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Design of the Fast Steering Secondary Mirror Assembly for the Giant Magellan Telescope

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ABSTRACT

The Fast-Steering secondary mirror (FSM) of Giant Magellan Telescope (GMT) consists of seven 1.1m diameter segments with effective diameter of 3.2m. Each of the seven segments is held by three axial supports and a central lateral support with a vacuum system to mirror gravity compensations. The mirror assembly has a tip-tilt feature capable of guiding to attenuate telescope wind shake and mount control jitter. Both on-axis and off-axis mirror segments were optimized under various design considerations using finite element analyses and optical performance analyses. An integrated model of the mirror cell assembly was developed including sub-assembly parts consisting of axial supports, lateral support, breakaway mechanism, seismic restraints, and pressure seal. In this paper for the optimized FSM mirror and mirror cell assembly, the design considerations are addressed, and performance prediction results are discussed in detail with respect to the specifications.

Keywords: extremely large telescope, Giant Magellan Telescope, Fast Steering Secondary Mirror, optics, opto-mechanics, mirror support, tip-tilt, optical performances

1. INTRODUCTION

For the design and development of the FSM mirror assembly, a conservative approach was implemented to minimize development efforts and fabrication risks. This approach ensures that a set of FSM segment assemblies become readily available on schedule for telescope commissioning and early operations in a seeing limited mode. We facilitated and updated legacy design features of the Magellan Telescope secondary mirror assembly for the FSM assembly, which mainly consists of a lightweight mirror with vacuum pressure and three axial piezo-electric actuators for fast tip tilt motions. The design of mirrors and sub-assembly systems were optimized using finite element analyses and optical calculations. The FSM design was presented at the System Level Preliminary Design Review (SLPDR) of the GMT in 2014^[6]. In addition, the Korea Astronomy and Space Science Institute (KASI) has adopted the FSM design and fabricated a prototype mirror and its assembly (FSMP) for performance and capability demonstrations^{[3][4][5]}. Recently, we have undertaken design and development efforts for the delta PDR (dPDR) study and the final dPDR report was completed^[9]. The main objective of the dPDR study is to complete the FSM design to a PDR level by incorporating the lessons learned during the FSMP development and suggestions from the planning study (Phase 0). In this paper, the design details and the results of the key sub-assemblies are addressed.

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An artesian view of the top end of the GMT telescope and the M2 system assembly are shown in Figure 1. Each FSM segment contains a tip-tilt capability used to provide fast guiding to correct telescope wind shake and mount control jitter to optimize seeing limited performance of the telescope. Currently, the FSM and the Adaptive Secondary Mirror (ASM) share a common interface with M2 Positioning System to control global segment positions and individual segment-to-segment positions. The FSM segment assembly is mounted to the M2 Segment Positioners via a kinematic mount interface (a set of 3 spherolinders). The FSM system will provide a low risk secondary assembly, serving as a commissioning secondary system and as a backup system when the ASM is not in operation.

Figure. 1. GMT FSM Subsystem on the telescope top end frame (top left), and the top end structure with the FSM assembly (top right). Each FSM assembly featured with a positioner mount interface unit - kinematic interfaces (bottom).

The FSM mirror assembly was fully developed which consists of mirror and mirror cell as shown in Figure 2. It includes the sub-assembly components – axial supports, lateral support assembly, seismic restrain system, tangent bar, vacuum seal, aperture plate, edge sensors and metrology fiducials.

Figure 2: FSM mirror assembly in a sectional view (left); Sub-assembly components mounted on the FSM mirror - hidden cell (right)

The FSM error budget in terms of arc-seconds of the encircled energy at an 80% diameter (EE80) is listed in Table 1. The error budget is based on the Natural Seeing Image Quality Error Budget (GMT-SE-REF-00145) for GMT optical performance and top-level allocations of error budget. The allocation of each of the error contributions typically includes fabrication errors in the optics and mirror support, misalignments and distortion due to the mounting of the optics in the mirror cell and dynamic motion of the mirror and thermal effects that contribute to image. Detailed error budget allocations are described in Lee et al. [10]. We performed the design optimization based on the Error Budget in Table 1. For GMT FSM design and development, we took a conservative engineering approach and implemented concepts established from the f/11 Secondary Mirror of the Magellan Telescope.

Table 1. FSM error budget.

