, we get

$$G_{RMS} = \sqrt{\frac{\pi * 230.64 * 0.1}{2 * 2 * 0.05}} = 19.03$$

The mirror should be designed to withstand a minimum random vibration level of 3 x 19.03= 57.10 times gravity.

The Primary mirror mount is designed to withstand an inertial load of 70 g.

Boundary condition for thermal loads: Taking 20°C as the ambient temp. The system is subjected to a temperature variation of +30°C and -30°C. (ΔT=30°C). Figure 13 shows some of the FEA results of the primary mirror opto-mechanical components.

Figure 13 (a) shows stress in the mirror due to 70g inertia load in Z direction. Figure 13 (b) shows stress in the bipod ring due to 70g inertia load in Z direction. Figure 13 (c) shows stress in the mirror due to minus 30°C thermal load. Figure 13 (d) shows stress in the stand due to plus 30°C thermal load. The stresses in mirror, bipod ring mount and stand with 70g inertia load imposed in X, Y and Z directions and due to variation in temperature are presented in the section 4.4.2.3, summary of results.

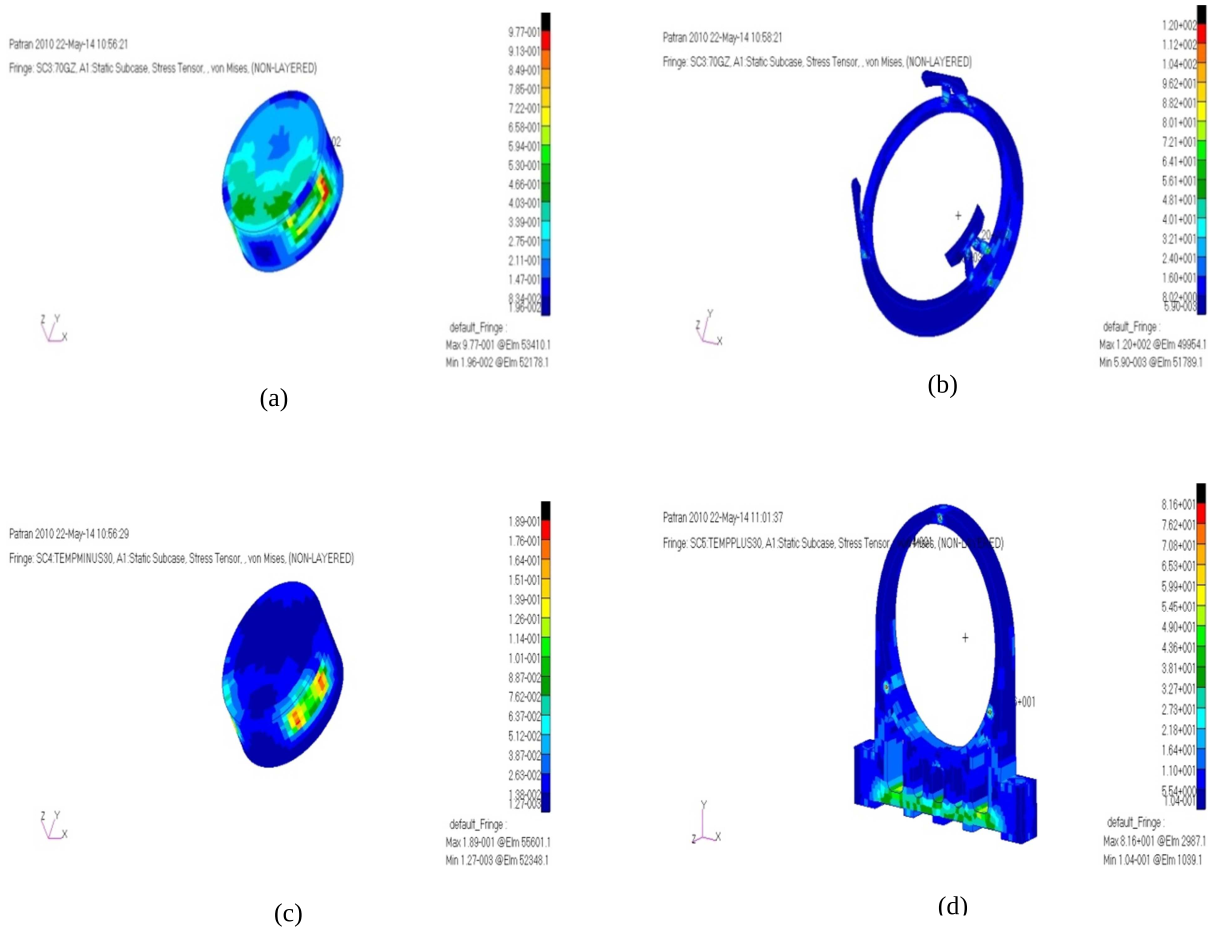

Figure 13: FEA results of the primary mirror (with bipod ring mount) opto-mechcanical components

