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Design Optimization of Bi-Pod Flexural Mount for Large Size Light Weighted Mirror for Space Applications

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ABSTRACT

Space borne high resolution imaging optical mirrors for space applications are designed to perform at their near diffraction limit performance and the space optics large size mirrors are to be light weighted in order to improve the satellite agility and mission life. To obtain such near diffraction limited performance, the optical surfaces of the mirror are fabricated and characterized to the surface accuracy of better than λ/60rms under zero gravity space environment which calls for optimum design of light weight pattern, Mechanical Fixation Devices (MFD) considering appropriate bonding and de-bonding methods for mounting. While fabricating and testing such large size light weighted mirror on ground [in 1g environment], the most significant effect on the mirror are: 1.Deflection due to its self weight [under gravity], 2. Distortion due to temperature effects on the mirror. 3. Distortions due to gluing with the MFDs. To overcome the gravity effect, the mirrors are supported typically on suitable mirror support systems and proper mechanical mounting devices are used for holding the mirror. The mirror support design is critical to ensure the requisite the optical performance. Detailed analysis has to be conduct using various finite element modeling methods. Parts have to be built and the results have to be verifying by conducting experimental simulations using samples. The geometry and structure of MFD is optimized through analytical methods using FEM.

Key words: Optomechanical Design, Mechanical fixation device, Zernike Polynomials, and Mirror Mount.

1. INTRODUCTION

Opto-mechanical engineering is a multi disciplinary field involving many technical disciplines such as materials, mechanical engineering, mirror design, lens design, tolerance analysis, and optics fabrication, glass to metal face interface analysis, thermodynamics, and many other engineering disciplines. Positioning the optical components of an optical system relative to each other and mounting of an optical elements with free of strain are the two primary concepts of opto-mechanical engineering. Strain is very important than the stress in opto-mechanical design. Self weight due to gravity is the most common load which is acting on the optical systems in the space. In mechanical engineering structural efficiency is based on the strength to weight ratio (E/ρ) where as in opto-mechanical structural efficiency is based on stiffness to weight ratio. Fundamental frequency is the main parameter which determines the structural efficiency of the opto-mechanical elements/support systems.

The performance of the mirror assembly requires the control of geometric stability, geometrical positioning and optical surface deformations against load transfer effects. The different loads which are influencing the mirror assembly performance may be manufacturing processes of the mirror, its mount, assembly, interface effects, gravity release static load, dynamic loads, thermal loads and other environmental load influences like pressure, humidity, corrosion etc.

The requirements of the optical surface can be achieved by fixing the optical elements to the mechanical support system which is called as the semi kinematic support system, also called as mirror fixation device. In kinematic systems the stiffness is more and deflection is very less. In order to meet the optical system requirements, the deflection of the mounts should be less and also should be stiffer enough to hold the mirror. Since the kinematic support systems are include the friction, backlash effects and other environmental effects. The deflections in the mounts should be such that it should be take care of more stresses rather than allowing the mirror to distort and ensure the better surface accuracy of the mirror.

The mirror mounts shall include kinematic, semi kinematic support systems of the light weight mirror during its testing such as whiffletree mounts, gimbal mounts, strap mounts etc and finally the Mechanical Mirror Fixation Devices (MFD's) such as Bi-pod flexural mounts which is fixed to the mirror by gluing with the adhesive to become a part of the mirror assembly.

The mounts used to support the optical mirror in space, which is designed for desired applications are reported in the literature survey. All of them have been designed to meet the optical performance requirements of the particular optical mirror and thus they are designed based on the application of the mirror and its requirement. And thus they are mirror specific. Here the mirror mount used is the Bi-pod flexural mount which is called as the Mechanical Mirror Fixation Device (MFD). There is no such kind of the closed form solutions to get an acceptable configuration of Mechanical Mirror Fixation Devices (MFDs). However the basic base line configurations may be generated with the closed form solutions which are available and are practically feasible. The same may be then optimized by changing the different geometrical parameters and by going through the finite element analysis till desired requirements of the optical performance is met.

