

High bandwidth fast steering mirror

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ABSTRACT

A high bandwidth, gimbale, fast steering mirror (FSM) assembly has been designed and tested at the Lockheed Martin Space Systems Company (LMSSC) Advanced Technology Center (ATC). The design requirements were to gimbal a 5 cm diameter mirror about its reflective surface, and provide 1 KHz tip/tilt/piston control while maintaining $\lambda/900$ flatness of the mirror. The simple, yet very compact and rugged device also has manual tip/tilt/piston alignment capability. The off-the-shelf Piezo translators (PZT) actuators enable reliable and repeatable closed loop control. The adopted solution achieves a good mass balance and gimbale motion about the center of the mirror front surface. Special care was taken to insure the best positioning means with the mounted mirror assembly held kinematically in place. The manual adjusters have very good resolution, with the capability to be locked in place. All solutions were thoroughly modeled and analyzed. This paper covers the design, analysis, fabrication, assembly, and testing of this device. The FSM was designed for ground test only.

Keywords: Fast steering mirror, gimbale, high bandwidth

1. INTRODUCTION

There are many solutions available commercially for fast steering mirrors. Some are categorized as scanners. They provide a high-frequency angular beam deflection in one or two planes, the later being a combination of two one-axis scanners. The mirrors are relatively small in diameter to obtain the high bandwidth.

Steering mirrors capable of producing three-axis movements (tip, tilt, and piston) are available. With relatively large diameter mirrors, they are not able to reach high frequencies and also meet the natural frequency limit. Some of these constructions use soft spring constraints allowing the driving frequency to be much higher than the resonant frequency.

In most of these devices, the mirror is rear-surface bonded to the moving platform. The application for the device described in this paper calls for extremely high quality wavefront and beam manipulation. The stringent requirements for maintaining the high surface quality of this mirror require special mounting solutions that provide mechanical and thermal stress relief when assembled.

The need for very high resolution and repeatable movement, along with small stroke and closed-loop control, narrows the choice of actuation down to PZT. The requirements, coupled with the possibility of manual adjustment and gimbale motion, create a challenge in finding an off-the-shelf FSM, or more dauntingly, building a custom device.

The FSM presented in this paper successfully meets all requirements embodied in a surprisingly compact and simple construction within a reasonable price tag. More testing remains to be performed to confirm our claims.

2. REQUIREMENTS

The purpose of the FSM was to provide fast and precise control in tip, tilt, and piston of one leg in a Michelson interferometer (fig. 1). The optics are all 50.8 mm diameters. The particular application of the interferometer requires the ability to adjust the path lengths to all points on the mirror in unison, in steps up to 10 microns, and to do so with a settling time of about 1 millisecond. The steady state error cannot be greater than 0.1 nm in path length. In actuator-based coordinates, this translates to a resolution of 0.05 nm of motion along the mirror normal. The ability to sense and actuate at these levels represents a sizeable leap beyond the current state-of-the-art design, and the technologies needed to fulfill them are currently under development.

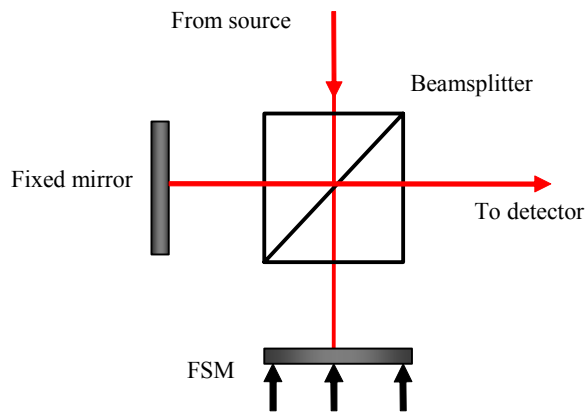


Figure 1 The FSM is the optical path-length adjustment mirror in the Michelson interferometer setup.

In addition to sensing and actuating at the sub-nm level, the mirror's surface figure distortion must be within similar tolerances. To ensure this, several design features are implemented. Among them are the mounting of the mirror on bipods to allow both the mirror and mirror mount to undergo thermal expansion without warping the mirror surface. The bipods themselves are designed such that the mirror-on-bipod vibration modes are all greater than 1000 Hz.

The actuators are mounted in a solid stainless steel base. There are no specific size or weight requirements on the base however; it must have no large vibrational modes below 2 kHz. Off-the-shelf alternatives were sought, but none had the desired combination of optic size, resolution, and response time. Table 1 provides a summary of the requirements and their values to which we designed the FSM.

Requirement	Value
Steady state piston error	0.1 nm
Steady state tip/tilt error	4 e-9 rad
Settling time	~1 ms
Maximum range	10 micron
Optic diameter	50.8 mm
Closed-loop bandwidth	~1 kHz
Surface distortion	< 1 nm
Ambient pressure	760 Torr
Ambient temperature	25°C

Table 1 The FSM was designed to achieve high bandwidth with minimal gimbal motion.

3. DESIGN DESCRIPTION

The FSM is designed to be used in a lab breadboard. The construction may be divided into three segments: moving parts, stationary parts, and the interface.

3.1 Moving Parts

The main component is a first-surface reflecting mirror: 50.8 mm in diameter and 12.7 mm thick. The 4/1 aspect ratio confers enough stiffness to obtain a high surface figure. It has a high reflecting silver coating optimized for a 45° angle of incidence. The clear aperture is an ellipse of 1.0 x .75 inches. The reflected wave front error is $\lambda/900$ RMS and $\lambda/250$ P-V @ 633 nm wavelength. The outside diameter is much larger than the clear aperture, allowing the manufacturing at the desired specifications and stress-free mounting of the optic. Figure 2 demonstrates the completed assembly of the moving parts.

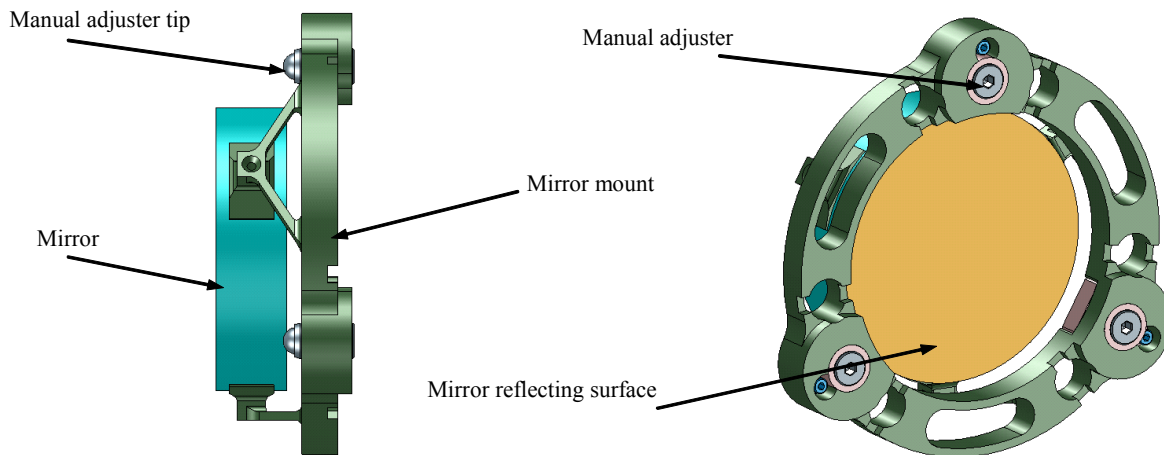


Figure 2: The first-surface reflecting mirror is mounted in such a way to allow stress-free mounting of the optic.

