Design of a Prototype Primary Mirror Segment Positioning Actuator for the Thirty Meter Telescope

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ABSTRACT

The Thirty Meter Telescope (TMT) is a collaborative project between the California Institute of Technology (CIT), the University of California (UC), the Association of Universities for Research in Astronomy (AURA), and the Association of Canadian Universities for Research in Astronomy (ACURA).

In order for the Thirty Meter Telescope (TMT) to achieve the required optical performance, each of its 738 primary mirror segments must be positioned relative to adjacent segments with nanometer-level accuracy. Three in plane degrees of freedom are controlled via a passive Segment Support Assembly which is described in another paper presented at this conference (paper 6273-45). The remaining three out of plane degrees of freedom, tip, tilt, and piston, are controlled via three actuators for each segment. Because of its size and the shear number of actuators, TMT will require an actuator design, departing from that used on the Keck telescopes, its successful predecessor. Sensitivity to wind loads and structural vibrations, the large dynamic range, low operating power, and extremely reliable operation, all achieved at an affordable unit cost, are the most demanding design requirements. This paper describes a concept that successfully meets the TMT requirements, along with analysis and performance predictions. The actuator concept is based on a prototype actuator developed for the California Extremely Large Telescope (CELT) project. It relies on techniques that achieve the required accuracy while providing a substantial amount of vibration attenuation and damping. A development plan consisting of a series of prototype actuators is envisioned to verify cost, reliability, and performance before mass production is initiated. The first prototype (P_1) of this development plan is now being built and should complete initial testing by the end of 2nd QTR 06.

Keywords: TMT, actuator, control system, primary mirror, M1CS, active optics, voice coil, Marjan Research

1. INTRODUCTION

1.1 TMT segmented primary mirror

The TMT[1] will provide a nine-fold increase in collecting area compared with today's largest ground-based optical telescopes. With its 30m aperture and the use of adaptive optics, the TMT will provide complementary science capability to the largest space telescope currently under development (i.e. the NASA James Webb space telescope). TMT's primary mirror will be composed of 738 segments, supported and controlled by an active and passive system to form the equivalent of a monolithic reflecting surface in much the same way as the 36-segment primary mirrors of the two Keck telescopes [2].

The 10 m Keck telescopes each use 108 actuators to position the 36 primary mirror segments. The Keck telescopes have been highly successful, from engineering as well as from a scientific point of view. They also have proven the concept of using actively-controlled segmented mirrors for building large optics. The Keck segment positioning actuators are reliable and able to meet demanding performance, operational, and environmental requirements.

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1.2 Key actuator challenges

The application of the Keck segmented mirror technology to the Thirty Meter Telescope (TMT) presents new challenges, particularly because of the more severe requirements on actuator size, weight, power consumption, cost and reliability. In 2001, a preliminary effort, undertaken by Marjan Research and associates, was sponsored by the CELT project to design and build a prototype (P_0) actuator [3, 4]. In addition to building a working prototype the CELT project sponsored an in depth trade study which compared various types of actuators.[5] The trade study was utilized to support the final design selection and evaluate performance. The resulting report concluded that while the Keck actuators represented proven technology and had a track record of reliable operation, a similar design would not be appropriate for the TMT M1 Control System actuators (M1CS_actuator) for several reasons.

First, TMT is a much larger telescope and thus more prone to optical performance degradation due to structural vibrations induced by wind, seismic or other disturbances. The analysis showed that a “rigid type” actuator (i.e., similar to the Keck segment actuators) is not suited to handle these vibrations. On a related note, the large size of the TMT results in larger mirror cell deformations compared to Keck and therefore requires actuators with nearly five times the stroke provided by the Keck actuators. The Keck actuators cannot easily be modified to achieve the larger stroke requirement.

Second, the positioning accuracy (< 5nm) required from the M1CS_actuators is extremely difficult to achieve with a mechanical geared ball/screw system because of friction, stiction, hysteresis, creep, and other non-linearities. The Keck actuators had in part solved this problem with the use of hydraulic bellow reducers, but at the risk of potential contamination of the optics from leaking hydraulic fluid, and increased cost and complexity. In addition, the Keck actuators do not provide a direct measurement of the position of the output shaft, so that the actual stroke must be inferred.

A total of around 2500 actuators (including spares) will be required for TMT, thus the size, weight, cost, reliability, and power consumption of an individual actuator becomes substantially more critical for the TMT M1CS_actuators compared with those built for the Keck. The Keck actuators clearly could not meet either the goals or requirements in any of these categories.