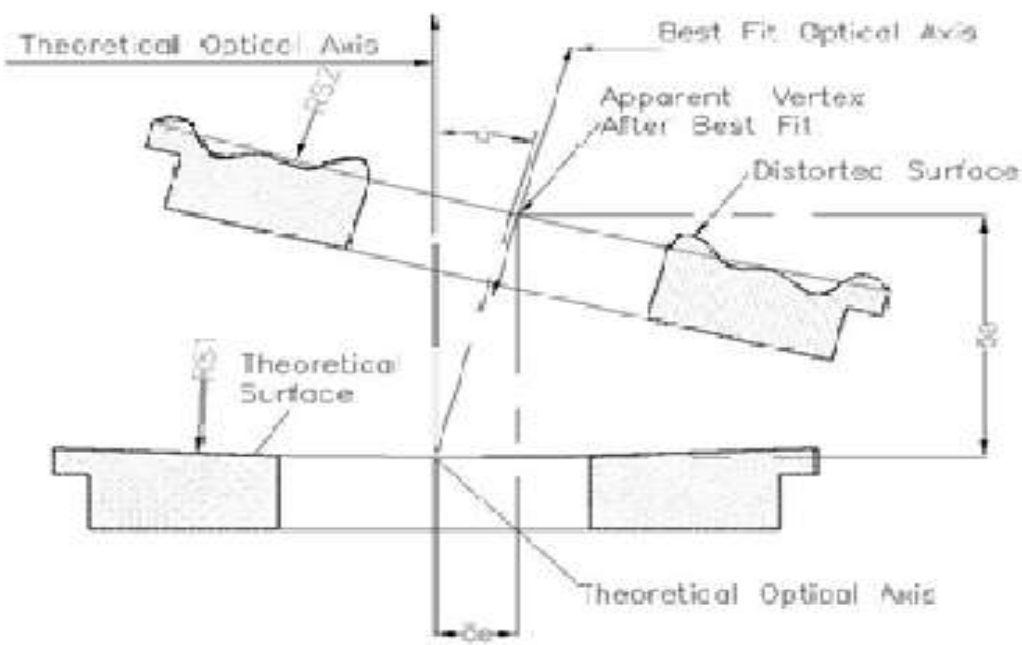

Figure.1: Typical deformed optical surface of mirror assembly under loading conditions.

Basic requirements of an optical element onboard the space craft are:

- Optical elements should be robust to take up the static and dynamic loading during launch and ground handling.
- It should function in the extreme temperatures.
- It should be light weight to meet the mass requirements.
- It should meet the optical requirements under static, dynamic, and thermal loading condition.

2. PRINCIPLES:

The configuration of Bi-Pod flexures Mounting of Mirror is shown in figure 2. The mirror is fabricated and having the light weighted pattern back side of the mirror. It has six elliptical Pockets which are extrude cut at the back side of the mirror having 60 degree each pocket with the Pitch circle diameter of approximately 500mm for mounting the Bi-Pod Flexures 120° apart for 60° orientation.

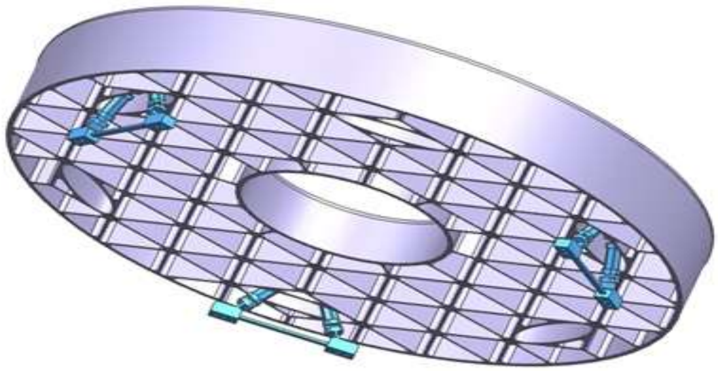

Figure 2: Geometrical Model of Optical System Assembly.

The Bi-Pod Flexure is coupled with the mirror by using an Epoxy Adhesive called 3M EC-2216 Gray. The Bi-Pod Flexure has the elliptical Pad which is fixed to the mirror's elliptical Pocket permanently using the Epoxy Adhesive. The apex of the triangle formed by a Bi-Pod Flexure Mount should point to the CG of the mirror in order to minimize the front surface distortion of the mirror due to gravity. The small amount of misalignment will lead to the Astigmatic wave front error.

3. DESIGN OF BI-POD FLEXURE MOUNT

The Bi-pod flexure mounts are configured to mount the optical mirrors used for space applications, such as light weight large size mirror, telescopes etc. to ensure the surface accuracy of the optical mirror in the space environment under self weight and dynamic loading conditions. The main objective of the bi-pod flexure mounts is to mount the mirror with strain free, and to not induce any stresses on the mirror for dynamic and self weight of the mirror in space environment. Since the dynamic load required a highly stiff mount and thermal loads require compliance or a flexural mount, a trade off is to be made for optimizing both thermal and dynamic load transfer effects in a given mount.

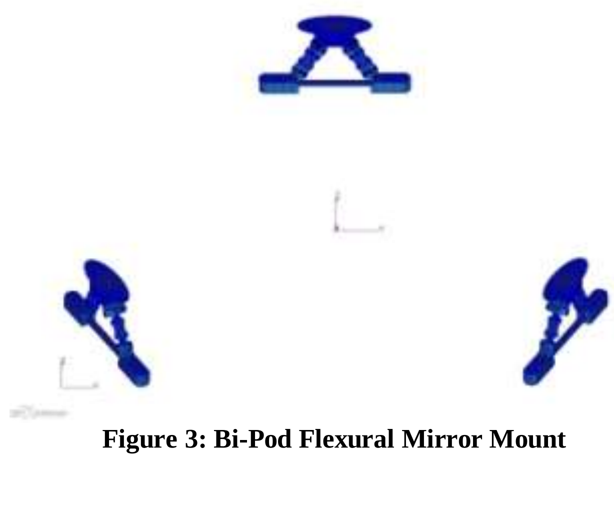

Figure 3: Bi-Pod Flexural Mirror Mount

The geometry of the Bi-Pod Flexure is a very important parameter which is deciding the stress and mode frequency of the Mount. Various Shapes and Size of the Mounts were designed for this particular application using the CAD software. Figure shows one of the Bi-Pod Flexure designed for this particular application.

4. FEM SIMULATION

PATRAN & NASTRAN tools were used for Finite Element Analysis in simulating the actual environment with the given boundary condition and measure the deflection of the front surface of the mirror and stresses developed in the mount due to gravity loading as well as the temperature loading. The Meshing of the model is done by the FEA software (PATRAN) using CHEXA 8 Elements. Figure shows the solid mesh of the model of the optical system assembly.

4.1. DESCRIPTION OF FINITE ELEMENT MODEL

The finite element model shown in the figure 4 which is having CHEXA elements modeled in PATRAN /NASTRAN Software.