Careful consideration was given to the mount mass distribution, so that the assembly's center of gravity was located on the reflecting surface. Titanium is the chosen material for the mirror mount, for its specific stiffness. This design employs a monolithic mirror mount, consisting of a ring base with integral bipod flexures and bonding pads, with epoxy bond lines approximately .008 inches thick. Figure 3 depicts the mounted mirror and figure 4 is an image of the manufactured mirror mount, with a scale reference.

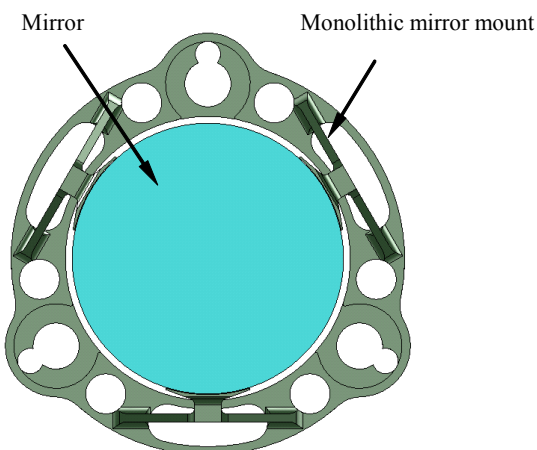


Figure 3: The mount mass distribution allows the assembly's center of gravity to be located on the reflecting surface.



Figure 4: The monolithic mirror mount is manufactured in Titanium to provide specific stiffness to the design of the mount.

Similar bipod designs, which are not monolithic, are often fastened to the ring base using threaded fasteners. This is adequate for environments where shock and vibration are minimal. In environments where shock and vibration are significant, a typical bolted friction interface can slip. This slip of the interface causes the bipod to impart some (small) moment into the optic, which will distort the mirror surface. For optics required to reflect high-quality wavefronts, this may be unacceptable. A monolithic design mirror mount (fig. 5) completely eliminates slippage by removing all interface joints.

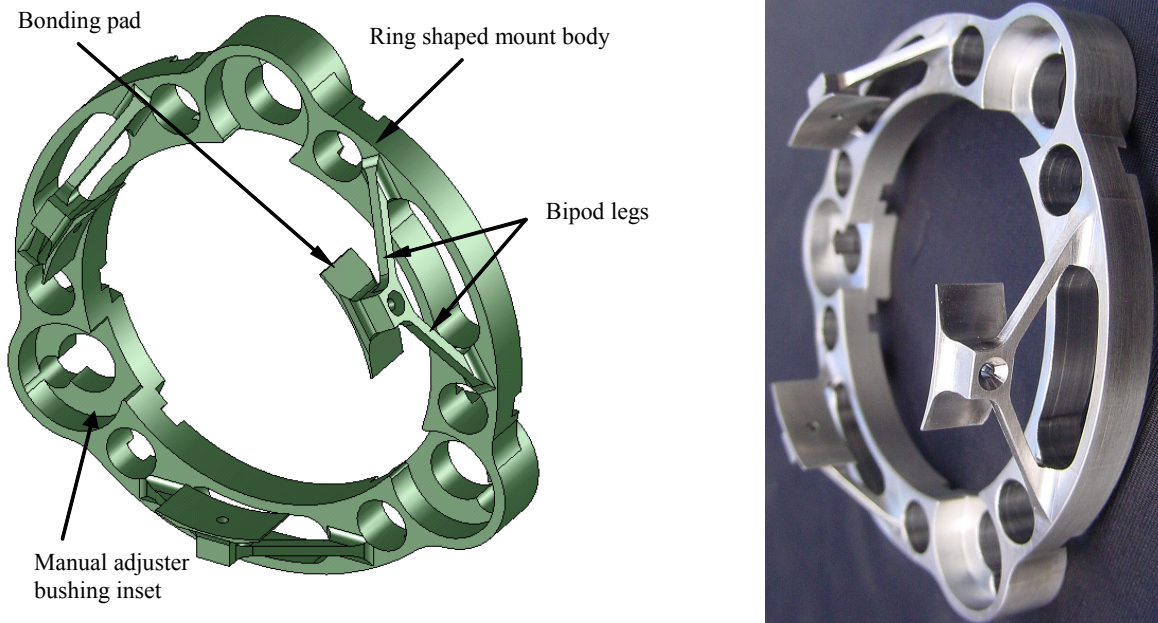


Figure 5: The monolithic design mirror mount completely eliminates slippage, allowing it to reflect high-quality wavefronts.

Figure 6 shows the detailed design of the mirror mount bipod flexure and bonding pad. The shape of the mirror bonding pad provides sufficient area to spread acceleration loads transferred to the optic while minimizing the stress caused by thermal soak. The bonding pad is thickest where it connects to the bipods, allowing efficient load transfer between bipod and bonding pad. The thickness of the bonding pad decreases toward the outer edges, which are nearly knife-like. By thinning the bonding pad to knife-like edges, the radial stiffness of the bonding pad is reduced, lowering the stress that is imparted to the optic during thermal soak.

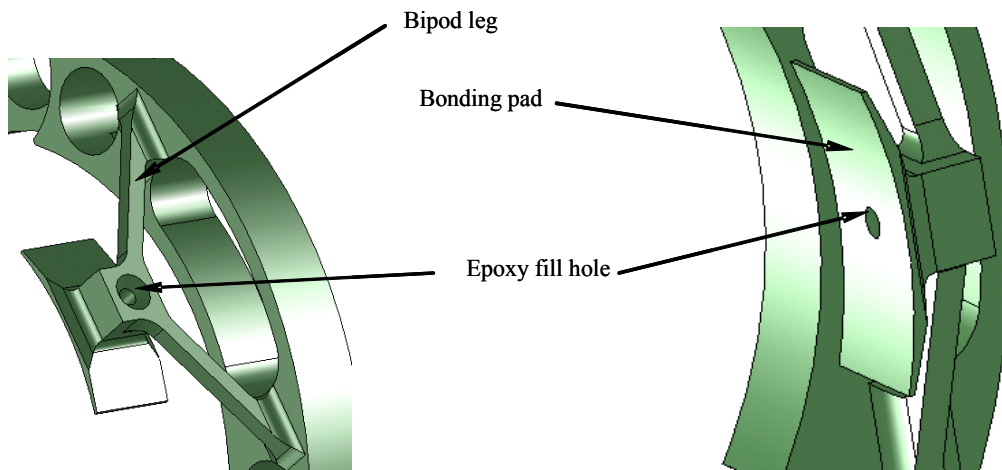


Figure 6: Our mirror mount bonding pads provide sufficient area to spread acceleration loads and minimize stress.