1.3 The soft actuator concept

The approach chosen for the P_0 and for the current P_1 design, utilizes a high bandwidth actuation device such as a voice coil motor for precise position control of the mirror segment; similar to the main component of most loud speakers. This type of motor has no inherent mechanical stiffness unless electrical power is provided to the voice coil. In order to make this type of motor work in a positioning system, a position sensor must be used to detect the position of the actuator output shaft relative to the actuator case. A local feedback control system is used to provide the exact amount of current in the voice coil so that the position sensor remains at the position commanded by the M1 Control System.

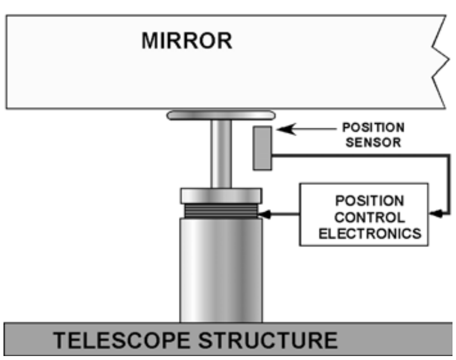

Figure 1. Voice coil driven soft actuator

The design concept of an actuator using a voice coil motor is a common approach to positioning electromechanical devices (Figure 1). It is relatively simple to implement, provides high bandwidth control when necessary, is lightweight, compact, relatively inexpensive, has very few moving parts, can move through a large mechanical range, is capable of large forces, and does not require lubrication. Many highly precise opto-mechanical systems, such as actively controlled mirrors used in telescopes[6, 7] and other optical applications, use voice coil motors for actuation. Voice coil driven actuators are referred to as “soft” actuators in contrast to the rigid positioning actuators such as the Keck segment actuators.

1.4 Gravity offloading

Because the telescope is constantly changing orientation during observations, the gravity force exerted by the mirror along the actuator axis is also changing. While a voice-coil device could potentially be large enough to handle the mirror weight and its relative variations, the corresponding power requirements and thermal effects would be prohibitive. The solution was to compensate for the weight changes though the action of a spring as shown in Figure 2. The auxiliary motor adjusts the compression of the spring so as to balance the gravity load of the mirror. The residual error is then easily handled by the voice-coil with a very low power impact.

1.5 Overview of the current actuator design

Based on the trade studies developed in the CELT actuator report, the experience of the authors, as well as others, with voice-coil based ("soft") actuator systems for large optics; and the positive results obtained with the P_0, the current P_1 design leverages similar proven concepts. A number of substantial design improvements are incorporated in P_1 which are aimed at further reducing weight, cost, and complexity while maintaining the high performance levels demanded by the TMT requirements.

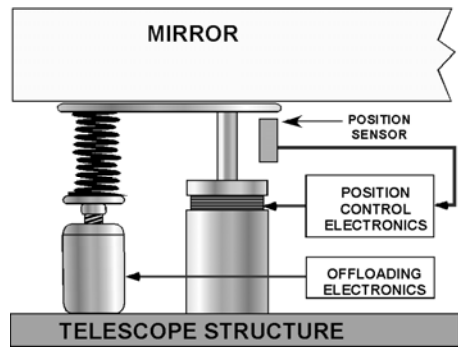

Figure 2. The soft actuator concept uses a combination of voice-coil and a motor driven spring in parallel to achieve minimum power dissipation and maximum positioning accuracy

In this design, the accurate positioning of the output shaft of the actuator is obtained by using a closed loop control system that senses the output motion of the actuator drive link with an extremely accurate interferometric position sensor. The output force is provided by a voice coil actuator directly coupled to the actuator output shaft.

In order to minimize power dissipation the voice coil actuator only has to operate on inertial loads rather than the gravitational force on the mirror segment; the weight of the mirror is continuously and automatically cancelled by a spring. The tension of this spring is adjusted at very low bandwidth by an electric motor/gear box combination in response to the average value of the current required by the voice coil actuator. When the average current in the voice coil actuator has been reduced to essentially zero, the motor-driven spring automatically shuts off.

The concept of "force offloading", of which this is an example, is a proven concept and has been used in aircraft for nearly a century. The trim tabs that the pilot adjusts to allow the airplane to fly "hands off" in pitch, yaw, and roll, is the aerodynamic analog of the motor-driven spring force-offload system for the TMT actuator.