2. FSM MIRROR SYSTEM

The fully integrated FSM system has a segmented meniscus concave mirror which has an equivalent diameter of 3.2m. It converts the beam reflected from the f/0.8 Primary Mirror into an f/8 Gregorian beam for the science instruments. The GMT reference document, GMT-REF-01559, describes the M2 coordinate system (M2 CSYS). The M2 CSYS origin is located exactly at the vertex of the M2 Mirror. M2 Mirror primary axis is the axis of revolution of the M2 Mirror, which is coincident with the reference optical axis. In its nominal position (Direct-Gregorian Narrow-Field (DGNF) configuration) the origin of the M2 CSYS (i.e. M2 Mirror vertex) is located 24162.4761 mm above the OSS CSYS origin (i.e. X, Y, Z= [0, 0, +24162.4761] in OSS CSYS) as defined in GMT Optical Design document, The 7 mirrors in M2-S(1-7) CSYS are shown below, and M2-S7 is identical to M2 CSYS.

The center segment (on-axis segment, S7) has a physical diameter of 1058.3 mm with a clear aperture 1050.3 mm. Each of the off-axis segment (S1-S6) has a physical diameter of 1049.5 mm with a clear aperture of 1041.5 mm. The Off-Axis Distance (OAD) for all 6 segments is the same OAD value (1087.7400 mm) which is defined in the GMT-SE-DOC-00010, GMT Optical Design Document. All optical surface parameters (i.e. Radius of Curvature & conic constant) have been defined in the GMT-SE-DOC-00010. The Off-Axis Angle (θ_{m2}) is define as the optical normal of each mirror segment at the OAD. The Off- Axis Angle has been directly calculated from Optical Surface Equation in conjunction with parameters defined in the GMT Optical Design Document, this calculation is shown in the *GMT-DOC-01119*. The Off-Axis Angle is $\theta_{m2}=14.777498^\circ$.

2.1. Finite Element Mirror Modeling

A finite element (FE) model of each of the on-axis mirror and off-axis mirror with lateral flexure was created in NX software FE package. This FE model assumes a solid concave lightweight mirror with a diameter of 1066.3 mm (on-axis), or 1057.5 mm (off-axis), 120 mm thick, and a radius of curvature of 4163.901 mm with a conic constant of -

0.7169273 which were basically generated from optical design software Zemax™. A lateral flexure model was generated from prototype of lateral flexure consist of EDM cut with central diaphragm thickness of 0.4 mm connected at the CG location of the mirror. A local coordinate system in the FE model was assumed as follows: (1) the negative Z-axis corresponds to the line which connects the vertex of the primary mirror segment to the vertex of the secondary mirror segment which is conjugated to primary segment; (2) the positive X-axis corresponds to the telescope's mechanical elevation axis; (3) the positive Y-axis is defined by the right hand rule. This coordinates system is shown in Figure 3. For the mirror material, Zerodur™ was used with its Coefficient of Thermal Expansion (CTE) of Class-0, 20×10^{-9} m/m/°C. The boundary condition were assumed as fixed axial direction (Z direction of the global coordinates) with fixed translation (X, Y and Z direction) at the bolt holes of central diaphragm to connected lateral support systems shown in Figure 4.

Figure 3: M2 Coordinate systems, M2-S(1-7) CSYS

Figure 4. Typical FE models of FSM mirrors in Isometric front view and boundary conditions in the analysis

3.1. FSM On-Axis Mirror

In order to evaluate FSM mirror performances, we calculated self-weight distortions and thermal distortions of the FSM on-axis segment at a seeing limited operation. The mechanical deformation of the on-axis mirror was obtained

when key loads were applied to the model. The loads considered are axial gravity, lateral gravity, temperature soak of 1 C and temperature gradient. For the axial gravity case, gravity was applied parallel to the optical axis, and atmospheric pressure was applied on the mirror surfaces except for backside of the mirror. For the thermal distortions, unit temperature difference was applied the whole mirror model and linear temperature gradient along Z direction, indicates the top surface is 1°C cooler than the back surface. The results were listed in terms of raw surface errors (Raw), corrected surface errors (PTF) removed piston, tilts and focus from raw deformation, slope error and EE80. The results were summarized and listed in the Table 2.

As a typical axial gravity case, the optical surface evaluations were made with the vacuum pressure to accommodate the mirror mass. The corrected surface errors after removing piston, tilt and focus are shown in Figure 5. The results show a P-V surface error of 21.9 nm and RMS surface error of 3.0 nm with slope error of 0.105 urad and EE80 of 0.014 arcsec.

Table 2. FSM error budget.