Thinning the bonding pad to knife-like edges also reduces the peel load on the adhesive. This effect is similar to the reduced stress concentrations that are observed in double scarf joints (Reference 1).

For optics in this size range, bonding pads are located at the optic edge, centered along the neutral plane of the optic, with the bonding pad width (direction along the optic centerline) being approximately 80% of the width of the optic. This minimizes the local edge effects, which would occur if the bonding pad extended all the way to the surface of the optic.

The bipod flexure mount has some significant advantages over other approaches. One approach often used is to mount the optic in a cell with compliant adhesive (typically room temperature vulcanizing [RTV] silicone rubber), located at the neutral plane of the optic (continuous ring or multiple “buttons”), which has a thickness defined by the one-dimensional athermalization equation. For many optic/mount combinations this button can be quite thick, often falling in the .030-.060 inch range. While this design is sufficient for many applications, it has been found to be dimensionally unstable in the several microradian regimes during temperature or pressure changes. By contrast, the dimensional stability is much improved due to the thinner epoxy bond line used in the bipod mount, combined with the increased stiffness and less moisture absorption of the epoxy, compared to RTV.

Another mounting approach is to use blade flexures instead of bipods. When blade flexures are used, they do not require the adhesive to be thick or compliant since the blade allows radial compliance. Thinner epoxy bond lines may be used, which increases the dimensional stability. However, while blade flexures are capable of providing radial compliance, they are also very efficient in transmitting moment (about the optic radial direction) into the optic. This is problematic if the base ring is deformed when it is fastened to the structure. Bipods are much less efficient in transmitting moment. This can also be explained in terms of kinematic constraints. Blade flexures are stiff in three degrees of freedom (DOF); optic axial and tangential translation plus rotation about the radial direction. Therefore, three blade flexures constrain nine DOF, which over constrains the optic. Each bipod, on the other hand, constrains two DOF. Thus, three bipods constrain six DOF, which is kinematic, or an exact constraint.

Figure 7 shows one of the three manual adjuster subassemblies located 120° apart in the ring-shaped body of the mirror mount. The manual adjusters insure rough manual positioning of the mounted mirror relative to the non-moving part of the FSM. Possible DOF are: tip, tilt, and piston.

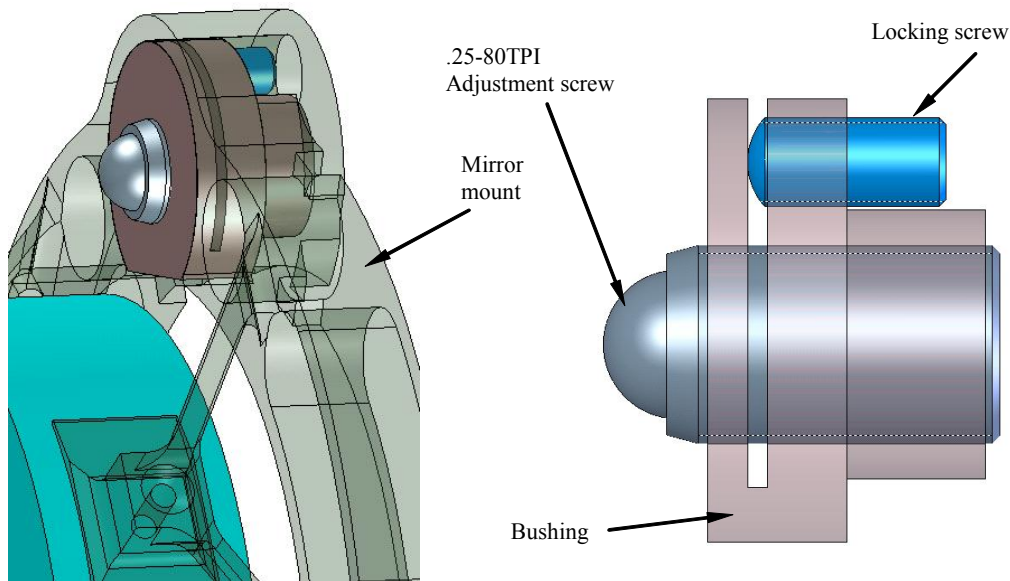


Figure 7: Manual adjusters insure rough manual positioning of the mirror relative to the non-moving parts.

The 80 threads per inch (TPI) adjustment screw can be locked in place due to the slotted flange of the brass bushing. The locking set screw threads into one half of the flange and when turned, pushes against the other half, increasing the gap and therefore tensioning the threads of the adjustment screw. This axial loading of the screw creates a minimal impact on its position.

3.2 Fixed parts

The main component from this category is the bracket. It is the structural backbone of the assembly, manufactured out of stainless steel. The bracket has an “L” shape with reinforcement gussets connecting the vertical and base plates conferring added stiffness in the Z direction. Figure 8 displays the completed assembly of the fixed parts.

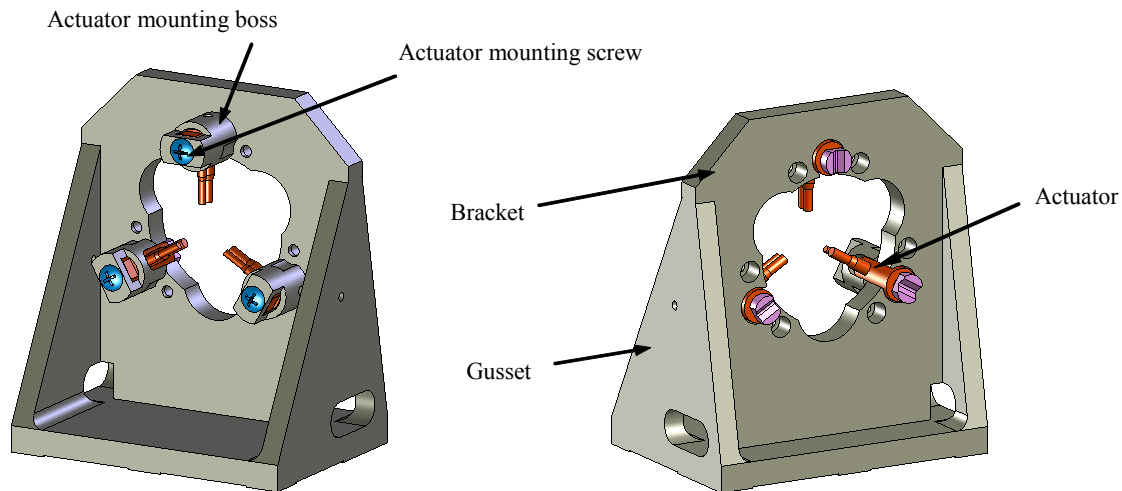


Figure 8: The stainless steel bracket provides good stability and added stiffness in Z direction.

The base of the actuators are mounted to the bracket with a single screw, each in three hollow bosses positioned 120° apart around the central opening of the vertical plate of the bracket. Figure 8 shows the opening that allows the mounted mirror to get as close as possible to the actuators to minimize the cantilevered portion, which sticks out from the boss. Each boss has a hole through which RTV can be injected in the clearance gap between the inside of the boss's cylindrical surface and the actuator body. The elastomer allows a more reliable mounting in the bracket, dampening the possible vibrations of the actuator body which would otherwise be cantilevered from its bottom.

