# Design and preliminary test of a precision segment positioning actuator for the California Extremely Large Telescope primary mirror

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## ABSTRACT

In order for the California Extremely Large Telescope (CELT) to achieve the required optical performance, each of its 1000 primary mirror segments must be positioned relative to adjacent segments with nanometer-level accuracy. This can be accomplished using three actuators for each segment to actively control the segment in tip, tilt, and piston. The Keck telescopes utilize a segmented primary mirror similar to CELT employing a highly successful actuator design. However, because of its size and the shear number of actuators (3000 vs. 108 for Keck), CELT will require a different design. 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 examines four actuator concepts and presents a trade-off between them. The concept that best met the CELT requirements is described along with an analysis of its performance. The concept is based on techniques that achieve the required accuracy while providing a substantial amount of vibration attenuation and damping. A prototype actuator has been built to validate this concept. Preliminary tests confirm predicted behavior and future tests will establish a sound baseline for final design and production.

## 1. INTRODUCTION

### 1.1 CELT segmented primary mirror

The California Extremely Large Telescope (CELT) is being designed as a 30m aperture instrument providing a nine-fold increase in collecting area compared with today's largest ground-based telescopes and, through the use of adaptive optics, a five-fold improvement on the predicted near-IR spatial resolution of NASA's Next Generation Space Telescope (NGST). CELT will have a major impact on the capabilities of observational astronomy, improving the performance of the Keck telescopes in the same way that the Hubble Space Telescope (HST) improved imaging resolution from the then-best telescopes in the mid 1990's[1]. CELT's 30m primary mirror will be composed of 1000 segments, supported and controlled by an active system to emulate a monolithic reflecting surface in much the same way as the 36-segment primary mirrors of the Keck telescopes[2].

Substantial experience exists in the design and operation of the position actuation systems for the segments of segmented mirror telescopes. The aerospace industry, through programs like LODE (Large Optics Demonstration Experiment) and LAMP (Large Active Mirror Program) has experimented with actively controlled large segmented mirrors since the early 1970's. More recently other segmented telescopes have been proposed, such as the NGST, the successor to the Hubble Space Telescope, with a segmented 6-m primary mirror that can deploy in space. The most credible experience with mirror segment actuation has been gained by the designers of the 10-m Keck telescopes. These telescopes each use 108 actuators to position 36 primary mirror segments. They have proven that actively controlled, segmented mirrors is a sound technology for building large optics. The Keck segment positioning actuators are reliable and able to meet demanding performance requirements.

**1.2 Background—rigid and soft actuators**

Given that the Keck actuators have a track record of reliable operation under real-world conditions, it would seem logical to develop a CELT actuator based directly on the Keck design with only minor upgrades and improvements. Although this is a valid option for CELT, there are other approaches to the actuator design that can provide better performance with higher reliability, lower weight, cost and complexity. Because CELT will require 3000 segment actuators for operational conditions (a total of 4000 when including spares), the size, weight, cost, reliability, and power consumption of an individual actuator are factors substantially more critical than they were for the original Keck design nearly 15 years ago. This paper will develop the design rationale for a CELT actuator and provide analysis and preliminary calculations to substantiate the proposed concept. First it is important to distinguish between two major classes of actuators. The "rigid" actuator, characterized by a high inherent (mechanical) axial stiffness but a relatively slow response time, and the "soft" actuator, which derived its stiffness from a high bandwidth control loop but is otherwise more compliant when not powered.

**1.3 The rigid actuator**

The Keck telescope primary mirror control system employs a hybrid motorized screw actuator that has proved highly successful under actual operating conditions. The Keck actuators are considered rigid actuators because of their extremely high axial stiffness. They operate by converting the rotary motion of a DC torque motor into the linear motion required to move the mirror segment. A local control loop in each actuator uses a shaft encoder to ensure that the motor rotates exactly as commanded. The motor output directly drives a screw shaft with a 1mm pitch. A nut with recirculating rollers drives a "bellowfram" hydraulic motion reducer. This device reduces the nut motion by a factor 24, resulting in a resolution of 4 nm per motor step. A preload spring incorporated in the top end of the actuator eliminates mechanical backlash as the load on the actuator changes with telescope elevation angle. This type of actuator is the equivalent of a rigid connection between the mirror whiffletree and the mirror cell structure.

One of the main attractions of the rigid actuator is its operational simplicity in the context of the segment control system. This control system commands each individual actuator to move an incremental distance once each control cycle. As long as the control bandwidth of the position loop (encoder counts) is low enough, there are no active dynamic interactions between the mirror segment, the actuator, the mirror cell structure and the control system. The actuator behaves like an adjustable fixed mounting point between the whiffletree and the structure. In fact, if the environmental disturbances were small enough, or the telescope was sitting still, the actuators could be turned off. This concept has great appeal for designers confronting the task of controlling such a large electromechanical system. The system of segments and rigid actuators looks like a set of mirrors sitting on static supports that can be periodically adjusted by remote control using a computer. Dynamically, however, the whole primary behaves as a large, lightly damped structure.