On-axis mirror			Axial G	Axial G with P	Lateral G	Thermal Soak of 1C	Thermal Gradient Z
SFE	Raw [nm]	P-V	648.0	101.6	46.0	3.90	19.4
		RMS	151.0	28.3	4.6	0.50	5.4
	PTF [nm]	P-V	417.0	21.6	46.8	3.30	1.9
		RMS	112.0	2.7	3.6	0.35	0.3
Slope	Raw [urad]	X	0.985	0.171	0.076	0.005	0.033
		Y	0.986	0.171	0.122	0.005	0.033
	PTF [urad]	X	0.733	0.105	0.076	0.005	0.003
		Y	0.734	0.105	0.122	0.005	0.003
Spot Diagram EE80	Raw [arcsec]		0.169	0.033	0.013	0.0005	0.006
	PTF [arcsec]		0.132	0.013	0.008	0.0005	0.0004

Figure 5. For an on-axis segment: Image quality analysis of the axial gravity print-through. the optical surface map of the gravity support print-through (3.0 nm RMS surface) with spot diagram in diameter (arc-seconds) and its encircled energy distribution. EE80 diameter shows 0.013 arc-seconds.

3.2. FSM Off-Axis Mirror

For an FSM off-axis mirror, the mechanical deformation of the mirror due to axial gravity was obtained by applying the gravitational load at a 14.78 degree angle to represent the actual orientation of the mirror. Atmospheric pressure was applied on the optical surface to balance the axial mirror gravity. In this off-axis mirror at zenith pointing, the mirror is deformed by the lateral component of its gravity. The optical surface errors obtained were PV of 561 nm and RSM of 138 nm, and the corrected surface errors with piston and tilts were PV of 27 nm and RSM of 2.7 nm. The lateral gravity with balancing pressure generated the optical surface error P-V of 62.6 nm and RMS of 8.4 nm, and the corrected surface error P-V of 53.8 nm and RMS of 7.7 nm with slope X of 0.107 urad and 0.150 urad, and EE80 of 0.015 arcsec as shown in Figure 6.

Figure 6. Off-axis mirror : Surface error under lateral gravity with pressure

Table 3. FSM error budget.

Off-axis mirror			Axial G	Axial G with P	Lateral G	Lateral G with P	Thermal Soak of 1C
SFE	Raw [nm]	P-V	917.6	167.5	260.0	62.6	4.2
		RMS	173.8	35.0	46.8	8.4	0.5
	PTF [nm]	P-V	530.0	68.2	176.5	53.8	3.6
		RMS	115.4	7.1	34.3	7.7	0.4
Slope	Raw [urad]	X	0.803	0.219	0.212	0.106	0.006
		Y	1.137	0.217	0.343	0.149	0.006
	PTF [urad]	X	0.700	0.132	0.210	0.107	0.006
		Y	0.765	0.143	0.285	0.150	0.006
Spot Diagram EE80	Raw [arcsec]		0.206	0.036	0.055	0.016	0.0005
	PTF [arcsec]		0.137	0.014	0.046	0.015	0.0005

2.2. FSM Mirror Sensitivity Analysis

To evaluate alignment sensitivity of the system several sensitivity cases have been conducted based on assumption of what-if cases, such as lateral flexure mis-location from CG location of the mirror, pressure error with axial and lateral forces, seal force variations. The optical surface errors from the sensitivity effects due to the different location from nominal position of lateral flexure (CG of mirror) under lateral gravity were calculated. The results show a maximum degradation of 4nm (after piston, tip, tilt, focus removed) by the location of lateral flexure with range of -0.3 to 0.3 mm. The optical surface errors from seal force change of 10N/m along the edge of the mirror are about EE80 of 0.003 arcsec (PTF); from unbalanced force effects in axial support (10N each) is about 0.002 arcsec (PTF), and in lateral support (10N) is also about 0.002 arcsec (PTF).

In addition, the temperature due to axial actuator and load cell has been calculated by heat transfer analysis and its thermal distortion was examined as well. Thermal deformation due to temperature gradient has been conducted and listed in former section. Another thermal distortion was considered during tip/tilt operations based on assumption of heat source of 0.2 W at axial support from load cell and PZT actuators. A simple heat transfer analysis with conduction only model were conducted under assumption of whole heat source transferred at three axial support location and all mirror surface except for back side surface were in natural convection of $5 \text{ W/m}^2 \text{ } ^\circ\text{C}$, at atmospheric condition with ambient temperature of $20 \text{ } ^\circ\text{C}$. The temperature distribution due to this thermal analysis shows the highest temperature of $20.9 \text{ } ^\circ\text{C}$. The optical performance shows the impact is negligible (P-V of 2.8 nm and corrected surface error P-V of 0.3 nm and RMS of 0.07 nm and EE80 of 0.0001 arcsec.).