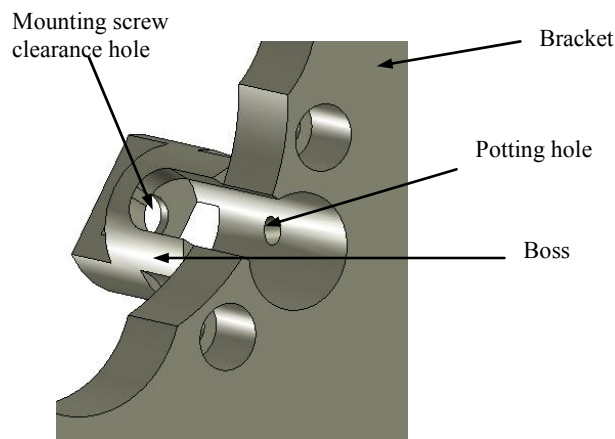


Figure 9: The actuator boss design minimizes cantilevered portion

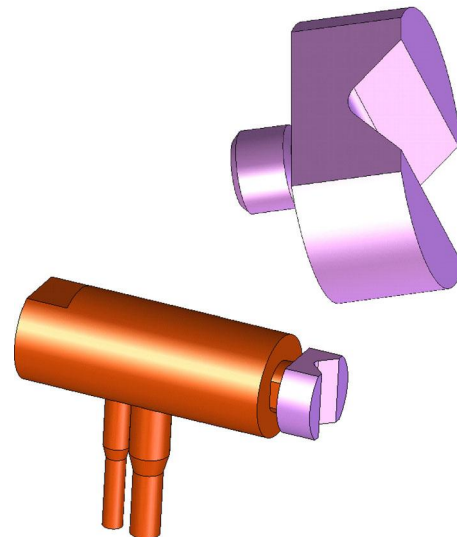


Figure 10: The actuators with “vee” groove attachments are part of the Kinematic mount, connecting the fixed and moving parts.

Figure 11 provides the specifications of off-the-shelf actuators: P-841.10 preloaded closed-loop LVPZT translators, manufactured by Physik Instrumente, as well as the standard top piece: P-840.95.

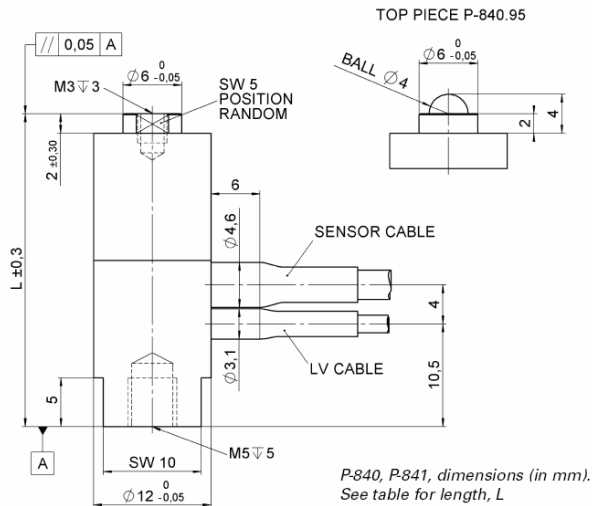


Figure 11 We design a custom attachment (fig. 10 (a)) to work with the off-the-shelf actuators from Physik Instrumente.

Technical Data

Models	Closed-Loop P-841.10	Units
Open-loop travel @ 0 to 100V	15	$\mu\text{m} \pm 20\%$
Closed-loop travel $\geq$	15	μm
* Integrated feedback sensor	SGS	
Closed-loop / open-loop ** resolution $<$	0.3 / 0.15	nm
** Static Large-signal stiffness*	57	N/ $\mu\text{m} \pm 20\%$
Push/pull force capacity	1000 / 50	N
Torque limit (at tip)	.035	Nm
Electrical capacitance	1.8	$\mu\text{F} \pm 20\%$
Dynamic operating current coefficient (DOCC)	15	$\mu\text{A} / (\text{Hz} \times \mu\text{m})$
Unloaded resonant frequency (f_0)	18	kHz $\pm 20\%$
Standard operating temperature range	-20 to +80	$^{\circ}\text{C}$
Voltage connection	VL	
Sensor connection	L	
Weight without cables	20	G $\pm 5\%$
Material case / end pieces	N-S	
Length L	32	Mm ± 0.3
Recommended Amplifier/Controller	C, D, G, H	
(codes explained p.6-46)		
* Closed-loop models are supplied with calibration data sheets;		
** Resolution of PZT actuators is not limited by friction or stiction.		
*** Noise equivalent motion with E-503 amplifier;		
*** Dynamic small-signal stiffness 30% higher		

Figure 10 shows that the ball tip attachment option (top piece P-840.95) was replaced by a custom-made “vee” groove attachment. When screwing them on the actuator, the grooves are aligned with the actuator cable by shimming. In this way, when the actuators are mounted in the bracket, the grooves are converging toward the axis of symmetry of the mirror. Figure 12 provides demonstration that when assembling the fixed and moving parts, the “vee” grooves of the actuator attachments and the ball tip of the manual adjustment screws come together forming a kinematic mount. The contact points lay in plane with the mirror surface, so that if tip/tilt motion is required, the mirror is gimbaled about the center of the reflecting surface. This feature, combined with the mass balancing about the same point, are designed to optimize the dynamic behavior of the mirror.

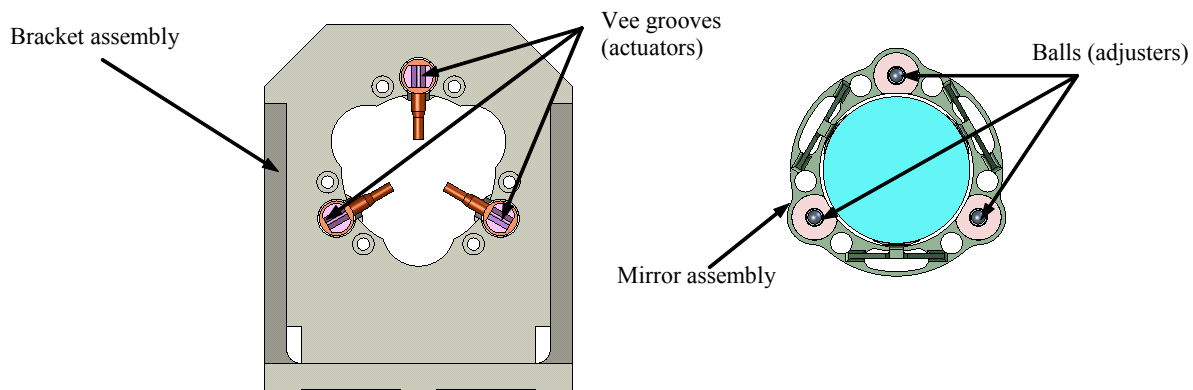


Figure 12: Control of the mirror position is optimized through the Kinematic mount of the mirror to the bracket.