In addition to providing active segment positioning through its axial motion, the actuator output shaft must also have substantial lateral stiffness in order to keep the coil centered in the magnetic field assembly and to stabilize the mirror segment against transverse loads generated by the weight of the whiffletree. In the P_0 design, a rather sophisticated system of custom-designed, specially manufactured flexures was used to provide both this support and as a means to partially counterweight the mirror. In the design of the P_1, the flexures selected to support the moving portions of the actuator are of a commercially available type that permit equivalent performance at reduced manufacturing costs.

2. ACTUATOR PROTOTYPE DESIGN

2.1 Requirements

The current TMT M1CS_ actuator is designed to satisfy the following requirements:

Performance Requirements:

Total stroke	> 4.3 mm
Accuracy	the greater of 5 nm or 1% of the requested move
Command rate	≥ 200 Hz
Slew rate R	≥ 50 m/s
Track rate	± 2000 nm/s
Tracking error	≤ 5 nm RMS
Small signal bandwidth (< 50 nm)	> 25 Hz
Position readout accuracy	≤ 5 µ
Axial load (performing)	0 to 750 N
Axial load (operating)	-200 to 750 N
Transverse load capacity	> 150 N
Transverse stiffness	> 4 N/µm (23,000 lbs/in)
Axial stiffness below 35 Hz	> 8 N/µm (57,000 lbs/in)
Weight	< 7 Kg
Local average power dissipation	< 2 W tracking with a < 1 Watt goal
Lifetime	> 50 Years

Operational conditions:

Orientation	0 to 105 degrees
Survivability	≤ 2.0 g shock
Operating	≤ 0.4 g shock
Altitude	0 to 5000 m
Temperature	-15 to 30 °C
Relative humidity	0 to 100 % condensing
MTBF	≥ 300,000 Hrs

2.2 Actuator general description

The TMT M1CS_ actuator is a "soft" actuator controlled by a (force-producing) voice-coil and an extremely accurate position sensor used for feedback (Figure 2). The static weight of the mirror is not supported by the voice-coil, which would require a large constant power expenditure, but is instead supported by a spring in parallel with the voice-coil, which offloads the quasi-static gravitational force. In order to accommodate the change in axial load as a function of zenith angle the spring is compressed or expanded by an auxiliary motor in such a way as to minimize the DC current in the coil. This system is referred to as the offloader or offloading system. The control system of the M1CS_actuator thus contains two loops: the position control loop which precisely controls the position of the actuator output shaft, and the comparatively low bandwidth offloading loop which minimizes the steady state current in the voice coil actuator.

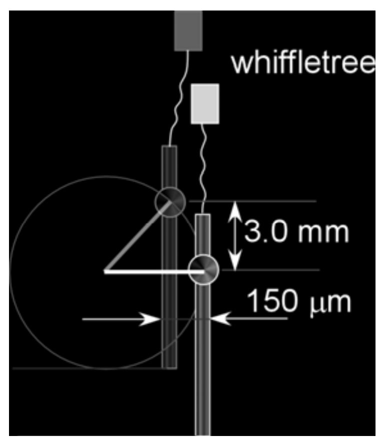

Figure 3. Simple linkage lateral motion

The implementation of the concept shown in Figure 2 requires three important practical additions. First the coil must be constrained to move in a quasi-linear motion along its axis so it will not contact the magnet and secondly the shaft attached to the coil must withstand transverse loads (up to 45 lbs) without deflection (23,000 lbs/in). The relatively long 4.3 mm stroke requirement, which could be easily increased to 6 mm in the future if required, eliminates the use of simple disk flexures. However, flexural pivots (such as the commercially available from C-Flex) can be incorporated in a linkage system to achieve the desired result.

Second, although it is feasible to use a direct ball/screw motor and a strong spring for offloading purposes; size, weight, reliability, cost and power can be significantly reduced though the use of a lever system.

Finally, when using simple linkages, the shaft does not move linearly throughout its stroke. Instead, linkage kinematics causes the shaft to move along a circular arc (Figure 3). Although this is a second order effect, the stroke of ±2.5 mm with a linkage length of 2 cm produces a lateral motion of the shaft of about 150 μm. More complicated linkages (e.g. a Watts linkage) can be used to alleviate this effect, but for cost and complexity reasons, this motion can easily be accommodated by using a flexural system to connect the actuator to the whiffletree.