2.2. FSM Mirror Natural Frequency

The natural frequencies were calculated and the first 20 characteristic mode shapes were examined. The fundamental mode was 567 Hz in duplicated astigmatic models. Focus mode is found at 920 Hz, and were followed by trefoil mode at 1226 Hz. The first 20 natural modes were listed and mode their shapes are shown in Figure 7. These bending mode shapes are very much similar to the low order Zernike modes. Segments have high stiffness and will have little effect on dynamic performance of the M2 system.

Figure 7. First 20 significant mirror bending mode shapes at a free-free boundary condition.

2.3. Dynamic response of FSM Mirror from tip/tilt motions

A dynamic mirror FE analysis was performed to estimate the responses to the tip/tilt motions. For the dynamic response analysis, the FSM tip/tilt motions were implemented based the GMTO document as follows: (1) the tip/tilt angle of ± 12 arcsec; (2) the tip/tilt operation bandwidth of 25 Hz; (3) the control shape function in a sinusoidal mode with a maximum stroke of $\pm 30 \text{ } \mu\text{m}$ from the tip/tilt motion on the edge of the mirror. The optical responses

to the tip/tilt motions were estimated. The results from tilt about x axis and tilt about y axis are the optical surface RMS of 23 nm in a strong astigmatic mode, and EE80 of 0.018 from the given tip/tilt motions. It is note that the effects are noticeable.

3. FSM MIRROR CELL ASSEMBLY

The FSM mirror support system contains three axial actuators with a tip-tilt capability and a lateral support flexure mounted at the center of the mirror. As shown in Figure 8, the mirror cell is pressurized to balance the mirror gravity during operation, and the balancing between gravity and pressure is to be achieved by pressure control, vacuum control and load cell sensors. The gravity force vector will be cancelled by the orthogonal sum of the Lateral support and the vacuum pressure force. This load distribution will vary with the zenith angle. Any mismatch in the proper vacuum pressure will applied a static load on the piezo actuators. Each of the Piezo actuators is mounted to the Mirror Cell through Load cells, these will provide feedback force to adjust the vacuum pressure. The vacuum pressure control loop will adjust the pressure of the vacuum chamber until the net sum of the three piezo actuator load cells is zero.

Figure 8. FSM Cell Assembly is as shown in a half model at a cut-away view without hexapod (left); a FSM cell control schematic layout, the mirror cell which is pressurized to balance the mirror gravity (right).

3.1. FSM cell Assembly design

The main goal of the mirror cell structure design is to protect the mirror and to provide appropriate stiffness against the static loads and for dynamic operations. The cell design was implemented with two cell configurations, the one for the on-axis FSM, and the other for the off-axis FSMs. The off-axis FSM cells are designed to be interchangeable with each other and can accommodate the left and right hand symmetry of the positioner and the interfaces. Figure 9 shows a single cell of the FSM assembly and an exploded view of the cell structural components. The mirror cell design, a weldment arrangement with thin walled aluminum plates made out of heat treated AL 6061-T6, provides a higher stiffness and compactness, and is favorable in cell fabrication. Stiffeners and reinforced mounting design at the support locations provide a strong structural integrity for direct load path. Such structural integrity would offer appropriate stiffness against the static loads and for dynamic operations. The FSM tip-tilt feature will accommodate the dynamic disturbances from wind vibrations and telescope jitters. The latest specification of the tip-tilt system requires a fast tip-tilt of ± 6 arcsecs at a maximum bandwidth of 25 Hz.

Figure 9. Single FSM segment cell with windscreen hidden (left), exploded view of FSM cell (right).

As shown in Figure 10, the solid model of the FSM Assembly consists of sub-assembly parts including the FSM mirror and Support System integrated in the mirror cell. The cell model shows sectional views with the following components: a) edge seal for vacuum system, b) axial support break away clamps, c) lateral diaphragm support, d) mirror segment, e) clear aperture mask, f) wind screen, g) mirror cell, h) interface to M2 positioner, i) lateral support guide rod, j) lateral support tube clamp, k) lateral support axial adjustment tube, l) load cell for axial support, and m) axial support with preload springs around perimeter of piezo electric actuator for fast tip/tilt.