Positioning requirements in the nanometer regime present a serious challenge for this type of mechanical actuator. Friction, stiction, hysteresis, creep, thermal distortion, and other nonlinearities cause the actuator to have a substantial stochastic component to its operation and do not allow the rotation of the motor to translate into a repeatable, consistent motion of the output shaft.

The Keck actuators owe their good performance to the clever use of extremely precise lead screw combined with the bellowfram, along with closed loop control on the motor rotation. The price for this level of performance is complexity, weight, and high cost. It is estimated that at the time they were built in the early 1990's, the Keck actuators cost about $10,000 each. At this price, the CELT actuator set would cost $40M.
In addition, the actuator's stiffness, usually seen as an advantage, has instead a serious consequence for the overall dynamic of the system. That is, the rigid actuator directly transfers forces from the mirror cell structure to the whiffletree and segment and from the segment/whiffletree back to the structure. Thus any disturbances from either direction will eventually cause some level of mechanical excitation. In addition, because of the actuator high axial stiffness, there is no means of dissipating the energy. In fact, the actuators control system can potentially add to the vibrational energy and drive the system unstable. The Keck segment position control system avoids this unstable condition by reducing the operating bandwidth to a very low level. The current mode of operation of the Keck system is essentially quasi-static with an effective bandwidth of about 0.25 Hz.

## 1.4 The soft actuator

Another means of providing precise position control for a mirror segment is to connect the mirror to a high bandwidth actuation device such as a voice coil motor, 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 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 output shaft of the actuator 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 segment-positioning control system.

The design concept of an actuator using a voice coil motor is a common approach to positioning electromechanical devices. It is relatively simple to implement and can provide high bandwidth control when necessary. Lightweight, compact, relatively inexpensive, and with very few moving parts, this actuator is capable of generating very large forces in a very short time and does not require lubrication. Most highly precise opto-mechanical systems, such as actively controlled mirrors used in telescopes 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.

The segments of a segmented mirror terrestrial telescope are constantly subjected to motion of the support structure and disturbances from the environment. Thus there is no operational, power saving, or lifetime advantage to an actuator with set-and-forget capability. Even though voice coil actuators must have electrical power and be functioning whenever the segment control system is in operation, as well as be monitored and positioned continuously by their local control loop, their operating duty cycle is really no different from the rigid actuator in a realistic operational environment. Figure 1-1 illustrates a voice coil driven soft actuator with a control loop closed around a position sensor.

## 1.5 Gravity Offloading

In terrestrial telescopes, any segment actuator, regardless of design, must supply the static force due to the mirror segment weight. The rigid actuator passively supports the full weight of the segment because it is essentially rigid. However, a voice coil driven actuator would have to use some of its capability to provide this force, independently of other requirements related to mirror control dynamics. Gravity offloading is a means of removing all of this static force, allowing the “soft” actuator to operate at extremely low power levels. Offloading serves the purpose of simultaneously reducing requirements for both actuator power and force. An offloaded voice coil (of any capacity) not only uses less energy because it doesn’t have to work as hard, it can also be substantially smaller than one designed for a non-offloaded system because it only has to provide forces associated with dynamic requirements such as repositioning the segment and correcting for mechanical, acoustic, aerodynamic, and other disturbances.

There are a number of means of providing gravity offloading falling into two broad categories, passive and active offloading. Passive offloading utilizes a counterweight on a lever arm to exactly offset the weight of

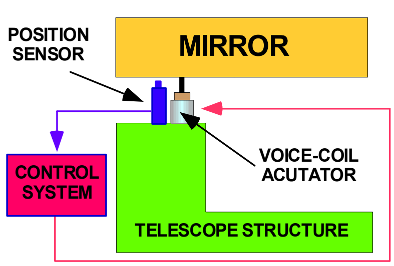


Figure 1-1: Voice coil driven soft actuator

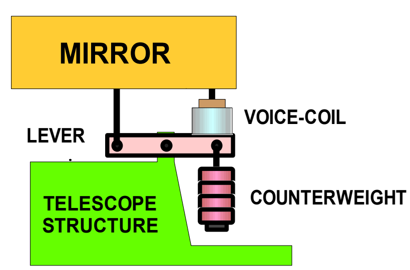


Figure 1-2 Passive counterweight offload for a voice-coil actuator

the segment. An interesting feature of a levered counterweight offloading system is that once the system is in balance, it remains in balance independent of the elevation angle of the telescope. Figure 1-2 shows a voice coil driven segment actuator that uses passive counterweights for offloading.