2.3 Mechanical Design

A conceptual design meeting the requirements in Section 2.1 is shown in Figure 4. One of the most important features of this design is that it is absolutely friction/stiction free because all the motions are obtained through flexural elements. Even though the offloading mechanism includes a motor/gear box system, the motion is transmitted through a very soft spring and its effects are entirely compensated for by the control system. The entire mechanism is contained in a rectangular box of approximately 4 x 8 x 12 inches. The voice-coil is a commercially available off-the-shelf BEI LA-25-42, which has been used previously in the P_0 actuator. It has a hollow center that permits a shaft attached to the coil to go all the way through so that it can be attached on both sides of the magnet to the linkage system. It is especially important to support the shaft at two well-separated points in order to handle moments produced by lateral loads at the tip.

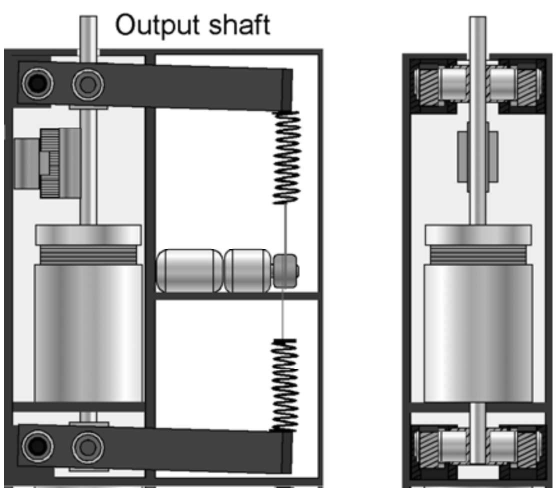

Figure 4. A "cartoon" of the P_1 actuator design concept including the use of a mechanical offload mechanism

The offloader springs are adjusted through a single, motor-driven spool and support the output shaft through two 7/1 reduction levers. This arrangement provides gravity offloading even in the case of a negative gravity vector (when the telescope is pointed at the horizon, some mirrors are actually looking downward by as much as 15 degrees). The flexural pivots are of the commercially available C-Flex type and are chosen for their angular deflection capability, transverse stiffness, and low operating stress levels in order to enhance lifetime.

A picture of a single-ended flexure is shown in Figure 5. The C-Flex flexures come in a wide range of sizes and capabilities. A baseline model for this design is the single flexure type H-20, which is a 5/8 inch diameter flexure with transverse load bearing capability of over 220 lbs and stiffness in all axes exceeding 25,000 lb/in. The H-20 has an angular motion capability of ±7.5 degrees. The net result is a composite stiffness of 33,000 lb/in that easily exceeds the requirement of 23,000 lbs/in as shown in Figure 6.

The offloading system uses a small stepper motor and an associated gearbox. The offloading springs are connected to the offloading motor mechanism via a flexible braided pre-stretched cable of the type developed for robotic applications. The cable is wound onto a spindle which is driven to either wind or unwind the cable. The rotation of the spindle therefore controls the length of the springs and thus the offloading force.

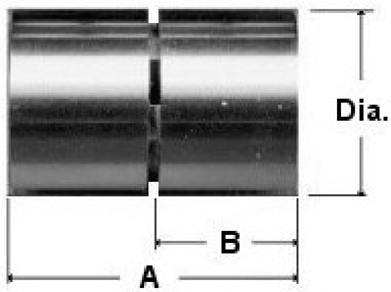

Figure 5. A typical C-flex bearing

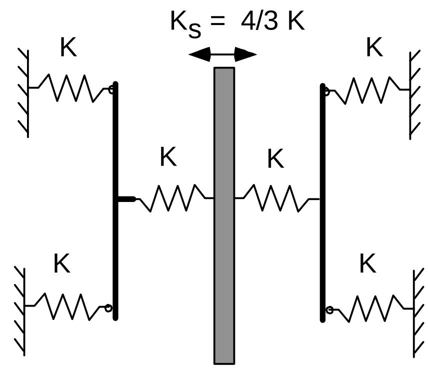

Figure 6. Equivalent lateral stiffness model of the actuator

The practical implementation of the concepts described above is shown in Figure 7. Two main changes are made to the original P_0 conceptual design: first, transfer pulleys are used to accommodate the actual length of the springs while minimizing the actuator envelope, and, second, an additional 5/1 gear and pinion system is added to the stepper/gear box assembly to achieve the desired 190/1 overall gear ratio. The mechanism is supported by a rectangular frame made of two standard 4x4 aluminum tubes bolted to each other. The overall envelope is shown in figure 8.