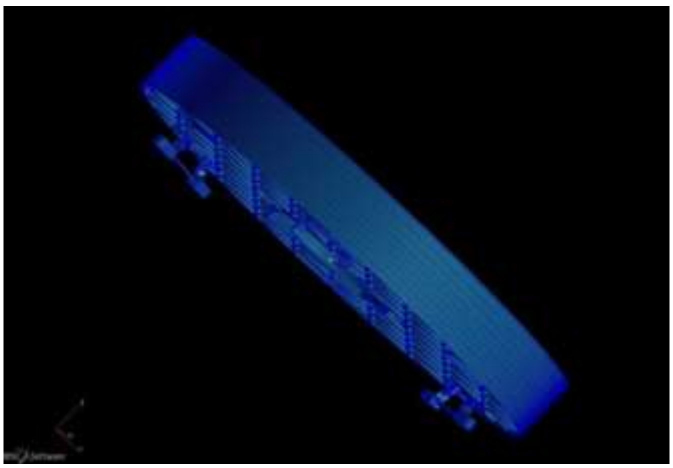

Figure.4 The finite element model of the Optical mirror assembly.

4.2. MATERIALS PROPERTIES:

The following Mechanical properties of different Materials are used in the FE analysis are tabulated in table 4.1.

Table 4.1 The Material Properties of the Optical System.

PARTS	MIRROR	MFD	GLUE
MATERIALS	**ZERODUR (SCHOOT)**	**SOFT INVAR**	**EPOXY EC 2216 GRAY B/A**
Elastic modulus E (MPa)	90600	147000	300
Poisson ratio	0.24	0.29	0.46
Density ρ (Kg/m^3)	2530	8050	1250
Coefficient of thermal expansion α (K^{-1})	$0.05x10^{-6}$	$1.26x10^{-6}$	$100.0x10^{-6}$
Thermal conductivity K (W/mK)@300K	1.64	12.6	
Material yield strength (MPa)	10	300 (0.2%Hysterisis)	5 (150°C<T<250°C)
Ultimate strength (MPa)	60	500	

4.3. FEM ANALYSIS EXERCISES:

Following exercises are very essential (but not limited to) for evaluating the Mirror mount performance:

- Natural frequency of assembly with both free-free and base rigidly clamped.
- 33g quasi static loading for the assembly along optical axis and normal to Optical axis.
- Temperature excursion for $\Delta T \pm 40°C$ with reference to reference temperature of 20 °C.

4.4. ACCEPTANCE CRITERIA:

- In all the FEM analysis exercises the performance is evaluated in terms of stresses developed in Mirror.
- The MYS of the mirror should not exceed 10 MPa and in Glue should not exceed 5 MPa to static load of 33g.

Table 4.2: Criteria for safety margins

Sl. No.	Material	Criteria
1.	Zerodur	$\sigma_{ys} <$ 10 N / mm2
2.	Glue EC 2216 (3M)	$\sigma_{ys} <$ 5 N / mm2 (Stability)
3.	Invar	$\sigma_{ys} <$ 300 N / mm2

4.5. LOADS AND BOUNDARY CONDITIONS:

The following loads and boundary conditions are analyzed for the Optical System:

a. Modal analysis:

- **Clamped Natural Frequency:** The base of the Bi-Pod Flexure Mount (MFD) is rigidly fixed with all six degrees of freedom and analyzed without any other loads.
- **Free-Free Natural Frequency:** The Mirror alone is analyzed without fixing and loads.

b. Quasistatic Analysis:

The base of the MFD is fixed with the all six degrees of freedom and following loads are analyzed.

1. 33g along X axis.
2. 33g along Y axis.
3. 33g along Z axis.

c. Temperature Loads:

The base of the MFD is fixed with the all six degrees of freedom and following Temperature loads are analyzed. Reference Temperature is 20°C

1. +50°C.
2. -10°C.

4.6. BENCH MARK EXERCISE:

4.6.1. The MFD is optimized in the following methods:

1. The figure.5 shows the Benchmark Exercise carried out for the MFD in which the mirror's weight is applied on the MFD in the Y direction in order to ensure the surface figure of the mirror, due to the deflection of the MFD in radial direction. The MFD is optimized to satisfy the mirror requirements by smaller deflection in the radial direction of the mirror. Since the mirror deflection in the radial direction due to the temperature loads is given by:

$$\delta = \alpha \times \Delta T \times L$$

Where ,

α is the coefficient of thermal expansion
ΔT is the temperature difference
L is the diameter of the mirror