In active offloading, the amount of required offload is determined by measuring the static current in the voice coil actuator and adjusting the offload until the static current is driven to zero. An auxiliary electric motor may be used to control the extension or compression of a spring that applies the offload force. This motor is usually very small, operates at low speed and only intermittently, and requires a negligible amount of power. The motor is controlled to extend or compress the spring as a function of the DC value of the current in the voice coil actuator. During the (short) time when the offloader is making its adjustment, a small transient force is generated by the compression (or extension) of the spring, causing a disturbance to the system. However, the high bandwidth control loop of the voice coil actuator easily reduces this disturbance to a negligible level.

Voice coil driven soft actuators are completely dependent on the use of local closed loop control systems, one for the segment position control and one for active offloading. The architecture and design of these actuator control systems are the key to making the actuator meet performance requirements, be readily producible within a reasonable budget, and have acceptable reliability and operational lifetime. For these reasons, the analysis of the control systems and their impact on actuator sizing, duty cycle, segment control system performance, power dissipation, disturbance rejection, sensor requirements, and mechanical configuration takes up a major portion of this paper. All of these subjects are examined in detail in Section 3.

## 2. CELT Actuator Concept Descriptions

### 2.1 Requirements

The CELT Segment Positioning Actuators (CSPA) must satisfy a number of primary requirements that are listed below:

| | |
|---|---|
| Range | > 1.2 mm |
| Rms. position error over 20 minutes | < 7 nm |
| Slew rate | > 10 μm/s (full stroke in 2 minutes) |
| Transverse load capacity | > 5 Kg |
| Axial load capacity | > 30 Kg |
| Transverse stiffness | > 0.1 N/μm |
| Axial stiffness | > 10 N/μm (~ 100 Hz resonance) |
| Local average power dissipation | < 2 W |
| Lifetime | > 10 Years |
| Survival temperature | -18 to +22 °C |
| Operating temperature | -6 to +10 °C |
| Operating humidity | 1 to 100% condensing |

### 2.2 Design philosophy

The review of the requirements reveals the three most challenging items of the CSPA design:

a) achieving high position accuracy with a relatively large stroke, a dynamic range of almost 200,000/1.
b) being able to operate with this accuracy under a load that can vary from zero to 30 Kg.
c) keeping the axial stiffness high enough to sustain external disturbances without loss of accuracy.

Item c) is in fact equivalent to requiring that the system moves less than 7nm under a 0.07 N disturbance. This number is important since it can be used to evaluate system performance in a more result oriented mode, the idea being that one could use other means (rather than straight mechanical stiffness) to achieve the same disturbance rejection result.

Similarly, it is seen from item b) that a 30Kg load change (i.e. 294 N) would produce a position variation of 29.4 μm given the required stiffness of 10 N/μm. This variation is almost four orders of magnitude larger than the error budget of 7 nm. However this precision has to hold for only 20 minutes. If one assumes an average motion of the telescope of 15 arcsec/s (Earth rate), at 45 deg elevation, the actuator load change due to gravity is about 18 N, thus a position change of 1.8μm. This number is still 260 times larger than the precision requirement.

A similar challenge arises from possible temperature variations. For example, a 30 cm long steel actuator will expand about 3.5 μm /°C. Conversely, a temperature variation of only 2/1000 °C will produce 7 nm change.

For all these reasons, the mechanical design of the actuator must carefully address the positioning requirements as well as providing a clear definition of what "actuator position" really means. The cartoon shown in Figure 2-1 represents a simplified structural model of the actuator and the possible sensing schemes of the actuator position. In this representation, the actuator is made of three essential parts, an output shaft (relatively rigid) that transmits the motion to the mirror or mirror interface, a motion mechanism (e.g. screw drive), and a drive mechanism (e.g. motor/gear box). True sensing of the position is shown as #1, directly between the output of the actuator and a reference point on the telescope structure. If the output shaft is sufficiently rigid (and thermally stable) to satisfy the requirements discussed previously, then sensing can occur at the motion mechanism (#2). Often, sensing is simply provided at the drive mechanism itself (e.g. counting motor turns) which now requires more demanding performance from the motion mechanism (#3). Accuracy requirements also have a direct effect upon the drive mechanism design, which must have the correct resolution, accuracy and repeatability, and function under variable load conditions.

Additional considerations for the CSPA design include space and weight constraints, reliability and cost. The general philosophy of the design is thus to emphasize simplicity, direct actuation and sensing.