3.3 Interface parts

There are a total of six extension springs, positioned in pairs, on each side of the three actuators. Figure 13 shows one end of the springs attach to dowel pins located in a groove of the mirror mount while the other end of the springs attach to a custom hook that is held to the bracket. The threaded shaft of the hook and the two hexagonal nuts allow for attachment and preload adjustment.

The springs are off-the-shelf components and were chosen to have a nominal preload of 38 lbf. This value of the preload and the mass of the moving parts allow a driving frequency of up to 2.7 kHz before losing contact between the grooves and the balls (the required driving frequency is 1 kHz.). Figure 14 provides schematics of the completed assembly of the FSM.

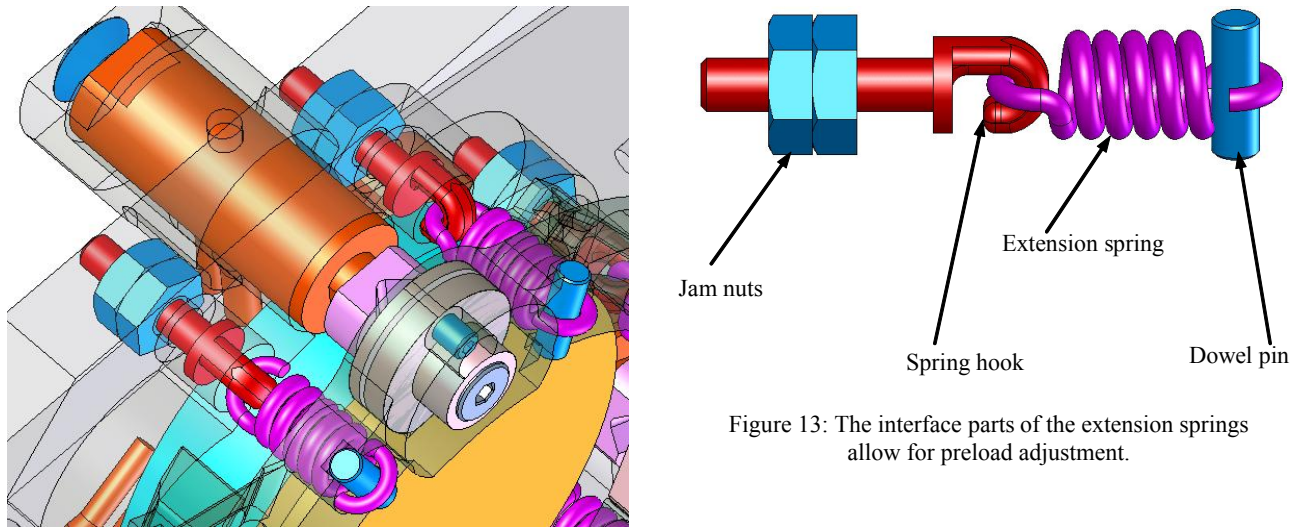


Figure 13: The interface parts of the extension springs allow for preload adjustment.

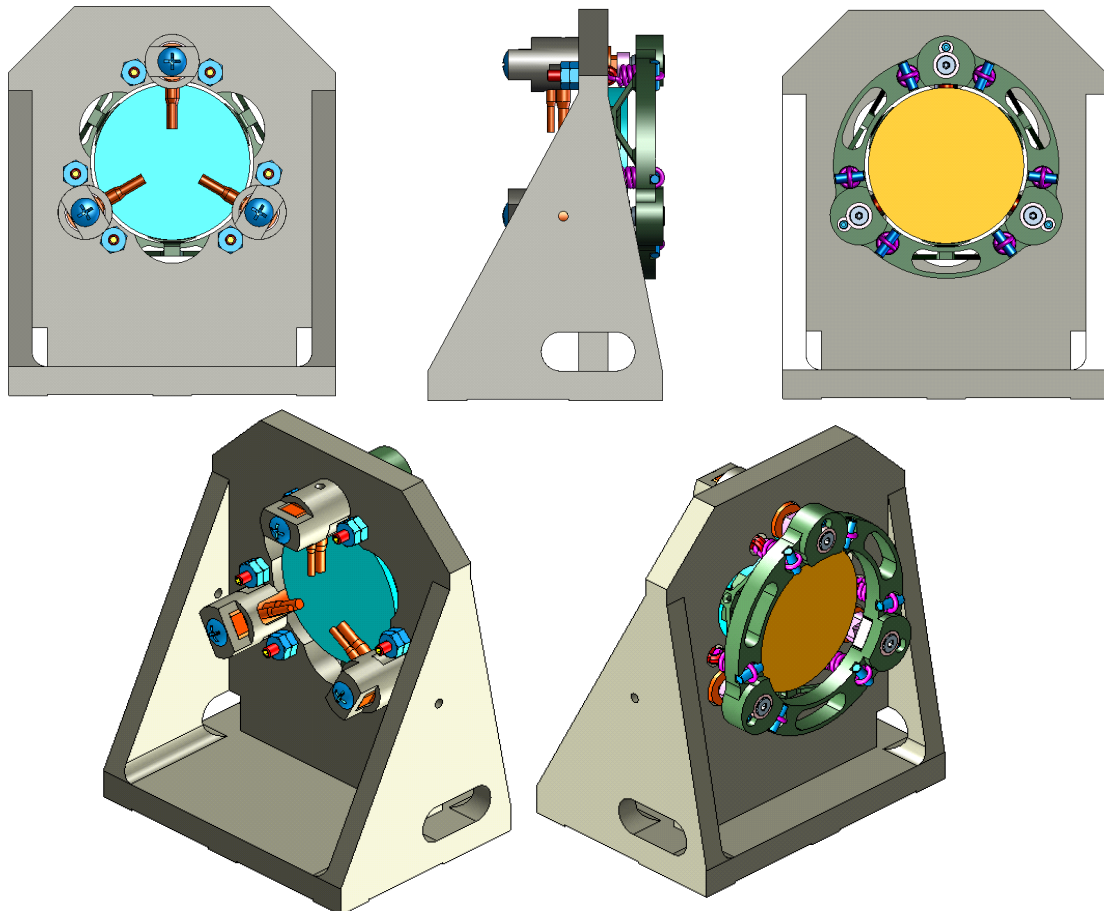


Figure 14: The completed FSM assembly allows for a driving frequency of up to 2.7 kHz, well above the required 1 kHz.

4. ANALYSIS

Stress, modal, and surface distortion analyses were performed on the FSM to verify it would meet structural and optical requirements.

4.1 Requirements

The analysis was driven by the following assumptions:

- 100g static design load, that should conservatively envelope loads from random vibration
- 1000 Hz minimum piston and tip/tilt modes required
- 2000 Hz minimum piston and tip/tilt modes goal
- Mirror surface distortions less than 0.84 nm RMS
- 38 lbf preload per spring, to prevent dynamic gapping
- ± 5 F temperature variation
- Safety factor of 2.0 for qualification by analysis.

4.2 Finite Element Model

The FSM was meshed in I-Deas Master Series version 9 and exported through a translator for MSC/NASTRAN version 2001. Analysis runs and model checks were performed with NX/NASTRAN version 2.0.