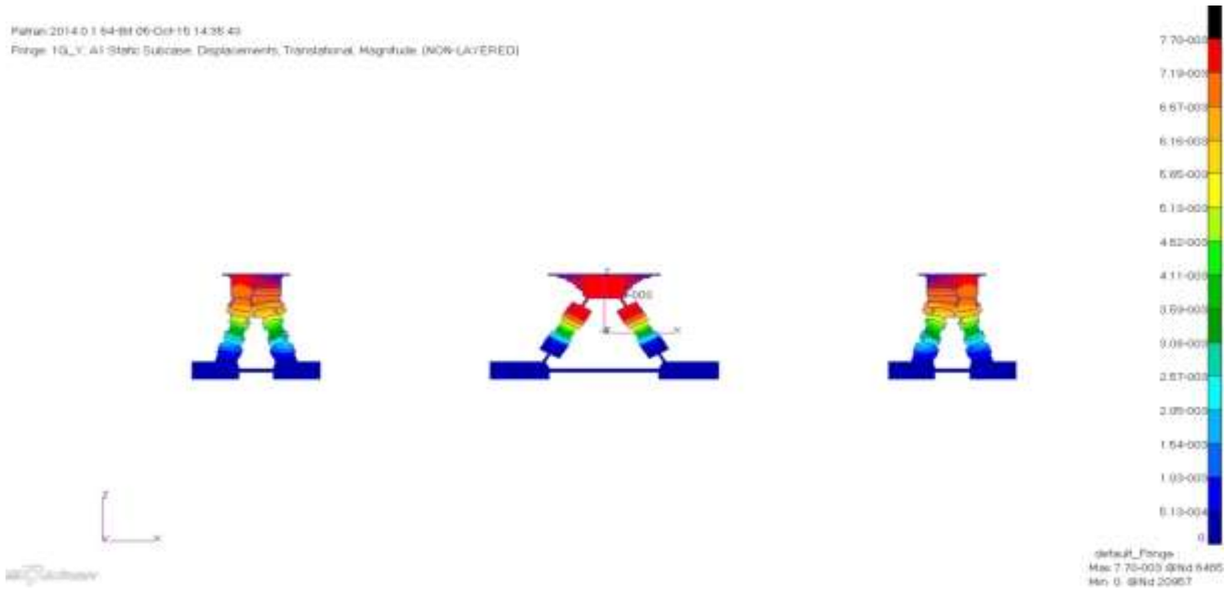

Figure 5: The bench mark exercise of deflection of the MFD.

Form the above equation the radial deflection of the mirror is known. In order to ensure the mirror's surface figure the MFD should deflect lesser than that. Hence the MFD deflection due to the mirror's weight lesser than the radial deflection of the mirror due to temperature loads. Hence the MFD is optimized for the application. Figure 5 shows the deflection of MFD in the Y direction due to the mirror's weight.

2. Since the mount has to deflect less and it should be stiffer enough to hold the mirror in order to maintain the mirror's surface accuracy the optimization of the flexures of the mount is necessarily needed. In order to increase the stiffness of the mount to achieve the low deflection in the mount the mount's flexure thickness is varied to meet the requirements. Hence the Flexure thickness is optimized and the thickness of the flexure is approximately 4mm.

3. Since the mount has to fix in the proper position in order to achieve the better surface accuracy of the mirror the Pitch Circle Diameter of the MFD pockets are very important. From the pitch circle diameter the performance of the mirror will affected which is known as Zernike polynomial called coma. The value of the coma is minimized by varying the PCD of the MFD Pockets. The value of the PCD is optimized approximately 500mm.

4. Astigmatism is one of the Zernike Polynomial which is helpful to analyze the phase map of the mirror surface deformation; this is affected due to the MFD Pocket depths. In order to meet the mirror's surface accuracy the MFD pocket depth has to be optimized. The optimized value of the MFD pocket depth is approximately 85mm.

5. ZERNIKE POLYNOMIALS

In order to analyze performance of the mirror, the deflection on the mirror surface is computed interferometrically which is a non contact method of testing the mirror surface deflection. In this process the phase map of thsse mirror is analyze with a set of polynomial known as Zernike polynomial.

Zernike polynomials are useful in the interpretation of optical surface data, including finite element derived surface deformations and wave front maps due to different loading conditions (test conditions). The Zernike polynomial represents a set of discrete data by a series of base surfaces each multiplied by a coefficient and summed. For example, surface data, Δ, may be represented by the polynomial series ϕ_i given in the equation. The polynomials terms are summed with the contribution of each polynomial determined by the coefficients a_i .

$$\Delta = \sum_i a_i \phi_i = a_0 + a_1\phi_1 + a_2\phi_2 + a_3\phi_3 + ... + a_i\phi_i \qquad .... .1$$

The Zernike polynomial has several features that are particularly useful to optical systems.

1. The polynomials set are orthogonal over a normalized circular aperture. Deleting terms from or adding terms to the set of polynomials does not affect the value of the remaining or the original coefficients.

2. These polynomials are fitted to the given surface data using least square fit method. I.e. the polynomials are minimized with respect to the RMS departure from the reference surface. Thus using truncated set of Zernike terms to describe the data set or deleting terms from the Zernike set will always results in a decrease in the RMS of surface data.
3. Zernike terms may be related to one of the classical Seidel aberrations used to describe optical aberrations.
The Zernike polynomials are a complete set of polynomials with variables in radial, r, and azimuthally, ɵ, extent. A complete mathematical description for a given Zernike surface ΔZ(r, ɵ), is provided by equation where A_{nm} and B_{nm} are the Zernike coefficients:

$$\Delta Z(r,\theta) = A_{00} + \sum_{n=2}^{\infty} A_{n0} R_n^0(r) + \sum_{n=1}^{\infty}\sum_{m=1}^{n} R_n^m [A_{nm}\cos(m\theta) + B_{nm}\sin(m\theta)] \qquad \text{...........2}$$

The radial depends of Zernike polynomials is given by the following expression:

$$R_n^m(r) = \sum_{s=0}^{\frac{n-m}{2}} (-1)^s \frac{(n-s)!}{s!(\frac{n+m}{2}-s)!(\frac{n-m}{2}-s)!} r^{(n-2s)} \qquad \text{.................3}$$

The variables n and m in the equations are integer values and are known as radial and circumferential wave number, respectively. There exists few caveats in deriving the individual Zernike terms from the Zernike equations listed above, where in : (a) n-m must be an even number, (b) n ≥ m. There are two popular set of Zernike polynomials that differ only in the way they are ordered. The standard set has an infinite number of terms. The fringe set is reordered subset of Standard Zernike terms, with total of 37 terms. The fringe set includes the higher order radially symmetric terms while excluding the higher order terms. Usually the **fringe set** is used for interpretation of optical test results by Interferometry.