4.4.2.3. Summary of results

Table 5 and 6 shows the summary of results for components of primary mirror with bipod ring mount subjected to quasi-static and thermal loads respectively. The stresses for all the individual components are given.

Table 5: Quasi-Static Analysis results for components of primary mirror with bipod ring mount

Sl. No.	Component	Von Mises Stress(Mpa) in component for 70g load			Allowable stress(Mpa)
		X	Y	Z	
1	Mirror	1.60	1.51	0.98	5
2	Ring	272	262	120	300

3	Stand	20.7	29.5	73.2	800

Table 6: Thermal stress analysis for components of primary mirror with bipod ring mount

Sl. No.	Component	Von Mises Stress(Mpa) in component due to thermal loads		Allowable stress(Mpa)
		+30°C	-30°C	
1	Mirror	0.2	0.2	5
2	Ring	32.6	32.6	300
3	Stand	81.6	81.6	800

A comparison of stresses induced in the mirror for leaf spring and bipod ring mount are shown in the form of graphs (figure 14 and 15) when they are subjected to inertia and thermal loads respectively.

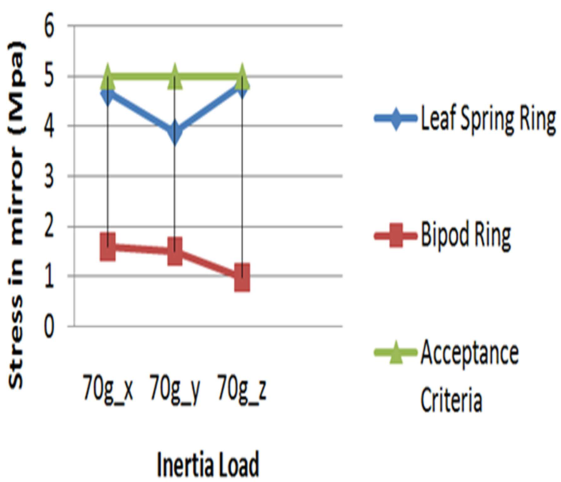

Figure 14: Stress in mirror for inertia loads with leaf spring and bipod ring mounts

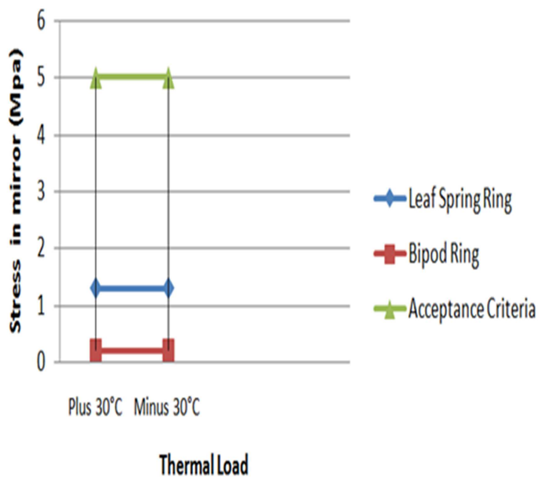

Figure 15: Stress in mirror for thermal load with leaf spring and bipod ring mounts

4.5. Analysis of Secondary Mirror Mount

4.5.1. Modal Analysis

Boundary condition: The base is fixed with 6 degrees of freedom. Figure 16 shows the finite element model of secondary mirror with leaf spring ring mount. Figure 17 shows the Modal analysis result. Natural Frequency obtained is 499.11 Hz