Since very small surface distortions had to be predicted, a finely meshed model (fig. 15) was required. The model fidelity chosen was more than adequate to predict normal modes since dynamic models can generally be coarser than stress models. Both the optic and the mounting ring were modeled with second order, 10-node tetrahedrons. The mesh on the optic was locally refined in the clear aperture to predict accurate surface distortions. The bipod legs were modeled with beam elements for simplicity, so their cross section could be changed easily, if needed.

The model was restrained at the actuator and spring interfaces to represent these kinematic DOF.

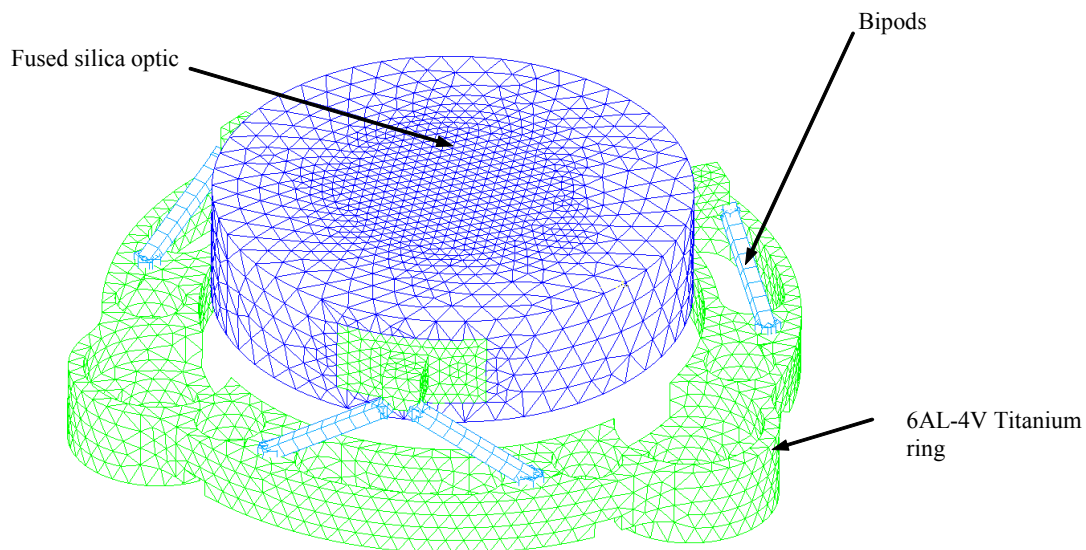


Figure 15: A selectively refined mesh in the finite element model was required to predict very small surface distortions.

One of the driving requirements for the FSM was to have elastic modes of the optic well above 1000 Hz so that the optic could be driven at 1000 Hz without elastic coupling. Table 2 provides the driving modes against which analysis was run.

A list of the primary modes and their effective modal mass are shown in Table 2. Figures 16 through 19 shows mode shapes for the three primary modes.

Table 2: Modes and Modal Effective Mass

Mode	Frequency (Hz)	Effective Mass						Description
		ux	uy	uz	rx	ry	rz	
1	1583	5%	71%	0%	5%	0%	0%	Decenter
2	1585	71%	5%	0%	0%	5%	0%	Decenter
3	2446	0%	0%	64%	0%	0%	0%	Piston
4	3899	0%	0%	0%	14%	13%	0%	Tip/Tilt

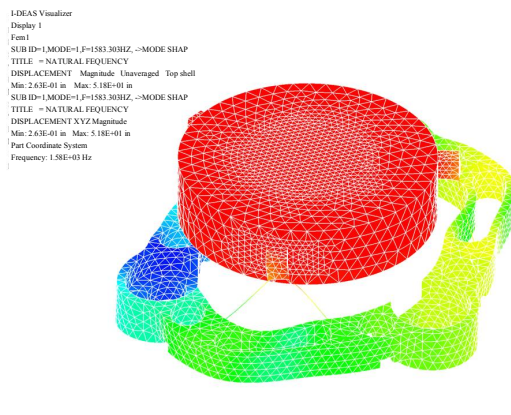


Figure 16: Mode 1 (1580 Hz) - Decenter

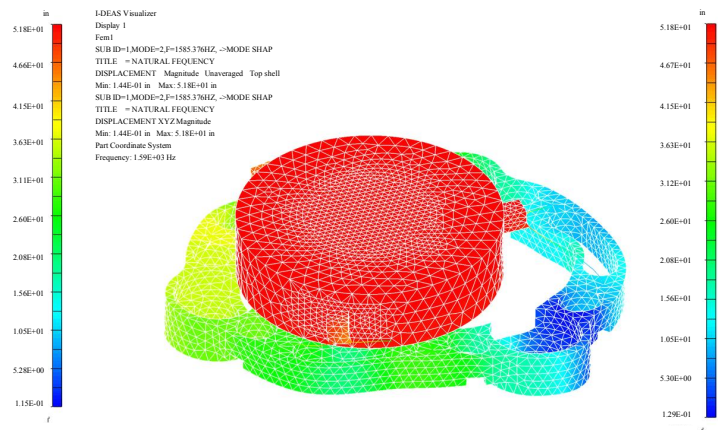


Figure 17: Mode 2 (1590 Hz) - Decenter

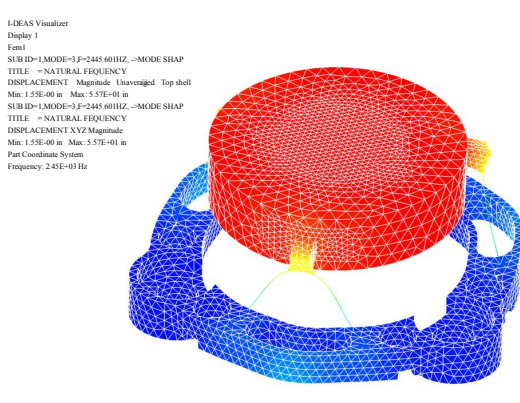


Figure 18: Mode 3 (2450 Hz) - Piston

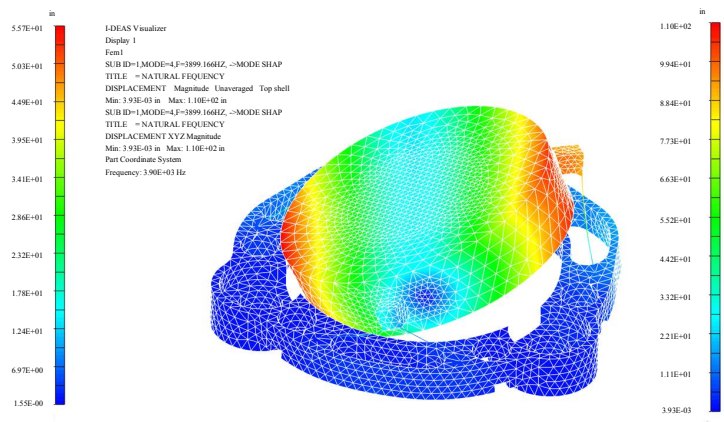


Figure 19: Mode 4 (3900 Hz) – Tip/Tilt

As expected, the stiffness is driven by the bipod and base ring stiffness. An initial iteration with a bipod leg cross section of 0.03” x 0.03” had to be increased to 0.06” x 0.06”. This somewhat compromised 1g sag, thermal distortion, and mechanical preload surface distortions on the mirror, but these distortions were still adequate as shown later in this paper. If higher stiffness were required, the base ring would be the next candidate to be stiffened. However, adding mass would increase the mass on the actuators, which is also undesirable.