Figure 10. Sub-assembly parts in sectional views of the mirror cell. Left: FSM support systems mounted in the mirror cell. Three axial supports are shown in dark blue, and lateral support mounted at the center of the FSM mirror; Right: features in the FSM Cell Assembly in sectional view with edge seal for vacuum system, clear aperture mask, wind screen, seismic restraint, and vee-groove for mirror installation (Jack pad shown as a redundancy as an intermediate design; however, it is removed for dPDR)

The mirror cell is pressurized to balance the mirror gravity during telescope operation. The gravity force parallel to the lateral support is balanced by differential pressure force on the mirror. Each of the three piezo actuators is mounted to the mirror cell through a load cell, which provides feedback reading of force to adjust the vacuum pressure. The balanced mirror gravity with the pressure at a zero load cell reading keeps the mirror in a floating state, where the gravity print-through effect on the optical surface becomes minimum. Seal position trade studies to determine the preferred location in two different cases; (1) edge seal located at the back of the mirror and (2) located at the front flange. This difference causes a different distribution of force on the mirror and a corresponding difference in the optical performance; EE80 of 0.010 arcsec for Case 1 and 0.030 for Case 2. As a preferable option for the metrology fiducials and edge sensors on the mirror, we selected Case 1 with edge seal located at the back of the mirror as shown in 11. In addition, we made trades for various the shapes of the windscreen. A two-part windscreen will provide benefits in assembling and integrations.

Figure 11. Vacuum seal location between the mirror and the mirror cell (left) and windscreen in two parts (right)

3.2. Axial supports and Breakaway system

The axial support assembly consists of readily available commercial parts. The axial support design is based on the Magellan secondary which has three degrees-of-freedom: tip, tilt, and piston (local z axis). Each axial support consists of the following key components: 1) a breakaway clamp assembly for seismic overload protection, 2) a PI P-845.30 PZT actuator, 3) two PI P-176.60 flexible tip flexures to accommodate tip-tilt motion, 4) an Interface 1010-AF-5K load cell, and 5), preload springs to keep the PZT actuator in compression. An off-the-shelf, multi-axis controller provides tip-tilt correction at moderate bandwidths. PI provides four types of PI P-845 piezo actuators with the technical data. From the tilt specifications, FSM actuator requires to have a travel range of 36 μ m, and a resolution of 60 nm. PI Model number P-845.30 was selected. Its travel range of 45 μ m meets the requirement and its 1 nm resolution is sufficient for the operation. Technical configuration of the selected P-845.30 is: Stroke : 45 mm, Static stiffness : 75 N/mm, Push force capacity : 3000 N, Weight : 132 g, Length : 83 mm.

The FSM Assembly has a feature to limit the force exerting mirror as a safety mechanism. The mechanism will response to the events when the external load are higher than the design preload in the mechanism. During unforeseen circumstances of excessive loads on the secondary mirrors, the preload force on the breakaway system of the axial support may be exceeded. In such cases, the axial loading is no longer fully resisted by the breakaway system, but also by the axial seismic restraints. The axial seismic restraints act to inhibit the motion of the mirror under high positive and negative axial motion, while still transferring stresses away from the mirror in order that, even under extreme conditions, the mirror survives. Figure 12 shows the axial actuator assembly with breakaway system and its exploded view. The breakaway system is preloaded by six preloaded outer springs and the piezo cap is pressed against the plunger by the preloaded center spring. This breakaway assembly acts as a rigid link between the mirror and the axial actuators as long as the transferred forced do not exceed the preloads of the springs. The total preload of the outer springs is approximately 600 N in 6 equally spaced arrangement. The stiffness of each outer spring (catalog item: DIE springs, D-1101) is 9.8 N/mm (58.8 N/mm total). Likewise the preload of the center spring is 600 N, part number of Dayton H62-125 D-10. The stiffness of the center spring is 61.5 N/mm.

Figure 12. Section view of FSM cell, (b) axial actuator mounted to cell and breakaway system, and (c) axial support breakaway system

3.3. Lateral support system

During the telescope operation, the lateral components of external forces and lateral gravity are to be taken by the lateral support. For on-axis mirror, as the telescope is pointing away from the Zenith, then the gravity of FSM should be taken by lateral support proportional to its zenith angle. Whereas, the off-axis mirror will have its gravity vector at 14.67 degrees from the local z-axis which represents the actual orientation of the mirror. FSM support systems are design to accommodate the lateral components solely by lateral support and axial components are solely by the axial supports with vacuum pressure. For decoupling the lateral load component from the axial gravity, lateral

support commonly features a flexure mechanism. The lateral support assembly consists of a lateral support flexure, lateral support guide rod, axial adjustment tube, and a clamp. The lateral flexure is mounted at the central pocket of the mirror. It is design to take loads only in the lateral direction, and is flexible in the axial direction. The lateral support guide rod is connected to the flexure at the bottom, and is bolted to a top plate at the other end. The guide rod is integrated into the lateral support axial adjustment tube with two plates – one at the bottom and the other at the top. The axial adjustment tube can move up and down to adjust the axial position of the flexure aligned to the center of gravity. The support clamp is used to secure the axial position of the lateral flexure. The adjustable tube provides the correct positioning of the lateral support so that it remains undeflected out of plane.