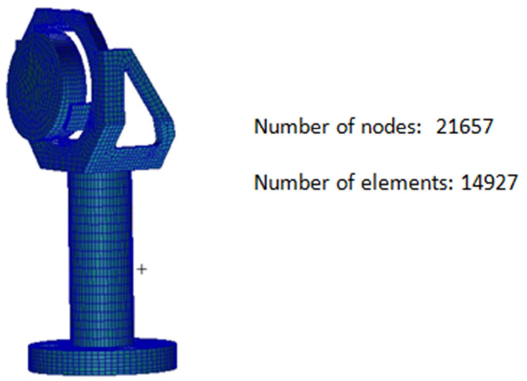

Figure 16: Finite element model of secondary mirror opto-mechanical system

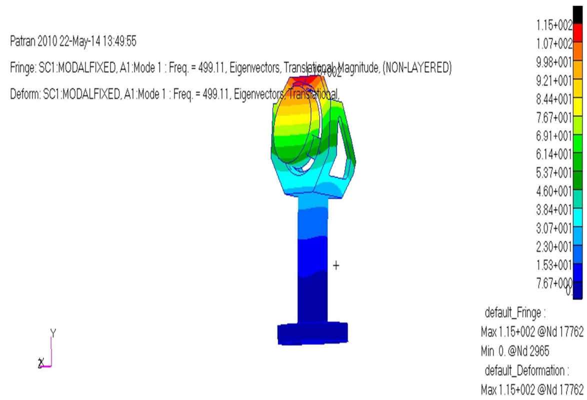

Figure 17: Modal analysis result of secondary mirror opto-mechanical system

4.5.2. Linear Static Analysis

Boundary condition for quasi-static loads: Applying Miles' equation (equation 1) for the secondary mirror opto-mechanical system, we get

$$G_{RMS} = \sqrt{\frac{\pi * 499.11 * 0.1}{2 * 2 * 0.05}} = 28$$

The mirror should be designed to withstand a minimum random vibration level of 3 * 28 = 84 times gravity. The secondary mirror mount is designed to withstand an inertial load of 90 g.

Boundary condition for thermal loads: Taking 20°C as the ambient temp. The system is subjected to a temperature variation of +30°C and -30°C. (ΔT=30°C). Figure 18 shows some of the FEA results of the secondary mirror opto-mechanical components.

Figure 18 (a) shows stress in the mirror due to 90g inertia load in Y direction. Figure 18 (b) shows stress in the ring due to 90g inertia load in Z direction. Figure 18 (c) shows stress in the stand due to 90g inertia load in X direction. Figure 18 (d) shows stress in the mirror due to minus 30°C thermal load. The stresses in mirror, leaf spring ring mount and stand with 90g inertia load imposed in X, Y and Z directions and due to variation in temperature are presented in section 4.5.3, summary of results.