4.3 Surface Distortion

Another critical requirement was to minimize surface distortion on the mirror.

The mirror clear aperture normal surface distortions were calculated for each load case and rigid body displacements and rotations, were subtracted if present. Table 3 summarizes the resulting surface distortions. Figures 20 through 24 shows all load-case surface distortions are well below the 0.84 nm RMS requirement.

Table 3: Surface Distortion Calculations

Load Case	Surface Distortion (nm RMS)
1g sag in X	0.02
1g sag in Y	0.02
1g sag in Z	0.11
38 lb preload at each spring	0.11
5F temperature delta	0.05

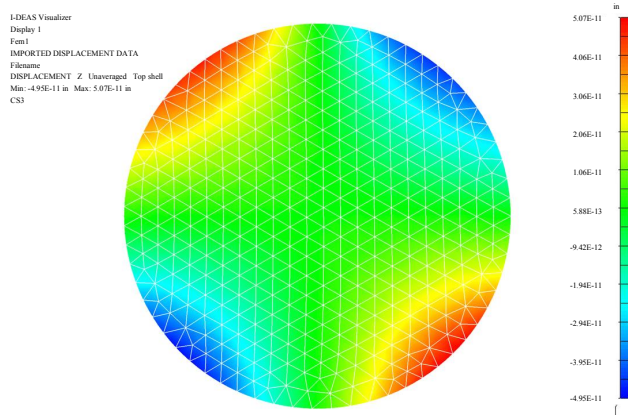


Figure 20: Clear Aperture Surface Distortion (nm) – 1g X

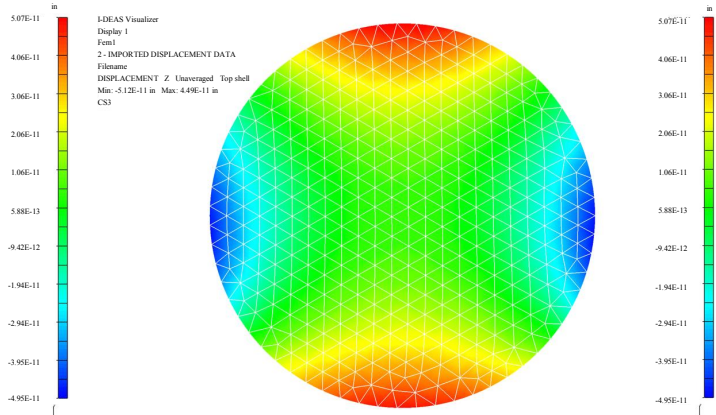


Figure 21: Clear Aperture Surface Distortion (nm) – 1g Y

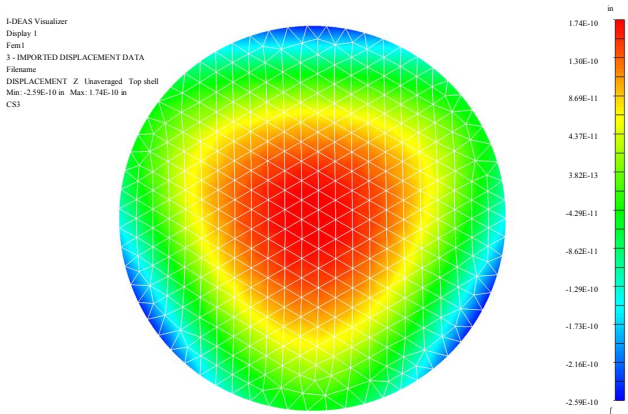


Figure 22: Clear Aperture Surface Distortion (nm) 1g Z

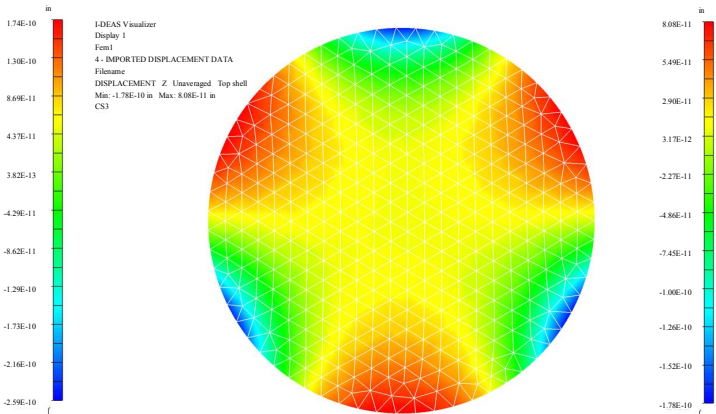


Figure 23: Clear Aperture Surface Distortion (nm) 5 deg F Temperature Delta

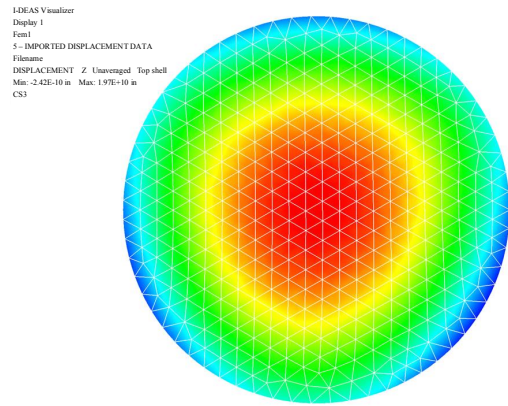


Figure 24: Clear Aperture Surface Distortion (nm) – 38 lb Preload

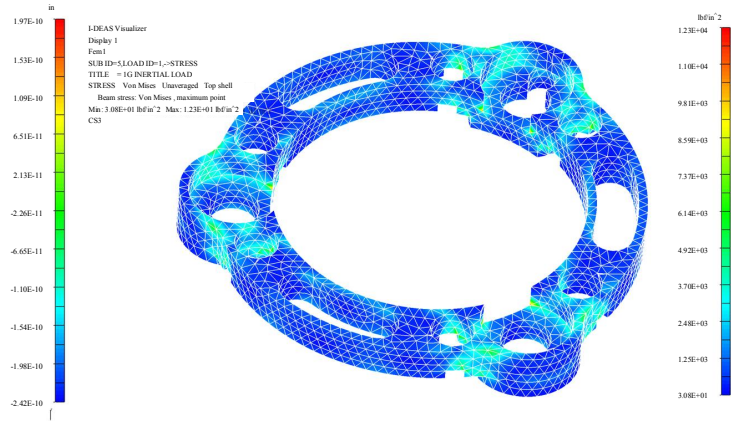


Figure 25: Ring Stress

4.4 Stress Analysis

The FSM were mostly stiffness driven, but there were localized areas where stresses were high. Overall, a safety factor of 2.0 for qualification by analysis was used. The margin of safety was then calculated as follows:

$$\text{Margin of Safety} = \left[\frac{\text{Allowable Load}}{(\text{Safety Factor} \times \text{Working Load})} \right] - 1$$

A margin of safety greater than zero was required. Table 4 summarizes the margins of safety.