We have conducted a trade study and explored several design variations for the lateral flexure shape configuration using FE analysis. We selected the design shown in Figure 13 based on the merits between stiffness and thermal compliance, prior experiences, and lessons learned from existing telescope projects. The lateral flexure is modelled as bonded to the mirror using epoxy 3M-2216 with a bond-line thickness of 250 μm on 12 bonding pads in total (two pads for each EDM part). The lateral flexure has been analyzed to determine its performances and optimize the design. The analyses included static gravity loading, nonlinear behaviors, thermal effects, and buckling analysis. For the stiffness calculations, a 100N was applied in lateral direction and axial direction, respectively. Figure 14 (left) shows the lateral deformation with its lateral stiffness of 76.7 N/ μm . Likely in the same figure (right), the axial deformation is shown with its axial stiffness of 0.064 N/ μm . In addition, a torsional stiffness of 0.17N-m/rad was calculated.

Figure 13. Assembled view (left) and exploded view (right) of the lateral flexure.

Figure 14. Assembled view (left) and exploded view (right) of the lateral flexure.

A stability analysis of the flexure was performed. FE model contains the mirror and the flexure and a unit lateral load was applied to the flexure. For buckling, a critical force of 4700 N was calculated with a diaphragm thickness of 0.4 mm. The FE results indicate that the flexure design remains stable to 4700 N as shown in Figure below.

Figure 15. Stability analysis, deformed shape from Buckling load of 4700N

3.4. Tangent support

When dynamic stabilities of the lateral flexure FE model were explored, we found an insufficient rotational stability due to the diaphragm disk while the six radial flexures were stiff enough. This lack of torsional stiffness can be improved by introducing a tangent support. A recent specification requires a tip/tilt bandwidth of 25 Hz, we designed a Tangent bar for dynamic decoupling. The tangent support is mounted near the outer edge of the FSM mirror and connected to the cell on one side and to the mirror on the other side as shown in Figure 16. The tangent support constrains its local axial degree-of-freedom which elevate the torsional stiffness about the local axis. The design implemented a stack of two PI P-176.60 flexible tip flexures. Another option is a simple slender rod with a flexible hinge cut-out in both ends. This slender rod type tangent bar was used in Magellan M3 mirror. Two options were extensively investigated and the merits were evaluated. Nominal dimensions of the tangent bar in flexure type (V1) and in rod (V2) are shown in the Figure below.

Figure 16. Tangent bar mount, one end mounted on the cell and the other mounted on the back surface of the mirror; and Nominal dimensions of Tangent bars

For a trade study, a FE analysis was made assuming that one end is fixed and the other end is free to move which represents a relative motion between the mirror and the cell. The stiffness values were calculated in both flexure types. After a trade study of comparing with a slender rod type bar which was used for the Magellan M3 mirror, the design of a stack of two PI P-176.60 flexible tip flexures has been selected for FSM.

3.5. Seismic restraint system

To prevent excessive loading on the axial and lateral mirror supports during a seismic event or rough handling of the component, three seismic restraint systems (seismic bumps) are included in the design. The restraints are located in

three places equally spaced on the back side of the mirror opposite the axial support locations. This feature is designed for excessive loads on the mirrors, the preload force on the breakaway system of the axial support may be exceeded. In this case, the axial loading is no longer fully resisted by the breakaway system. Similar to the breakaway mount, the axial seismic restraint is mounted to both the mirror cell and one of the mirror chambers, and connected by a metal rod. Images for the FSM cell and seismic restraint system are shown in Figure 10. These three seismic restraints are for the maximum accelerations anticipated for Survival Level Earthquake (SLE) at top of the M2 top end frame. The maximum accelerations in the design are 4.2 g in X-axis and Y, and 1.6 g along Z when the telescope is in Z-axis. The FE analysis indicated that the maximum stress on the mirror is approximately 3.9 MPa which is a safety factor of 9 to the ultimate stress and a factor 3 to the allowable stress of Zerodur according to the standards applied to a GMT instrument, the GMT-Consortium Large Earth Finder (G-CLEF)^[16].

3.6. Integrated modeling of the FSM assembly

To evaluate the FSM design on a system level an integrated FE model is used. The integrated model allows to quantify these stresses in order to predict possible damage. In addition the integrated model is used to obtain the mode shapes and natural frequencies of the assembled mirror cell. In addition, we used the integrated model to predict temperature distributions due to heat dissipation of the piezo-elements, load cells and pneumatic valves as well as external heat sources. However the current analysis focused on stresses at the glass interface during seismic events and mirror surface deformation during regular operation with and without active tip-tilt actuation. Even though integrated models generally are lower fidelity, the integrated model of the FSM uses high resolution at the mirror interfaces to obtain accurate stress results as shown in the following sections.