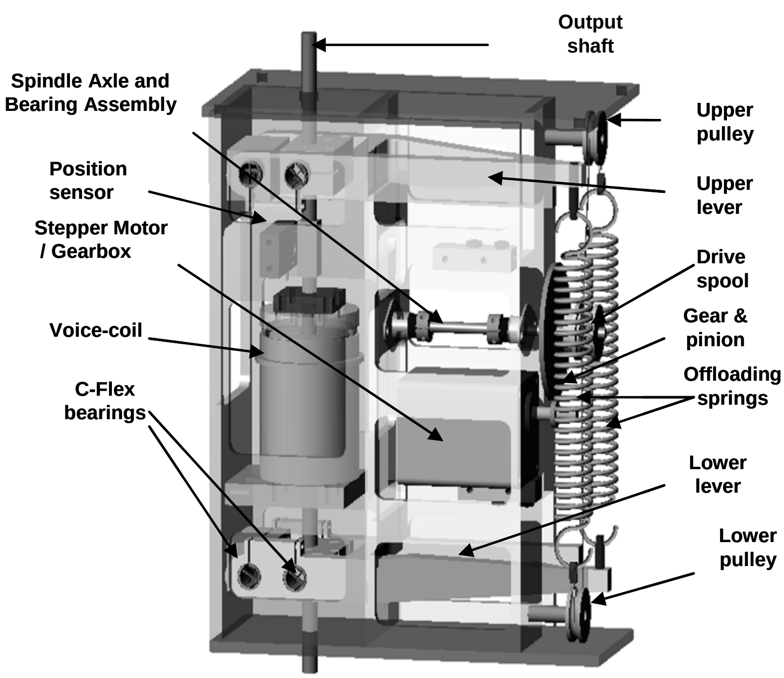

Figure 7. Actual implementation of the TMT soft actuator

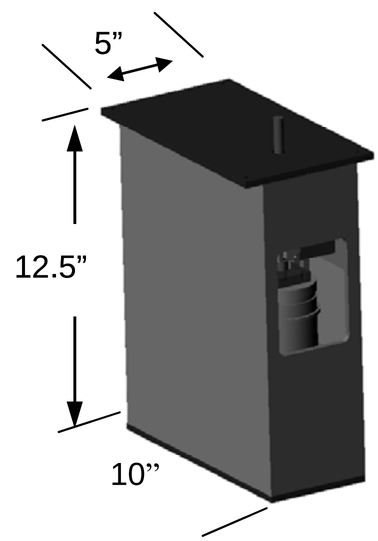

Figure 8. Mechanical envelope of the TMT soft actuator

2.4 Electronics Design

The electronic system supporting the M1CS_actuator is shown in Figure 9. The position sensor is a newly developed commercially available unit built by MicroE Systems (Mercury II). It is a reflective interferometric incremental position sensor, which operates by illuminating a grating with a laser diode to produce interference patterns as the grating is moved past the read head. A photosensitive detector detects the passage of the fringes and the signals are pre-processed in the sensor electronics chip that counts the fringes. The chip simultaneously interpolates the sinusoidaly varying intensity signal occurring between fringes and delivers a phase-quadrature signal that can then be decoded as a 32-bit word representing the change in position. The newly available model of the Mercury sensor has a 1.2 nm resolution.

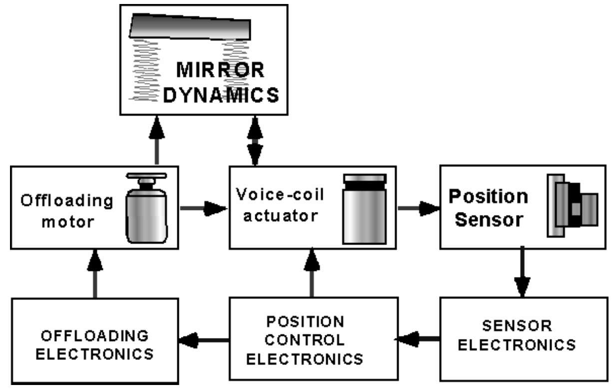

Figure 9. Overall view of the actuator and its controlling electronics. The offloading electronics are shown for clarity to be separate from the position control electronics. In the final design these will be combined into a single control unit.