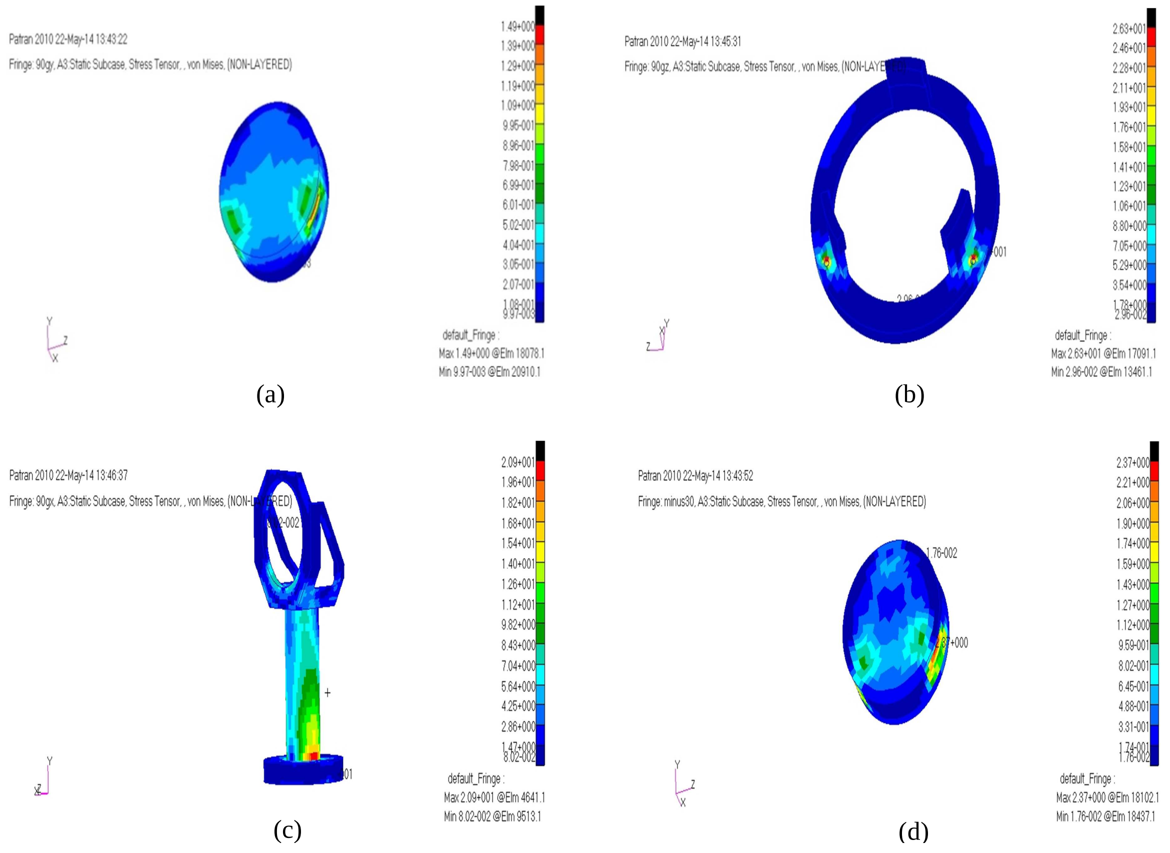

Figure 18: FEA results of the secondary mirror opto-mechcanical components

4.5.3. Summary of results

Tables 7 and 8 show the summary of results for components of secondary mirror subjected to quasi-static and thermal loads respectively. The stresses for all the individual components are given.

Table 7: Quasi-Static Analysis results for components of secondary mirror opto-mechanical system

Sl. No.	Component	Von Mises Stress(Mpa) in component for 90g load			Allowable stress(Mpa)
		X	Y	Z	

1	Mirror	1.71	1.49	1.02	5
2	Ring	12.4	18.7	26.3	300
3	Stand	20.9	5.93	20	800

Table 8: Thermal stress analysis results for components of secondary mirror opto-mechanical system

Sl. No.	Component	Von Mises Stress(Mpa) in component due to thermal loads		Allowable stress(Mpa)
		+30°C	-30°C	
1	Mirror	2.37	2.37	5
2	Ring	31	31	300
3	Stand	87.2	87.2	800

5. Analytical Calculations

5.1. Glue Area

Guidelines for determining the appropriate adhesive area for bonding prisms to mechanical mounts have appeared in the literature (Yoder, 1988, 2002) [2]. In general, the minimum area of the bond, Q_{MIN}, is determined by equation 2.

$$Q_{MIN} = W \frac{a_G f_s}{J} \tag{2}$$

Where,

W *is* the weight of the optic

a_G is the worst-case expected acceleration factor

f_s is the safety factor

J is the shear or tensile strength of the adhesive joint (usually approximately equal).

A safety factor of at least 2 and possibly as large as four to allow for some unplanned, non optimum condition, such as inadequate cleaning during processing, is to be taken.

For primary mirror, we have

$$Q_{MIN} = 2.3\frac{9.81 * 70 * 2}{17.2} = 183.65 \text{ mm}^2$$

Since the load is shared by 3 blades, a minimum bond area of 183.65/3 = 62.25 mm^2 per blade is required to withstand the inertia loads. The bond area chosen for primary mirror mount is 611.56 mm^2.

For secondary mirror, we have

$$Q_{MIN} = 0.065\frac{9.81*90*2}{17.2} = 6.67 \text{ mm}^2$$

For the secondary mirror a minimum bond area of 2.22 mm^2 per blade is required to withstand the inertia loads. The bond area chosen for secondary mirror mount is 76.97 mm^2.