Each of the Zernike terms is described below:

1. Piston:
The first term of Zernike series, piston, is just a DC terms representing the bias to original data.

2. Tilt:
The tilt terms represents tilt of optical surface as a whole. The tilt can be removed by aligning the test set up.

3. Focus or Power:
Focus represents a quadratic or parabolic change in the radial extent of the surface shape.

4. Astigmatism:
Astigmatism is best described as shape of a horse's saddle, or potato chip, possessing unequal curvatures along perpendicular axes.

5. Coma:
Coma is a surface with a pair of humps, where one of the humps is inverted.

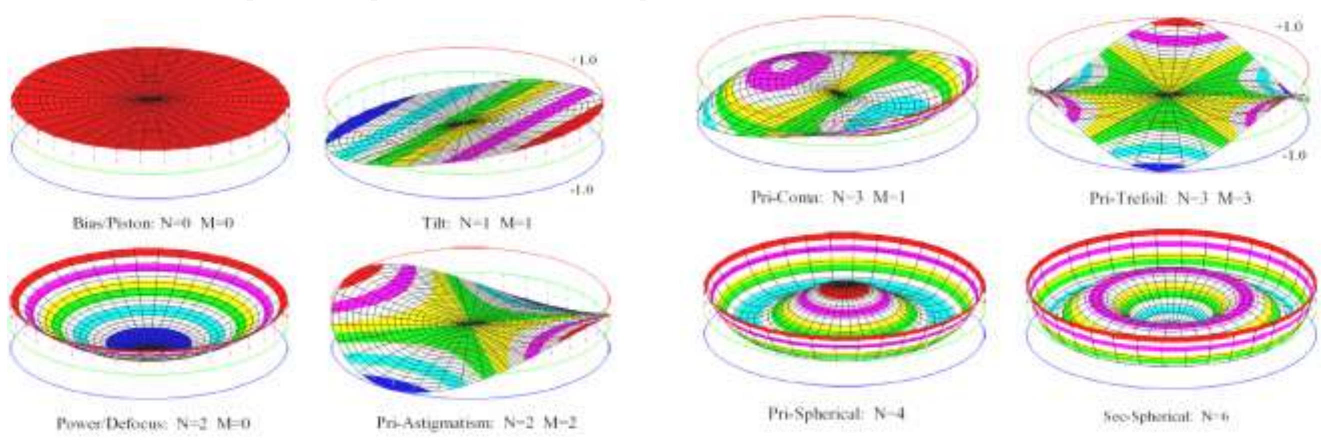

Figure 6: Zernike Aberration Coefficients (Typical Phase Map Plots)

The 36 fringe Zernike polynomial terms in polar coordinates are given in the below table.

Table 5.1: List of Zernike coefficients

Sl. No	ORDER	ZERNIKE POLYNOMIAL TERM	Name
1	N=0, m=0	1	Piston
2	N=1, m=-1	$r\cos(\theta)$	Tilt x
3	N=1, m=1	$r\sin(\theta)$	Tilt y
4	N=2, m=0	$2r^2-1$	Focus
5	N=2, m=-2	$r^2\cos(2\theta)$	Astig x
6	n=2, m=2	$r^2\sin(2\theta)$	Astig y
7	n=3, m=-1	$(3r^3-2r)\cos(\theta)$	Coma x
8	n=3, m=1	$(3r^3-2r)\sin(\theta)$	Coma y
9	n=4, m=0	$1-6r^2+6r^4$	Pri spherical
10	n=3, m=-3	$r^3\cos(3\theta)$	Trefoil x
11	n=3, m=3	$r^3\sin(3\theta)$	Trefoil y
12	n=4, m=-2	$(-3r^2+4r^4)\cos(2\theta)$	sec.astig x
13	n=4, m=2	$(-3r^2+4r^4)\sin(2\theta)$	sec.astig y
14	n=5, m=-1	$(3r-12r^3+10r^5)\cos[\theta]$	sec.com x
15	n=5, m=1	$(3r-12r^3+10r^5)\sin[\theta]$	sec.com y
16	n=6, m=0	$-1+12r^2-30r^4+20r^6$	sec. spherical
17	n=4, m=-4	$r^4\cos[4\theta]$	Tetra foil x
18	n=4, m=4	$r^4\sin[4\theta]$	Tetra foil y
19	n=5, m=-3	$(-4r^3+5r^5)\cos[3\theta]$	sec.trefoil x
20	n=5, m=3	$(-4r^3+5r^5)\sin[3\theta]$	sec.trefoil y
21	n=6, m=-2	$(6r^2-20r^4+15r^6)\cos[2\theta]$	Terit.astig x
22	n=6, m=2	$(6r^2-20r^4+15r^6)\sin[2\theta]$	Terit.astig y
23	n=7, m=-1	$(-4r+30r^3-60r^5+35r^7)\cos[\theta]$	Terit.Com x
24	n=7, m=1	$(-4r+30r^3-60r^5+35r^7)\sin[\theta]$	Terit.Com y
25	n=8, m=0	$1-20r^2+90r^4-140r^6+70r^8$	Terit.Spherical
26	n=5, m=-5	$r^5\cos[5\theta]$	Penta foil x
27	n=5, m=5	$r^5\sin[5\theta]$	Penta foil y
28	n=6, m=-4	$(-5r^4+6r^6)\cos[4\theta]$	sec.tetfoil x
29	n=6, m=4	$(-5r^4+6r^6)\sin[4\theta]$	sec.tetfoil y
30	n=7, m=-3	$(10r^3-30r^5+21r^7)\cos[3\theta]$	Teri.trefoil x
31	n=7, m=3	$(10r^3-30r^5+21r^7)\cos[3\theta]$	Teri.trefoil y
32	n=8, m=-2	$(-10r^2+60r^4-105r^6+56r^8)\cos[2\theta]$	Quat.astig x
33	n=8, m=2	$(-10r^2+60r^4-105r^6+56r^8)\sin[2\theta]$	Quat.astig y
34	n=9, m=-1	$(5r-60r^3+210r^5-280r^7+126r^9)\cos[\theta]$	Quat.com x
35	n=9, m=1	$(5r-60r^3+210r^5-280r^7+126r^9)\sin[\theta]$	Quat.com y
36	n=10, m=0	$(252r^{10}-630r^8+560r^6-210r^4+30r^2$	Quat.Spherical
37	n=12, m=0	$1-42r^2+420r^4-1680r^6+3150r^8-2772r^{10}+924r^{12}$	Quin spherical