The position control electronics contains essentially a phase/quadrature decoder for the sensor, a control algorithm running in a small digital computer that generates current commands to the voice-coil actuator and a low-noise power amplifier. The computer uses the value of the commanded current to the voice coil, to generate commands to the offloader motor so as to drive the DC value of the requested current to zero (Figure 10). This technique has been successfully implemented on the Gemini M2TS secondary mirror control system and works extremely well.

For clarity, the offloading electronics are presented here as separate from the control electronics, but in the final design it will be part of the same digital processing. The offloading control algorithm uses a logic often referred to as "dead-band-with-hysteresis". Its goal is to turn off the motor when the current in the coil is small, while simultaneously preventing a limit cycle phenomenon which often occurs with digital systems.

For the purpose of building a prototype, the implementation of electronic design must stay flexible enough so that changes can be made quickly. The proposed approach for the prototype will implement all the controls and testing capability in a small PC104 computer. Figure 11 shows a picture of the PC104 computer; the system's architecture is shown in Figure 12. In the final version, the sensor electronics may include FPGA or ASIC chips that will embed all the necessary logic for position control and offload operations.

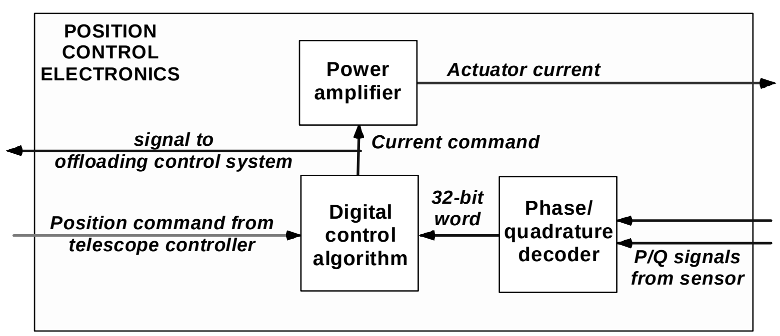

Figure 10. The offload control system drives the offloader so as to minimize the current and hence power dissipated in the voice coil actuator

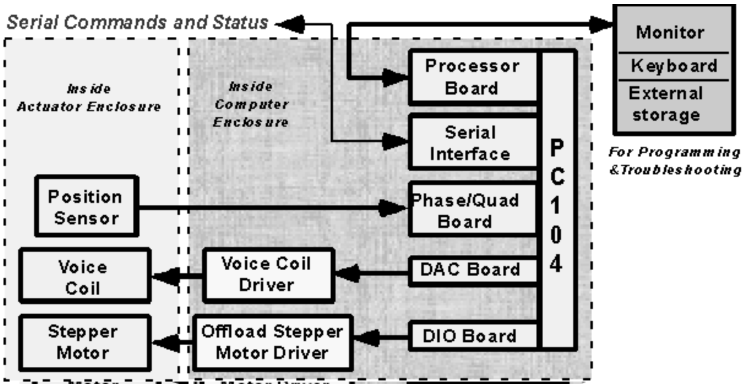

Figure 11. The offload control system drives the offloader so as to minimize the current and hence the power dissipated in the voice coil actuator

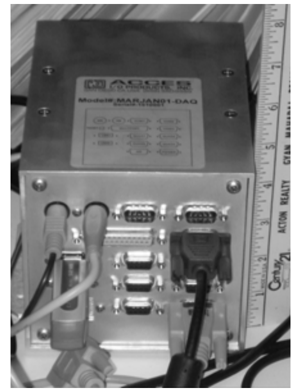

Figure 12. A picture of the PC104 computer that will be used to control the P_1 prototype actuator

2.5 Control System Design

The control system architecture is shown in Figure 13. It comprises a rolled-off PID controller for the high accuracy control of the actuator position and an offloader control loop. The system is designed with a 60 Hz closed-loop bandwidth. The unique damping capability of the "soft actuator" is exemplified in the transfer functions shown in Figure 14. When the mirror is disturbed by an external force, the control system not only maintains the desired position but also significantly reduces the resonant peak due to the mirror/whiffle tree spring/mass system. With a rigid actuator a strong resonance is observed (red curve). The motion of the mirror is reduced by an order of magnitude (blue curve) when the PID control system wrapped around the voice-coil actuator is turned on. The green curve is the motion of the actuator shaft. The difference between the blue and green curves is due to the whiffle tree which acts as a spring between the actuator shaft and the mirror.