**2.3 Soft vs. rigid actuation**

"Rigid" actuation occurs when the motion is mechanically imposed via gears, screws etc. The kinematics and the stiffness of the system thus purely define the position of the output shaft. In this case, external disturbances are attenuated in direct proportion to the system stiffness. However, this is only true for quasi-static disturbances. For time-varying disturbances, certain frequencies may be considerably amplified by structural resonances.

Using a simple spring/mass model for the actuator/mirror system with the 10 N/μm required stiffness, and assuming a sinusoidal force disturbance of 0.05 N is acting on the actuator, one may calculate the position error as a function of frequency as shown in Figure 2-2. The amount of natural damping significantly affects the error at resonance, but not much so at lower frequencies. In this example, while the error is 5 nm at very low frequencies, it reaches 7 nm at 60 Hz, even with 2% natural damping, an already high value for a usual steel structure. In order to meet the requirements from zero to 100 Hz, the damping should be as high as 33% as shown on the bottom curve of Figure 2-2. This is not achievable practically with purely passive means.

In the "soft" actuation concept, the desired position is achieved by constantly applying the correct force to the mirror. When the control system is turned off, the system is generally exhibiting a very low natural stiffness. When the control system is on, the stiffness is caused by the control laws that essentially mimic the behavior of a spring (i.e. force proportional to displacement).

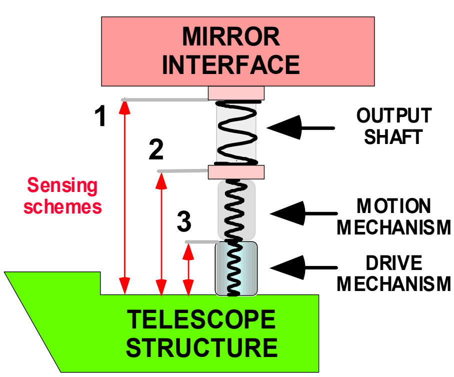


Figure 2-1. Actuator position sensing

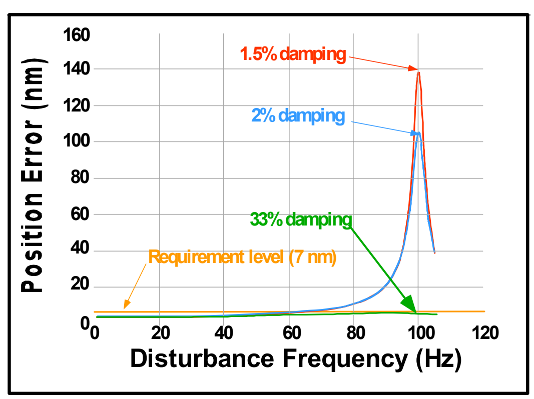


Figure 2-2. Position error for rigid actuator

However, there is a significant difference in this case as far as the damping is concerned because the feedback loop can be used to introduce as much damping as is necessary. In other words, the control laws can mimic a dash pot as easily as it does a spring. Thus the 33% damping case shown in Figure 2-2 is achievable. In fact, in most "optimal" control systems, the damping ratio is usually set at 70% .

In addition, the spring and damping effects can be made dependent on the frequency. For example, in a classical PID control system, the stiffness is effectively infinite at zero frequency, meaning that no matter what constant load or thermal expansion is present, the position of the actuator is always the same. This presents a significant advantage over the "rigid" type, because the accuracy of the system is independent of external loads, thermal expansion, or any other inaccuracies and biases that may have been introduced mechanically.

Another very important feature of the soft actuation schemes is its ability to introduce damping in the overall system. To exemplify this property, we may consider a more realistic dynamical model for the mirror as shown in Figure 2-3. In this model, the whiffletree is represented by a spring ($K_w$) between the mirror and the actuator output shaft. The actuator coil, spring and control system can be lumped into an equivalent spring/mass/damper system where spring and damping constants (K and D) are determined by the mass M of the interface and the control equation:

$$F_a = - G_p\, u - G_d\, du/dt \qquad (1)$$

where $F_a$ is the actuator force, $G_p$ and $G_d$ are proportional and differential gains respectively. The position sensor of the control system measures the displacement u. From the dynamic equation:

$$M\, d^2u/dt^2 = F_a \qquad (2)$$

one finds the relations:

$$K = G_p\,; \qquad D = G_d \qquad (3)$$

Adding the mirror and the whiffletree results in a coupled system. Calling z the inertial displacement of the mirror, the full equations of motion are given by:

$$M\, d^2u/dt^2 + D\, du/dt + (K + K_w)\, u = K_w\, z \qquad (4)$$

$$M_m\, d^2 z/dt^2 + D_m\, dz/dt + K_w\, z = K_w\, u + F_e \qquad (5)$$

where $F_e$ is the external disturbance force acting on the mirror (e.g. wind). The frequency response of the system can be calculated from Eqs. 4 and 5. In particular, the motion z of the mirror is given as a function of the frequency ω of excitation by:

$$z(\omega) = P(\omega) / Q(\omega)\ F_e \qquad (6)$$

where:

$$P(\omega) = [-\omega^2 + 2j\,(\zeta\omega_o + \alpha\zeta_m\omega_m)\,\omega + \omega_o^2\,] / M_m$$

$$Q(\omega) = \omega^4 - 2j\,[(\zeta\omega_o + (1+\alpha)\zeta_m\omega_m\,]\,\omega^3 - [\,(1+\alpha)\omega_m^2 + \omega_o^2 + 4\zeta\zeta_m\omega_o\omega_m\,]\,\omega^2 + 2j\,[\,(\zeta\omega_o + \zeta_m\omega_m)\omega_o\omega_m\,]\,\omega + \omega_o^2\,\omega_m^2$$

and:

$\alpha = M_w / M$, $\omega_o^2 = K / M$, $\omega_m^2 = K_w / M_m$, $2\,\zeta_m\,\omega_m = D_m / M_m$, and $2\,\zeta\,\omega_o = D / M$.

The motion z of the mirror (Eq. 6) is plotted on Figure 2-4 showing clearly that the active actuator is able to damp the motion of the mirror/whiffletree system, a feature that is not available at all with a rigid type of actuator as shown also on Figure 2-4. The motions u of both rigid and soft actuators (Eq. 5) resulting from a disturbance force applied directly to the actuator are also plotted on Figure 2-4 as a function of frequency. It is seen that the soft actuator reduces the resonant mirror response by a factor 20.
The following parameter values were used in this example:

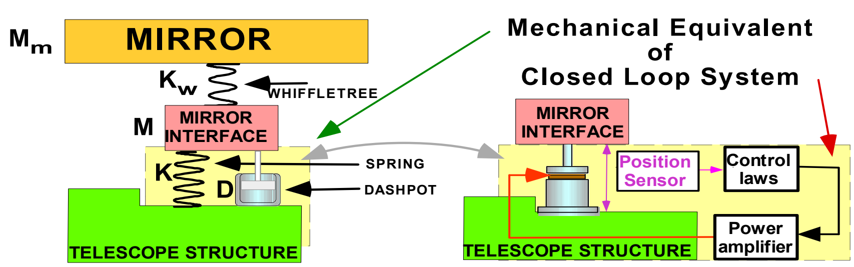


Figure 2-3. Soft actuator damping effect model

| | | | |
|---|---|---|---|
| Mirror/whiffletree system: | $M_w$ = 75 Kg, | $K_w$= 10 µm/N, | $\zeta_m$ = 0.01 |
| Actuator: | M=15 Kg, | $\omega_o$ = 691 rad/s (110Hz), | $\zeta$=2. |

For the rigid actuator it was assumed that K was 10 µm/N and ζ was also 1%.
Thus it is seen that a soft actuator has in fact a dual purpose: one is to position accurately the mirror, the second is to provide damping to the otherwise undamped mirror/whiffletree system.

**2.4 Off-loaded voice-coil concept**
To combine the advantages of rigid actuators (which can hold a load without dissipating power) and soft actuators (which provide great accuracy and stability), the concept presented in this paper uses a voice coil as the main position control actuator coupled with a mechanical off-load system.

The principle of such a system is shown in Figure 2-5. A spring, supporting the weight of the mirror, has its tension (or compression) adjusted by a motor/gear box/lead screw system to accommodate variations of the load due to changes in gravity vector as the telescope moves in elevation. The spring/motor system provides approximately the force necessary to balance the applied load L. The voice coil actuator provides the remainder of the force (δL) plus a time-varying force (f) that provides the fine adjustment for all other disturbances and errors. The control system reads the actuator position from a position sensor measuring the motion δx of the actuator shaft with respect to the telescope frame. Ideally this sensor is placed as close as possible to the mirror, but, as noted earlier, if the stiffness of the output shaft is large enough, this measurement can be made directly at the voice-coil actuator output.

**2.5 Active vs. passive offloading**
If the mechanism by which the offloading is accomplished is of the generic type represented in Figure 2-5, the motor needs to be actively driven to follow the changes in gravity loads. The usual scheme is to use the voice coil drive current as the driving signal, so that the offloader motor is always trying to drive the voice-coil actuator current to zero. At zenith, the spring is compressed and supports the whole weight of the mirror. This load must be carried as well by the ball/screw gearbox system, and the motor has to develop a reasonable amount of torque to handle this load.