An integrated model has been created including the mirror, the mirror cell, and key sub-assemblies for evaluation of the system performance. The main purpose of integrated modeling is to simulate scenarios in which the interaction between various components may be difficult to predict otherwise. Our analysis for the dPDR study has been focused on stresses at the glass interface during seismic events and mirror surface deformation during regular operation with and without active tip-tilt actuation. Integrated models are generally made at lower fidelity, but the modeling of FSM was done at high resolution at the mirror interfaces to obtain accurate stress results.

Total mass of each of the on-axis and off-axis FSM cell assembly is estimated approximately 170 kg based on a solid model. The estimated mass of 170kg does not include a reaction mass system for the tip-tilt mechanism which is under consideration. The entire mass of all seven FSM systems is approximately 1187kg. In order to achieve the design goals, static analyses with gravity induced deformation and thermal distortions were performed for each of the mirror cell assembly. The static deformations and stress levels are relatively small and favorably met the design goals.

Figure 17. First 4 natural models of the mirror cell assembly, the fundamental structural mode occurred at 44hz as a torsional mode; 2nd mode is translational mode of the cell (mode shapes are highly exaggerated for illustrative purpose).

For the seismic analysis, we use quasi static loads to the integrated model. The maximum principle stress in the glass was calculated at 6 MPa at the location of the lateral flexure in the lateral acceleration case which corresponds to a safety factor of 6 to the ultimate stress and a factor 2 to the allowable stress. The stress values in the components

other than the mirror are at least a factor of 5 below their yield stress limits. A modal frequency response simulation has been also performed to predict the mirror displacement and deformation during tip/tilt operation. In general, the results of deformation are proportional to the tip-tilt amplitude and to the square of the frequency and are consistent with those from the mirror only model discussed. As shown in Figure 17, the fundamental structural mode occurred at 75Hz which is a torsional mode of the mirror rotation around the lateral support in the cell. The second and third modes are in a translational mode at 87 hz which is directly associated with the lateral stiffness of the central flexure. The forth mode is in a piston motion at 106 hz. The results indicated that the cell has an excellent dynamic stiffness with respect to the tip-tilt motion at the goal of 75Hz (a rule of thumb – a factor 3 of the 25 hz tip/tilt operation).

3.7. FSM assembly Controls

Design of the control system has been revised from the previous PDR design^{[6][7]}. For the tip-tilt control, we have improved the controller for piezoelectric actuators to provide enhanced power for high frequency control and a simpler and more efficient control architecture. Control interface has also been changed to a direct EtherCAT connection by the new controller. Vacuum control scheme has been revised to overcome the limitations of the old design concept found during the FSMP tests such as a periodic variation in residual pressure and concerns of stability and maintainability. For dPDR, we employed a model-based design to enhance efficiencies and reduce risks. A preliminary Simulink model for the FSM tip/tilt control has been developed. The preliminary control model, for example, is based on hardware configuration of the FSMP with a proper arrangement of preload springs on axial actuators as shown in Figure 11. More details on the control system design including actuator force controls and vacuum pressure controls are described by Jeong et al. (2018)^[11] and Dribusch et al. (2018)^[13].

In addition, a control system design has been updated for along with FSM AIT plans and procedures - FSM subassembly interfaces, fabrication, assembly, and system integration by Han et al. (2018)^[15]. To evaluate the FSMP system design and to identify deficiencies in performance, we have tested tip-tilt performance of FSMP more extensively in a more reliable testing environment in four different inclination angles with a 30° increment at two sample tip/tilt motions (10 Hz and 20 Hz). Analysis and the test results by Kim et al. (2018)^[14] indicated that the design concept worked properly in general; however, potential challenges were identified in the PDR design such as non-linearity of the PZT actuator, noise in the controller and phase unbalance among the actuators. These aspects can be resolved by the design improvements.

Figure 18. Simulink Model for the Tip/Tilt Control System

4. LATERAL FLEXURE TESTS

FSM lateral support developed by the KASI team is a flexure with 6 EDM parts, and is mounted at the center hole of the FSM mirror. Two sets of prototype flexures were fabricated and physically tested. The goal of the lateral flexure test is to evaluate the stiffness values of the flexure. The input of the test system is manually controlled and the output is directly measured by multiple indicators and recorded. A dynamically insensitive interferometer, 4D Technology AccuFiz, was used to overcome environment effects. On the back of the gimbal mount, there are

components that consist of force controlling devices and measuring instruments. Six LVDT indicators are employed to measure the axial and lateral displacements of the mirror and flexure assembly.