5.2. Stresses in Cemented and Bonded Optics Due to Temperature Changes

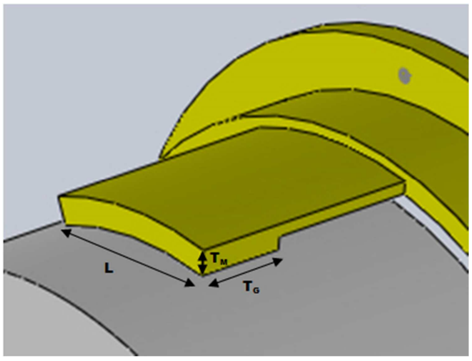

Fig. 19: Estimation of shear stress in adhesive

Vukobratovich (2002, 2003) [3] presented an analytical method, based on work by Chen and Nelson (1979), for estimating the shear stress developed in a bonded joint as a result of differential dimensional changes at temperatures other than that at assembly. Figure 19 shows the dimensions of the leaf spring which is used for estimating the shear stress in the adhesive. The pertinent equations are as follows:

$$S_s = \frac{(\alpha_M - \alpha_G)\,\Delta T\, S_e \tanh(\beta L)}{\beta\, t_e} \tag{3}$$

$$S_e = \frac{E_e}{2(1+\nu_e)} \tag{4}$$

$$\beta = \left[\left(\frac{S_e}{t_e}\right)\left(\frac{1}{E_M\, t_M} + \frac{1}{E_G\, t_G}\right)\right]^{1/2} \qquad (5)$$

Where,

S_s is the shear stress in the joint

α_M and α_G is the CTEs of the metal and glass, respectively

ΔT is the temperature change from assembly temperature

S_e is the shear modulus of the adhesive

tanh is the hyperbolic tangent function

L *is* the largest dimension (length, width, or diameter) of the bond

t_e is the thickness of the bond

E_M and E_G is Young's modulus for the metal and glass, respectively

ν_e is the Poisson's ratio for the adhesive

t_M and t_G is the thicknesses of the metal and glass components, respectively.

For primary mirror, we have

$$E_e = 689.5\ Mpa\ ;\ \ \nu_e = 0.43;\ \alpha_M = 1.26 * 10^{-6}\ /°C\ ;\ \alpha_G = 5 * 10^{-8}\ /°C\ ;\ L = 38.22mm\ ;\ t_M = 5\ mm$$

$$t_G = 16\ mm\ ;\ t_e = 0.1mm\ ;\ S_e = \frac{689.5}{2(1 + 0.43)} = 241.08\ Mpa$$

$$\beta = \left[\left(\frac{241.08}{0.1}\right)\left(\frac{1}{147000*5} + \frac{1}{90600*16}\right)\right]^{1/2} = 0.070\ /mm\ \ ;\ \beta L = 0.070 * 38.22 = 2.6754$$

$$S_s = \frac{(1.26 * 10^{-6} - 5 * 10^{-8})(30)(241.08)\ tanh(2.6754)}{0.070 * 0.1} = 1.23\ Mpa$$

This stress causes no damage to the mirror.

For secondary mirror, we have

L=24.50 mm ; $t_M = 2.94\ mm$; $t_G = 6\ mm$; $t_e = 0.1mm$; $S_s = 0.86\ Mpa$

This stress causes no damage to the mirror.

5.3. Thermal Stresses Developed Due to Temperature Excursions

Tensile or compressive stress may be caused by restricting thermal expansion. Thermal stress can be calculated as

$$\sigma = E\,\varepsilon \qquad (6)$$

$$= E\,\alpha\,\Delta T$$

Where

σ = stress due to temperature expansion (MPa)

E =Young's Modulus (Mpa)

ε = strain

α = temperature expansion coefficient ($/^oC$)

ΔT = temperature difference (^oC)

Differential displacement (ΔL) = Δα x ΔT x (Mirror Diameter /2)

$= (1.26 – 0.05) \times 30 \times (146/2) \times 10^{-6} = 2.6495 \times 10^{-3}$

$$\varepsilon = \Delta L / L \qquad (7)$$

$= (2.6495 \times 10^{-3}) / (146) = 1.815 \times 10^{-5}$

$\sigma = E\,\varepsilon = 90600 \times 1.815 \times 10^{-5} = 1.64$ Mpa

The analytical and FEA results are compared and shown in the table 9 below.

Table 9: Comparison of Analytical and FEM Results due to Thermal Load

Component	Stress by Analytical Calculation	Stress from FEA Result
Mirror	1.6 Mpa	1.3 Mpa

5.4. Stress Induced in Mirror due to Inertia Loads

Stresses are induced in the mirror when the mirror is subjected to inertia loads during launching of the rocket. This section gives an analytical way to calculate the stress.