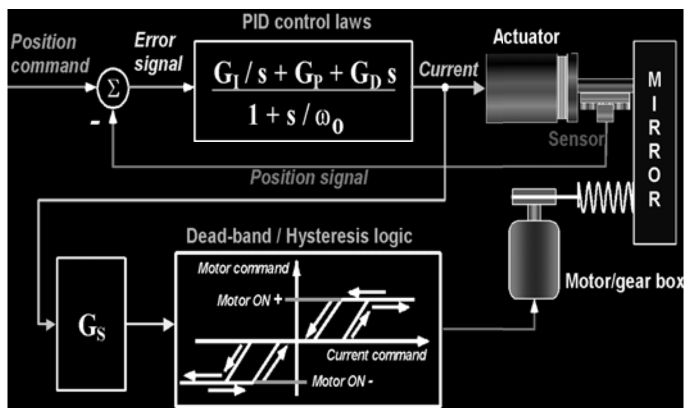

Figure 13. An illustrative control system schematic of the soft actuator including the dead band and hysteresis logic.

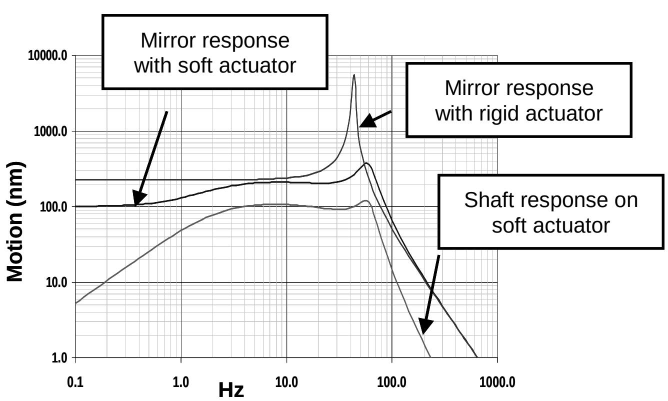

Figure 12. The disturbance rejection transfer functions for the hard actuator mirror, soft actuator mirror, and the soft actuator shaft. Notice how the soft actuator attenuates the resonance peak of the disturbance transfer function. The difference between the soft actuator mirror and shaft responses is due to the compliance of the mirror support whiffle trees.

3. EXPECTED PERFORMANCE

3.1 System characteristics

Mirror dynamics:

Mirror Mass	61.2 Kg
Whiffle tree mass	15.3 Kg
Actuator equivalent moving mass	7.0 Kg
Total dynamic mass on actuator shaft	22.3 Kg

Voice-coil characteristics:

Coil resistance	2.4 ohms
Electrical time constant	1.04 ms
Force constant	21.4 N/A
Power constant	13.8 $N/W^{1/2}$

Offloading system characteristics:

Spring constant	3258 N/m
Unloaded length	150 mm
Gear ratio	190/1
Shaft radius	10 mm
Lever ratio	7/1
Maximum motor speed	2 RPS (takes into account load and length changes)
Minimum motor turn angle	1.8 degrees

Control system:

Controller type	PID with roll-off filters
Sensor resolution	1.2 nm
Control bandwidth	60 Hz

3.2 Predicted values

The performance of the actuator is estimated using the formulas developed in the CELT report

Offloading system:

Change in spring length	41.7 mm	(over a -200 to 750 N load variation)
Spool rotation	238 degrees	(over a -200 to 750 N load variation)
Disturbance force at the actuator	0.040 N	(per step of stepper motor)
Resulting mirror position error	< 2 nm	(peak)
Maximum achievable shaft slew rate	9400 nm/s	(at 100 steps/s)

Position control:

Natural resonance frequency	7.3 Hz
Open-loop axial stiffness	0.16 N/μm
Resolution	1.2 nm
Stability	5 nm RMS
Closed loop stiffness at 35 Hz	>11.0 N/μm (with a 60Hz bandwidth)

Power dissipation due to tracking: (at 2 steps per second)

Voice coil	22 mW
Stepper	6 mW

Power dissipation due to wind

Voice coil	233 mW

4. OFFLOADER ACCELERATED LIFE TEST

The P1 actuator is required to have a life time of 50 years. The position control part of the actuator has no sliding or rotating parts, does not require lubrication, and relies only on flexural systems that can be essentially designed operate for infinite life time. The offloader mechanism however has a number of gears, bearings, rotating parts and cables, and thus it was decided to conduct an accelerated life test to ensure the reliability of the system. To that effect the offloader system was tested by running the stepper motor back and forth to the full range of spring extension. One complete cycle of the offloader is thus the physical equivalent of the telescope starting at the zenith, rotating down through 180 deg, until it is pointing straight down, then returning to the zenith. Of course, no operating telescope can ever have an elevation angle below the horizon, however from the perspective of operating the offloader in an accelerated life test mode it made sense to exercise the mechanism through the full range of possible mechanical motion.