The use of a lever system can reduce the demand on the motor/gear/screw/spring system. With a 10:1 lever for example, the design is greatly simplified, using an expansion spring driven by a small, winch type motor. The other way to achieve offloading is to use a passive counterweight to automatically balance the mirror weight. This method has been used consistently in the past for balancing astronomical telescopes and offers some advantages. The lever also reduces the total weight increase to a reasonable amount (e.g 10% with a 10:1 lever ratio). Such a design is simple and robust, with a minimum of electronic and mechanical components.

The drawback of this design is that it may require an adjustment of the counterweight for each actuator unit, to accommodate for possible manufacturing differences. If the weight unbalance error could be maintained below 1% for example, then the actuator would have to steadily produce at most 0.03N (about 1/10 ounce). With a typical voice coil the steady power dissipation would be about 2 mW.

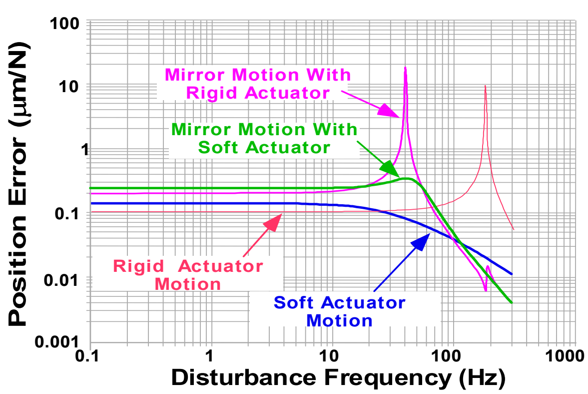


Figure 2-4. Mirror Response with Soft and Rigid Actuators

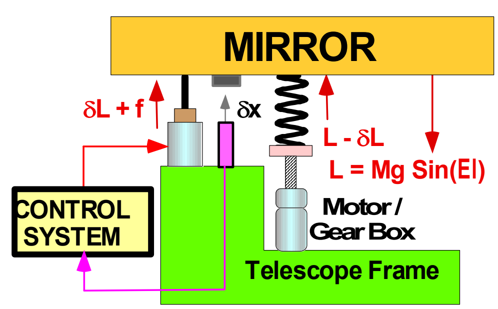


Figure 2-5. Off-loaded voice-coil

**2.6 Concept Selection**

In section 2.1 to 2.5 a tradeoff between rigid and soft actuators was analyzed. The analysis indicated that were a number of advantages to the soft actuator. These included lack of stiction and friction, no rubbing, sliding or lubricated parts, fewer overall components, capability for smoother, higher bandwidth control, and the ability to introduce damping not only of motions of the mirror segment, but of the support structure as well. As a result, the voice coil driven soft actuator is the basis for the design concepts recommended for CELT. Four actuator design concepts were developed. All utilized the offloaded voice coil as the primary control device. The four concepts are described briefly below.

**Concept A Direct Active Offload**

The direct active offload design is shown as Concept A in Figure 2-6. In this design a motor-driven compression spring is used to provide gravity offload. Both the offloader and the voice coil are attached to the actuator output shaft directly without the use of levers. There are no counterweights. This design is relatively lightweight and compact but requires more force and torque from the voice coil actuator and the offload motor than other concepts.

**Concept B Active Offload with Mechanical Advantage**

Concept B in Figure 2-7 also uses a motor-driven spring for gravity offload. However, in this concept, the spring is of the extension type and both the voice coil and the offload motor act through a lever that provides a mechanical advantage. This design is relatively lightweight like Concept A, but cannot be built in as compact package because of the lever geometry. The voice coil and offload motor can be substantially smaller and lower power than those in Concept A because of the lever arrangement.

**Concept C Passive Offload with Mechanical Advantage**

The third concept in Figure 2-8 does not use either a spring or a motor to provide offload. In Concept C, the offload force is generated by a counterweight acting through a lever. The voice coil acts through the same lever. This design has the largest and heaviest package of the four concepts, however, it is also the simplest, potentially the most reliable, and uses the least power. For optimum performance, it does require that the counterweight be carefully adjusted to exactly balance the load of the mirror segment and support hardware.

**Concept D Hybrid Active/Passive Offload with Mechanical Advantage**

This last design shown in Figure 2-9 combines the features of Concepts B and C. A counterweight acting through a lever provides most of the offload force. In order to make the actuator lighter and remove the requirement for accurate adjustment of the counterweight after installation into the segment support system, Concept D also employs a motor-driven extension spring. Because the combination of the partial counterweight and the lever system make the trim forces quite small, both the trim motor and spring can be compact, lightweight, and use very little power. Also the voice coil applies only very small forces to maintain accurate position. Concept D has packaging volume requirements similar to Concept C, but is lighter weight.

**2.8 Concept Selection Trade Table**

A summary of the advantages and disadvantages of design concepts A through D is presented in table 2-1. Because each of the concepts utilizes the same voice coil motor and local position feedback, the performance of the segment position system is not considered in the trade.