Displacements of the flexure were measured under both axial and lateral loads. From the mechanical test, the displacement measurements of the flexure were compared with the nonlinear finite element analysis. The results indicate that the test measurements and FE analysis predictions are in good agreement. The difference is approximately at 16 % in the axial stiffness, and at 6 % in the lateral stiffness. As a consequence, the simulation results are reliable for estimating the stiffness of the lateral flexure. The flexure test validates the design of the lateral flexure. For the axial stiffness test setup, two LVDT indicators in axial direction and one in lateral direction are placed at the back plate as shown in Figure 19. The pressure is varied from 0 to 100 psi, but load cell readings were unstable at low pressures, likely due to friction in the cylinder. For the actual test, the minimum pressure has been changed to 20 psi, which corresponds to the applied force range from 0.2 to 60N. In addition, the LVDT measuring vertical displacement is removed and placed at the vertex of the mirror in order to detect the displacement of mirror during the axial tests.

Figure 19. Left: Lateral flexure bonded to a test mirror with an epoxy layer of 250 μm ; Right Optical test setup with 4D AccuFiz system.

For lateral tests, the test consists of a piston (Max. force applied 1000N), Precision Pressure Regulator, Six LVDT Displacement Reading devices were placed on designated locations as shown in Figure 8 to monitor the lateral displacement of the moving block and the mirror substrate. This setup has positional uncertainties of $\pm 5 \mu\text{m}$, angular motion uncertainties of 100 μrad over 2500 μm travel of the moving block.

Figure 20. LVDT #1 ~ #3 are located on the top of the block to measure vertical displacement and pitch. LVDT #4 and #5 contact the back surface to measure lateral displacement and yaw. LVDT #6 measures lateral movement of the mirror substrate that will be used to compensate rigid body motion.

For the axial stiffness estimation. A nonlinear loading step of 5 N was applied up to 60 N at the post. The result from non-linear static analysis shows the behavior of flexure is in a linear, and the maximum magnitude of displacement occurs at the mounting holes on the flexure blade (right). A plot represents the displacements with

respect to the forces as shown in Figure 21. The maximum displacement of 462 μm was found corresponding to the axial load of 60 N. The plot shows a linear fitting, and the slope of the fitted line shows compliance of 7.69 $\mu\text{m}/\text{N}$. This implies that the axial stiffness of the flexure is calculated as 0.13 $\text{N}/\mu\text{m}$.

Figure 21. FSM prototype mirror in grinding and polishing stage on a working cell. Mirror diameter is 1094mm (left). FSMP mirror in final polishing stage and testing on another working cell with a diameter of 1064mm (right).

Likely for lateral stiffness evaluation, a nonlinear analysis for lateral stiffness test was performed to obtain its mechanical response due to the applied lateral load. The model was fixed on the edge of the mirror at ± 30 degrees points. Moreover, the top surface of the center adapter was constrained in z-direction to inhibit axial displacement, and a lateral force of 1000N was applied on the negative y-direction. From the analysis, the maximum magnitude of y-displacement is 0.0129 mm at the center of the flexure blade. Using the slope of fitted line in Figure 22, the lateral stiffness of the flexure is found to be 77.5 $\text{N}/\mu\text{m}$.

Figure 22. FSM prototype mirror in grinding and polishing stage on a working cell. Mirror diameter is 1094mm (left). FSMP mirror in final polishing stage and testing on another working cell with a diameter of 1064mm (right).

5. SUMMARY

Extensive finite element analyses and optical calculations were conducted for the optimized Fast Steering Secondary Mirror assembly. The FSM performance predictions indicated that the current GMT FSM mirror and its support system adequately met the optical performance. Compliance Matrix for Key Requirements was evaluated in terms of arc-seconds of the encircled energy at an 80% diameter listed in Table 1. For the evaluation of the lateral flexure design, experimental test scheme was developed. Through the axial and lateral experiments of the lateral flexure, the displacements were measured and stiffness values were estimated. The experiment results were compared with those from nonlinear finite element analysis. These experiment results and FE simulations were implemented to obtain the stiffness of the newly designed flexure. The results indicate that the lateral flexure experiments are in good agreement with the finite element simulation. For the axial stiffness, the difference is approximately at 16 %. For the lateral stiffness, the difference is approximately at 6 %. As a consequence, the simulation results are reliable for estimating the stiffness of the lateral flexure. The flexure experiment test adequately validates the design of the lateral flexure. A high fidelity finite element model will be required to evaluate more extensive sensitivity cases, structural interaction effects, thermal mismatches, and other opto-mechanical effects. Trade studies between the

lateral diaphragm flexure and Magellan type configuration should be continued for the merits among the performance, cost, and risk.

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