During the 50 years of use of the TMT the offloader spool will make an estimated total of 75,000 full rotations. The system was thus tested by running the stepper motor at about 1300 times faster than normal operation (390 rpm). At this rate, it takes 50 s to achieve a full cycle (i.e. from 0 to 360 and back to zero spool rotation angle), thus 1728 cycles can be achieved every day and a 50 year simulated life test takes about 44 days.

After an equivalent 10 years of use, one of the cables broke. Another cable broke after an equivalent of 20 years of use. The failure was traced back to uneven wrapping on the spool causing excessive friction between two consecutive turns of the cable. The spool has been redesigned with a helical guiding groove and no further failure was observed.

The photos of figures 16 and 18 show the offloader assembly during the accelerated life test. For the test, the ends of the two levers were replaced by fixed tabs. Two limit switches placed at the top of the spur gear control the reversing of the motion at each end of the cycle. One of limit switches also triggers an electronic counter (the small black box in the left bottom corner). Figure 17 shows a close-up of the spool, cables and springs.

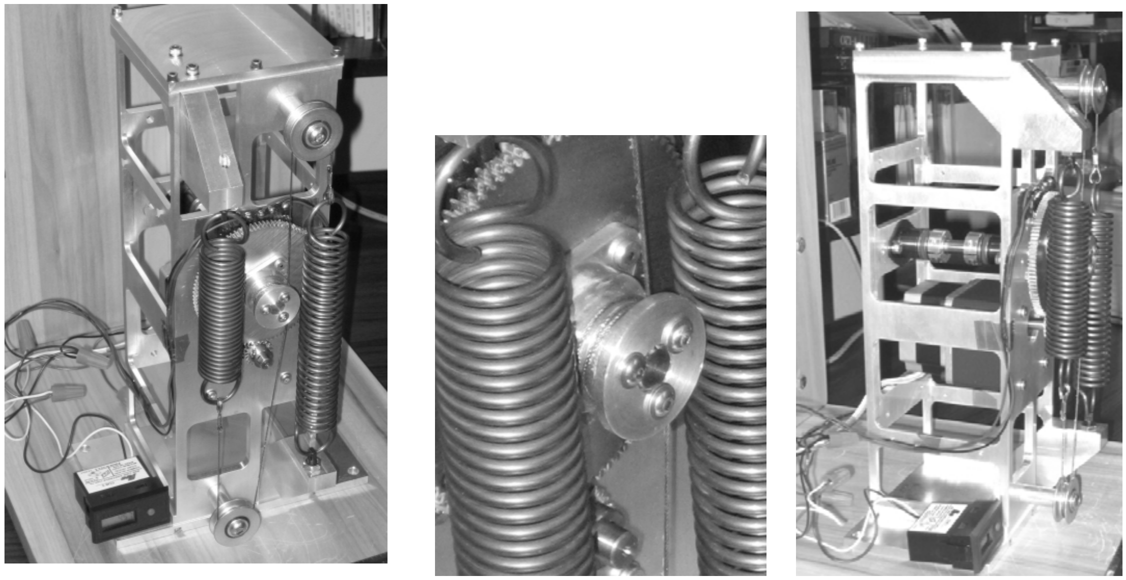

Figure 13. Offloader front view (left) and side views (right). The center picture is a close up of the spools, cables, and springs.

5. CONCLUSIONS

A P_1 prototype M1CS_actuator for the Thirty Meter Telescope is under development. The P_1 is expected to meet all the relevant M1CS performance requirements and provide additional capability by introducing damping in the structure.

The design is based upon the combination of a voice coil actuator that provides the fine positioning and an automatic offloading system. This technique provides the necessary force to balance the weight of the mirror at all elevations. A high level of performance is obtained by a control system that adjusts the current in the voice coil actuator based upon the readings from a very accurate interferometric position sensor. The control laws create equivalent stiffness and damping properties that are necessary to achieve high positioning accuracy even in the presence of external disturbances such as wind.

The mechanical design of positioning system contains no bearings, bushings or friction sensitive devices, and is entirely based on flexural elements. It utilizes a lever reduction system that significantly reduces the power consumption of the offloading system.

ACKNOWLEDGEMENT

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