6. RESULTS AND DISCUSSIONS

6.1. FRONT SURFACE DEFLECTION ANALYSIS:

The Zernike coefficient values for the front surface deflection due to 1g loading along the optical axis.

Table 6.1: The Zernike coefficients obtained for the 1g loading in Z direction.

NAME	CONVERSION FACTOR	CONVERSION FACTOR	OBTAINED ZERNIKE
Piston	1	1	0.00004133
Tilt x	1/2	0.5	0.00003456
Tilt y	1/2	0.5	-0.09023299
Focus	1/√3	0.577350269	-0.0000311
Astig x	1/√6	0.40824829	0.01766358
Astig y	1/√6	0.40824829	0.000034
Coma x	1/√8	0.353553391	0.000044
Coma y	1/√8	0.353553391	-0.03256912
Pri spherical	1/√5	0.447213595	-0.000013
Trefoil x	1/√8	0.353553391	-0.000035
Trefoil y	1/√8	0.353553391	0.000044
sec.astig x	1/√10	0.316227766	0.02244748
sec.astig y	1/√10	0.316227766	-0.00002835
sec.com x	1/√12	0.288675135	0.000007
sec.com y	1/√12	0.288675135	-0.00451772
sec. spherical	1/√7	0.377964473	-0.00002016
Tetra foil x	1/√10	0.316227766	0.0583305
Tetra foil y	1/√10	0.316227766	0.000010
sec.trefoil x	1/√12	0.288675135	0.000012
sec.trefoil y	1/√12	0.288675135	0.00003955
Terit.astig x	1/√14	0.267261242	-0.03487468
Terit.astig y	1/√14	0.267261242	0.000015
Terit.Com x	1/√16	0.25	-0.000026
Terit.Com y	1/√16	0.25	0.00248091
Terit.Spherical	1/√9	0.333333333	0.0000413
Penta foil x	1/√12	0.288675135	-0.000017
Penta foil y	1/√12	0.288675135	-0.0294956
sec.tetfoil x	1/√14	0.267261242	-0.02849587
sec.tetfoil y	1/√14	0.267261242	-0.00003719
Teri.trefoil x	1/√16	0.25	-0.000006
Teri.trefoil y	1/√16	0.25	-0.00002088
Quat.astig x	1/√18	0.23570226	0.02579537
Quat.astig y	1/√18	0.23570226	0.000017
Quat.com x	1/√20	0.223606798	-0.000024
Quat.com y	1/√20	0.223606798	0.00365518
Quat.Spherical	1/√11	0.301511345	-0.00002851

The Whiffle tree performance of the front surface deflection due to 1g loading along the optical axis found as RMS value $\lambda/91.9$. The RMS value of the mirror with MFD mounts 1g loading in the Y direction found as $\lambda/34.7$. The surface figure of the mirror with MFD should be better than the $\lambda/30$ can be acceptable.