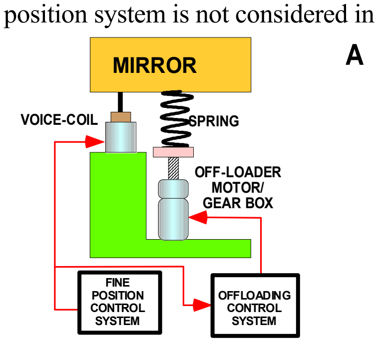


Figure 2-6 Concept A
Directly supported mirror with active offload

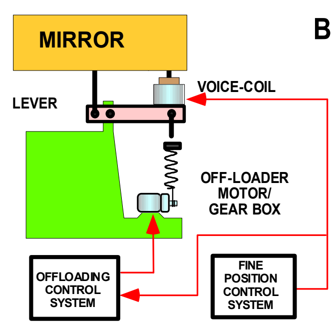


Figure 2-7 Concept B
Active offload with mechanical advantage

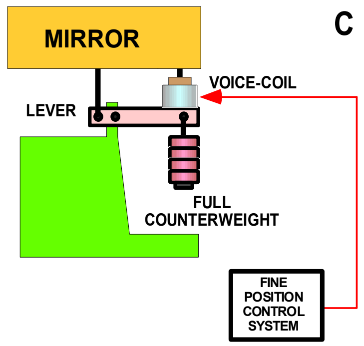


Figure 2-8 Concept C
Passive offload with mechanical advantage

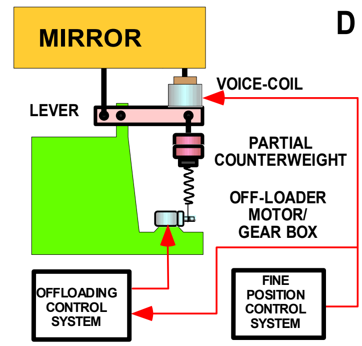


Figure 2-9 Concept D
Hybrid offload with mechanical advantage

| | SIZE | COST | WEIGHT | COMPLEXITY | POWER | RELIABILITY |
|---|---|---|---|---|---|---|
| **A** | smallest of 4 concepts | highest because of larger actuator and motor , plus screw drive for spring | lightest of 4 concepts<br><br>- no counterweight | compression spring is hard to package | uses most power of 4 designs | Least reliable because screw will wear |
| **B** | larger diameter than A | intermediate cost-need offload motor plus electronics but smaller than A | weight only slightly more than A because no counterweight and smaller voice coil | simpler than A but lever design is critical | uses less power than A but more than C and D | More reliable than A, Same as D, and less than C |
| **C** | same package diameter as B and D, shortest of B C D | least expensive of 4 designs no active offload | heaviest of 4 designs<br><br>-heavy counterweight | simplest of 4 designs- no active offload | lowest power of 4 designs lever plus no offload power is advantage | Most reliable of 4 designs because has fewest parts and no active offload |
| **D** | same package diameter as B and C, same length as B | same cost as B, more than C less than A | heavier than B, less than C | same complexity as B | lower power than A and B, more than C | More reliable than A or B because less stress on motor |

Table 2-1. Concept selection table

**2.9 Design Recommendation**

Based on rating details in the table, two of the design concepts appear to have the most appeal for CELT. Selection of a specific concept may be influenced by system-level considerations. If the primary emphasis in the CELT design is simplicity and reliability, then Concept C, which utilizes passive-only offloading, is the clear choice. This design has the fewest mechanical parts, is completely fail safe (the offload is always in balance), and has the simplest electronic subsystem. This concept requires a 3 Kg counterweight which does not seem excessive, but translates into 9000 kg of unproductive weight on the 3000-actuator CELT. This additional mass has impacts on the size of the rest of the structure, structural modes and frequencies, the sizing of the hydrostatic bearings, the AZ-EL drives, etc.

Based on the evaluation in table 2-1, Concept D appears to be the next best trade between weight and power. A reduced-size counterweight, as small as 1 kg, can substantially reduce the size of the offload spring, motor, and gearbox.. The motor-spring system automatically trims the offload force based on measuring the residual DC current in the voice coil actuator . This design was chosen for the building of a prototype.

## 3. ACTUATOR PROTOTYPE DESIGN

**3.1 Mechanical Design**

Based upon the trades discussed in section 2, a preliminary conceptual design was developed for the CSPA as shown in Figure 3-1. 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.

This design represents a significant departure from mechanical actuators that use any kind of bushings, bearings or screw drives. Three flexural levers arranged in a triangular configuration are attached, on one end, to an upper plate supporting the output shaft, and at the other end, to the counterweight. The three pivots are supported by the outside canister. All flexures are under tension, thus avoiding buckling. As a result, thinner flexures are used, reducing bending stresses and thus significantly increasing their lifetime.