Table 4: Margin of Safety Summary

Component	Material	Failure Mode	Allowable	Working Load	Safety Factor	Margin of Safety
Bipod	6-AL-4V Titanium	Compression	120 ksi	2 ksi	2	33.78
Bipod	6-AL-4V Titanium	Buckling	483 lbf	6 lb	2	37.61
Bond Pad	Epoxy Adhesive	Shear	2 ksi	0.3 ksi	2	2.50
Mirror	Fused Silica	Tensile Fracture	7800 ksi	TBD	10	TBD
Ring	6-AL-4V Titanium	Tension	120 ksi	55 ksi	2	0.09

Stresses in the optic were expected to be low, but a more accurate model of the mirror in the area of the bipod bonding pad was needed to accurately determine the stresses in this area.

Figure 25 demonstrates that the stresses in the ring near the actuator pockets were high, but acceptable. If possible, this area should be reinforced to reduce the stress concentration on future designs.

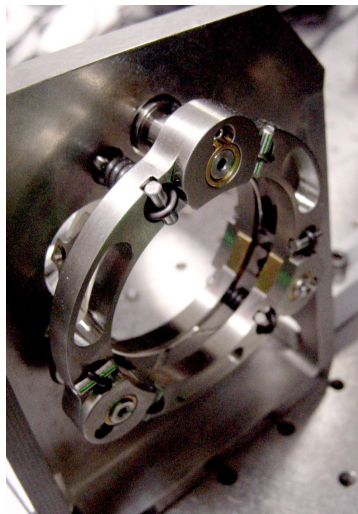
5. ASSEMBLY

The first step in assembling the FSM was to bond together the mirror and its mount. The mirror and the mount were positioned relative to each other so that the bonding pads were placed symmetrically to the neutral plane of the mirror blank. The adhesive (3M Scotch-Weld Epoxy Adhesive 2216B/A Gray) was injected through the fill holes of the pads and left to cure for 5 days. After the complete curing of the adhesive, the manual adjusters were assembled in the mirror mount to complete the moving parts subassembly.

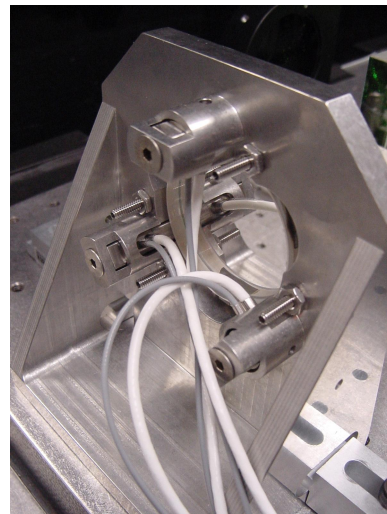
The “vee” groove attachments were screwed on the PZT actuators and the clocking was adjusted by proper shimming. The fixed parts subassembly was completed by mounting the actuators in the bracket.



Figure 26: Manufactured Parts



(Front view)



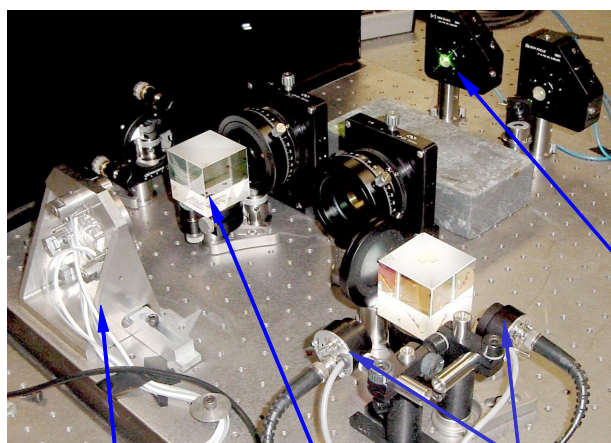
(Back view)

Figure 27: FSM Assembled and Mounted in the Test Setup

The FSM assembly was finalized by connecting the two subassemblies using the interface parts shown in figure 26 (pins, springs, spring hooks, and hexagonal nuts). The spring preload was adjusted by screwing the hexagonal nuts and the 80-TPI adjuster screw in and out. As displayed in figure 27 the relative position of mirror and bracket subassemblies should match the solid model design. This insures both the spring preload and the mass balancing.

6. TESTING RESULTS

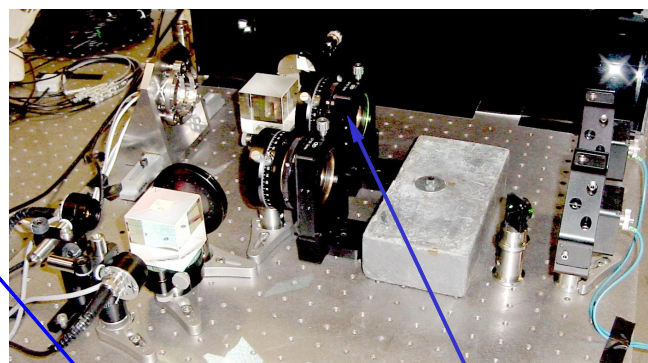
As often happens, we experienced poor timing with the funding of the project. It was temporarily out before we could significantly advance into the testing phase. We were able to perform a quick sanity check by powering up the actuators at the maximum stroke of $15\mu\text{m}$ and gradually increasing the driving frequency to 1 kHz, in increments of approximately 200Hz. The result was positive. This is a strictly qualitative assessment based on the smooth functioning of the device.



FSM

Beamsplitter

Detectors



Source

Polariser

Figure 28: The FSM is setup for testing once funding continues.

A thorough and exhaustive testing process is being devised and will resume after funding issues are solved. Figure 28 shows the work in progress on the testing setup, that is an interferometric metrology system.

7. LESSONS LEARNED

Some changes could be implemented to ease the assembly process. These involve mainly the interface parts. The dowel pins holding the springs at the mirror end tend to fall out the slots machined in the mirror mount, requiring great dexterity to put the assembly together. Modifying the mirror mount to contain the pins can easily solve this problem. More tool access room around the hexagonal nuts can also improve the preloading process of the springs.

8. SUMMARY

The challenge of an FSM with a closed-loop bandwidth of 1 kHz was a successful endeavor in and of itself. In attaining this goal, we also succeeded in the convergence of the gimbal point and the center of gravity at the exact same location on the surface of the mirror. This common position ensures minimal optical beam displacement and proper dynamic operation. Successfully manufacturing and building the device coupled with the partial testing makes this design promising. We anticipate that resuming further testing will validate the positive outlook that simulation gave us on the behavior of the mirror.

The mirror mount concept evolved through work on the Space Interferometry Mission (SIM) at the Jet Propulsion Laboratory (JPL). All design, assembly, and test work described in this publication was carried out at the Lockheed Martin Space Systems Company (LMSSC) Advanced Technology Center (ATC) by a team of engineers and scientists dedicated to developing aerospace technology and new concepts into reality through IRAD and government contracts.

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