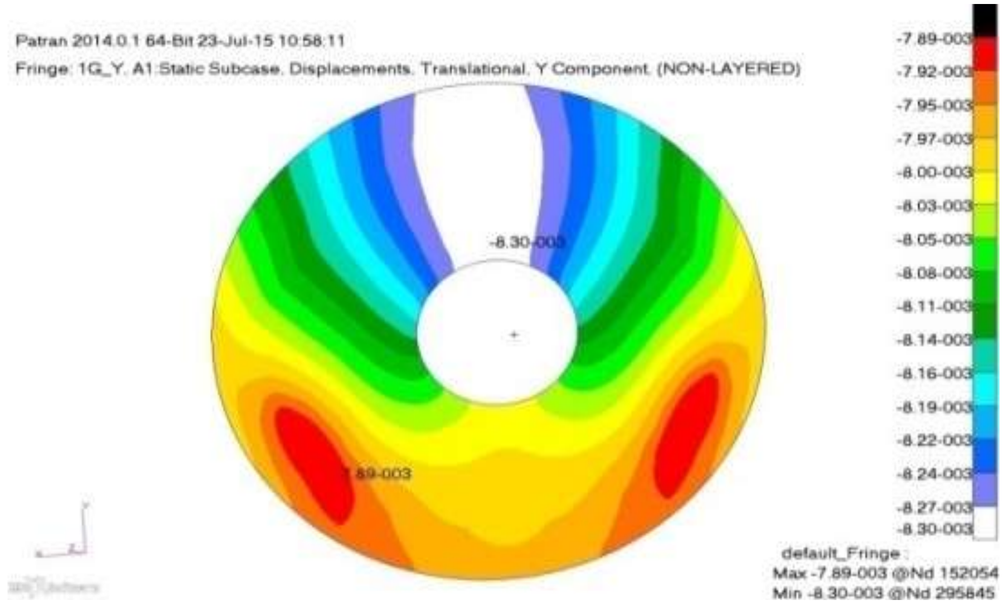

Figure 7: The front surface deflection of the Mirror due to 1g load in Y direction.

The front surface deflection of the mirror is analyzed by fixing the base of the MFD and 1g Gravity load was applied to the mirror. The figure 7 shows the front surface deflection of the mirror.

6.2. MODAL ANALYSIS

The results of the modal analysis are tabulated in the table below for both the Free-Free and Clamped conditions.

Table 6.2: Modal analysis Results.

	Natural frequency	
Mode	**Free- Free Eigen values**	**Clamped Eigen values**
Mode 1	0.00075015	175.6
Mode 2	0.000601895	175.6
Mode 3	0.00017591	219.68
Mode 4	0.000152	219.69
Mode 5	0.00013885	257.57
Mode 6	0.0003299	337.99
Mode 7	324.69	606.9
Mode 8	324.7	606.94
Mode 9	601.33	611.12
Mode 10	793.05	842.62

The fundamental Natural frequency modes both Free-Free and Clamped Natural Frequency is shown in the figure.8

Figure (a) Clamped Natural Frequency.

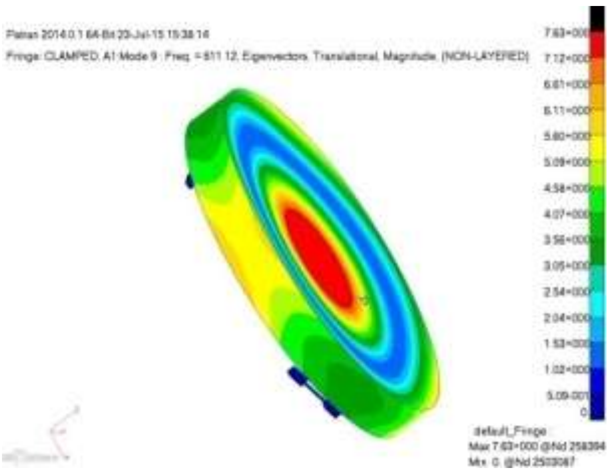

Figure (b) Free-Free Eigen Value

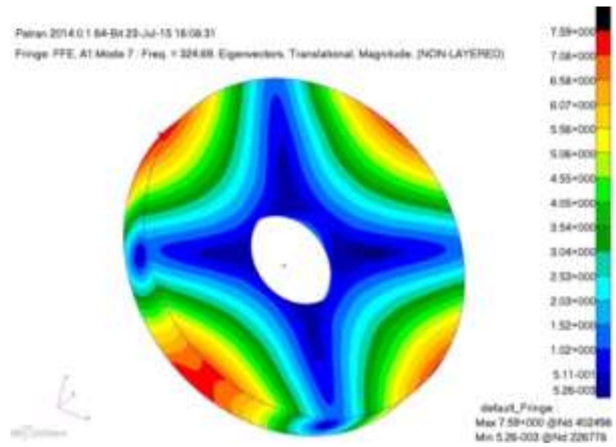

Figure 8: Modal Analysis (Natural Frequency of the Mirror).

6.3. STRUCTURAL ANALYSIS RESULTS

6.3.1. RESULTS OF QUASI STATIC ANALYSIS:

Table 6.3: Stresses in the Optical assembly

SL. No	Load cases	Locations	Von mises stresses in MPa	Max shear stresses in MPa
1	33g - X	Mirror	7.07	3.74
		MFD	254	129
		Glue	0.421	0.241
2	33g - Y	Mirror	6.22	3.27
		MFD	222	112
		Glue	0.364	0.210
3	33g - Z	Mirror	6.10	3.24
		MFD	74.7	37.8
		Glue	0.109	0.0546

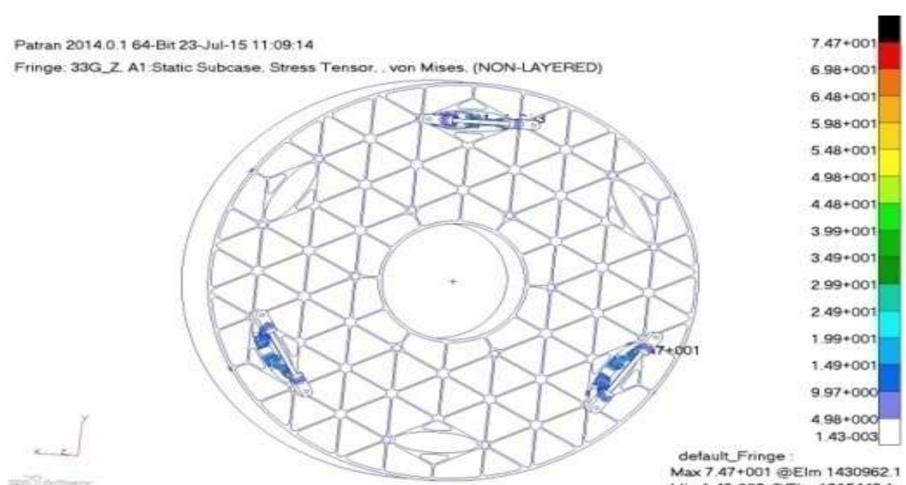

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Figure 9: Stress in the optical assembly

Figure 9 shows the stresses in the Optical assembly due to the inertia loading.