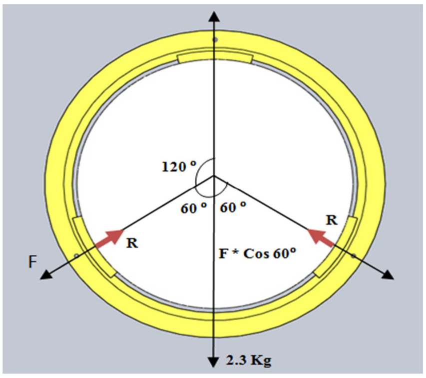

Figure 20: Stress in mirror due to inertia loads

Consider the free-body diagram shown in figure 20. The weight of the mirror is 2.3 Kg and it acts vertically downwards. F is the force acting on the blade due to weight of the mirror.

Force = Mass * Acceleration

Resolving the force (F), we get F * cos 60^o = m*g

= 2.3 * 9.81=22.563 N

For 70 g load, F * cos 60^o = 22.563*70

F= 3158.82 N

Since the load is shared by the two bottom blades,

$$\frac{F}{2} = \frac{3158.82}{2} = 1579.41\ N$$

R is the reaction force acting on the mirror.

$$Stress = \frac{Reaction\ Force}{Contact\ area\ of\ blade} = \frac{1579.41}{611.56} = 2.58\ Mpa\ \approx 2.6\ Mpa$$

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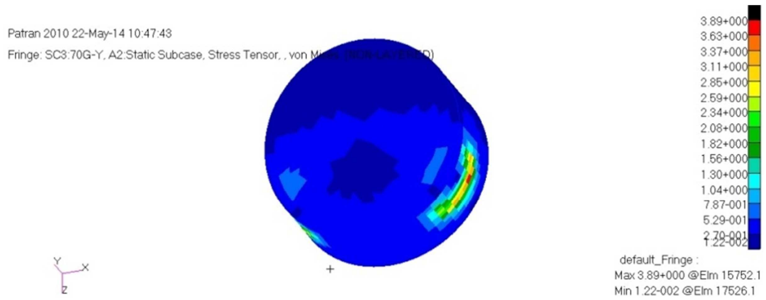

Figure 21: Primary mirror subjected to 70g inertia load in –Y direction

The analytical and FEA results are compared and shown in the table 10 below

Table 10: Comparison of Analytical and FEM Results due to Inertia Load

Component	Stress by Analytical Calculation	Stress from FEA Result
Mirror	2.6 Mpa	3.89 Mpa

In the analytical approach for calculation of stress in the mirror due to inertia loads, it is assumed that the mirror is of symmetric mass. However, the mirror used in the finite element model (figure 21) is of asymmetric mass. This explains the variation of analytical results with FEA results.

6. Conclusion

1. The opto-mechanical design implemented for primary and secondary mirrors in this project has yielded successful results. The overall weight of the assembly (primary mirror mount + secondary mirror mount) has met the mass requirements (<6.5 kg) by suitable optimization.
2. The opto-mechanical system should withstand the vibrations encountered during launching of the rocket. The natural frequency of the system (>200 Hz) should be away from the operating range of frequency (10-100 Hz). It should also survive the effects due to random vibration as the system is designed for higher inertia loads using Miles' equation [1].
3. The linear static analysis result of the opto-mechanical assembly has met the required targets. Mirror being a brittle material, the main concern in this project is whether the mirror would withstand the changes in environmental conditions. The stresses induced in the mirror due to thermal and inertial loads are well within the acceptance criteria (< 5 Mpa). The stresses induced in the mounts are also within the yield stress of the material which indicates the design of the assembly is safe.
4. Two types of ring mounts are compared. Namely leaf spring ring mount and bipod ring mount. From the finite element analysis results it is observed that the stresses induced in the mirror due to inertia and

thermal loads reduce when bipod ring mount is used. Bipod ring mount has given better results when compared to leaf spring ring mounts.

5. The materials/material properties selected for this work is ideal. Invar can be used as ring mount as it is compatible with Zerodur (Mirror). Titanium can be used as it has high strength/stiffness ratio. The glue material (EC2216) employed in the contact area is also compatible as the shear stress induced in the glue due to temperature excursions is low (< 5 Mpa), which keeps the mirror safe and it has better shear strength than most of the adhesives.
6. Analytical calculations for thermal stresses developed due to temperature excursions and stress induced in the mirror due to inertia loads comply with the FEA results. The validity of the FEA results is established. Hence, it can be concluded that the mounting configuration employed in this work can be used for mounting small off-axis mirrors for the optical systems used for the space applications.

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