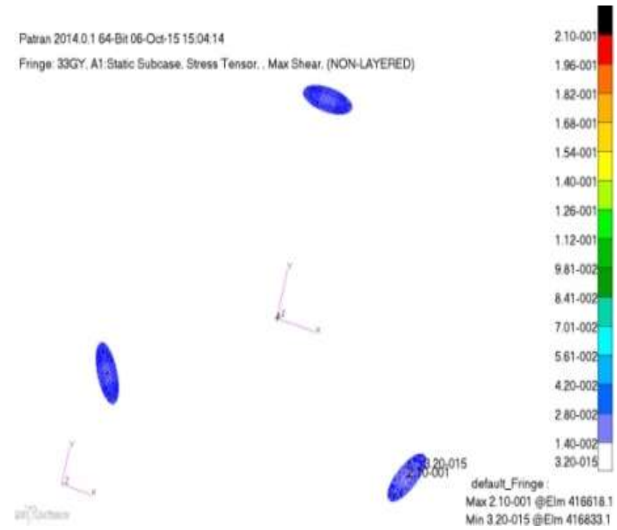

Figure.10a Stresses in MFD. **Figure.10b Stresses in Glue.**

Figure 10: Stresses in the MFD and Glue due to inertia loading.

6.3.2. RESULTS OF TEMPERATURE LOADS

Table 6.4 Temperature Stresses in the Optical assembly.

SL. No	Load cases	Locations	Von mises stresses in MPa	Max shear stresses in MPa
1	T plus 30°C	Mirror	0.214	0.133
		MFD	12	6.89
		Glue	3.57	1.80
2	T minus 30°C	Mirror	0.214	0.133
		MFD	12	6.89
		Glue	3.57	1.80

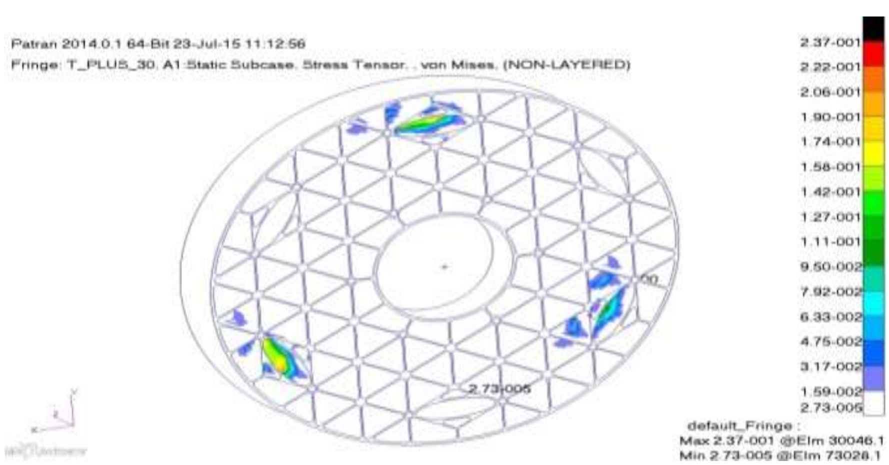

Figure 11: Temperature stress in the Optical assembly

Figure 11 shows the temperature stresses in the Optical assembly due to the temperature load of range -10°C to +50°C.

Figure 12a: Temperature Stresses in Glue.

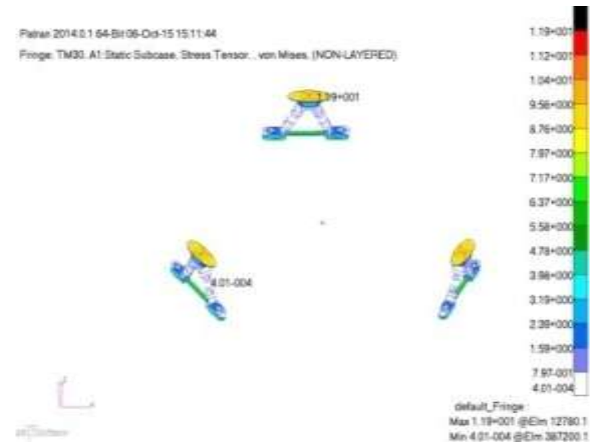

Figure 12b: Temperature Stresses in MFD.

Figure 12: Temperature Stresses in Glue and MFD.

CONCLUSION

Proper materials of the systems are selected. The different environmental loading conditions are analyzed. The Bi-pod flexural mirror fixation device is optimized to meet the requirements of the optical mirror. This Bi-Pod Flexure Mount can also be used for other light weighted optical mirrors in space applications. Hence Space researchers will get benefit from this.

SCOPE OF FUTURE WORK

In this project INVAR is used as the Mount material which is having the thermal coefficient of expansion as $1.26x10^{-6}$. In order to get better results, the materials having lesser CTE than this and stiffer can be used for future work.

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