The size and weight of the counterweight can be adjusted from a minimum required for structural rigidity, to the full balanced value. A small extension spring connects the counterweight to the offload motor/gear box system. The optimal choice for the spring stiffness is a function of the residual gravity unbalance after counter-weighting the system.

The voice-coil actuator has its coil attached to the output shaft. Its magnet is mounted on the counterweight plate thus saving on the total system weight and power.  The output shaft is attached to two disk flexures which allow it to translate but provide enough rigidity in the transverse axis to satisfy the 5Kg, 0.1 N/µm requirements. These flexures are situated at both ends of a tube (chimney) which acts as a geometric reference for the actuator system and the telescope structure interface. The distance between the two flexures provides the rigidity required to handle the gravity couple resulting from the offset weight of the mirror/whiffletree system.  The chimney is about 2” diameter and passes through a mounting hole inside a telescope structural node. Its bottom part is attached to the actuator’s outside canister that contains all the mechanisms.  The electronic boards are mounted inside the canister and are easily accessed though lateral doors. The final electro-mechanical design of the prototype actuator is shown in Figures 3.2.

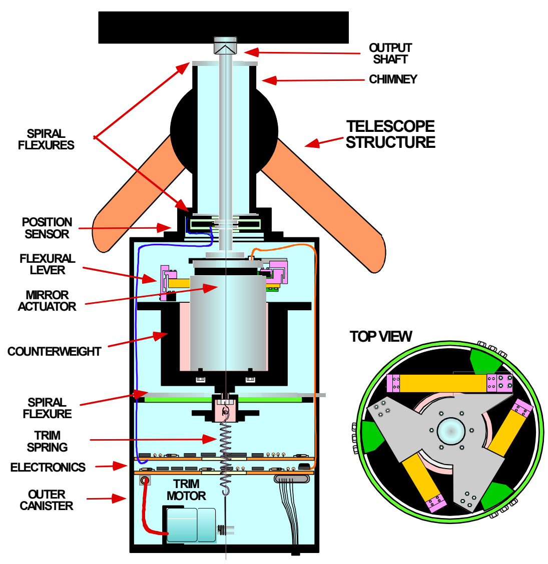


Figure 3-1 CPMA Conceptual Design

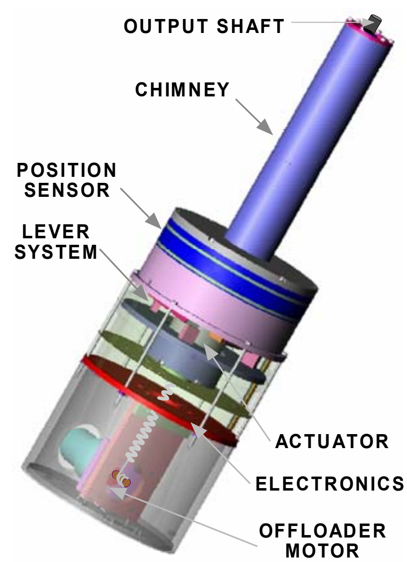


Figure 3-2 Final Electromechanical Design

### 3.2 Control system design

The control system wrapped around the voice coil actuator uses an analog position sensor to determine the position of the actuator shaft. From this signal, the control laws generate a current command to the actuator so that the actuator position u is maintained equal to a desired (commanded) position $u_{command}$ (Figure 3-3).  The control laws shown in figure 3-3 are of a classical PID controller where the actuator force is obtained as a linear combination of proportional, differential, and integral of the position error.  The differential term not

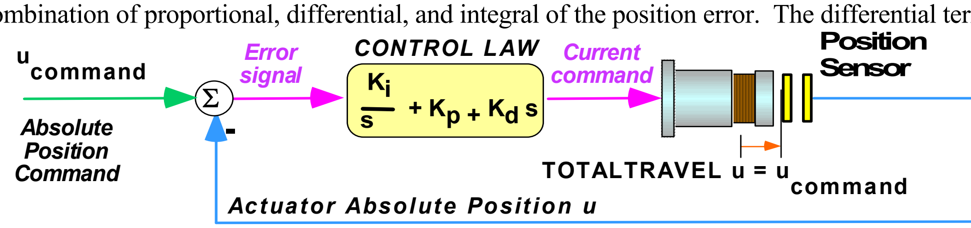


Figure 3-3 Position control system schematic

only provides the necessary damping to the actuator itself, but also to the mirror/whiffletree system as discussed in section II. The integral term essentially provides infinite stiffness at zero frequency, i.e. it makes sure that the static error is exactly zero.

The typical error response of this type of system under external sinusoidal disturbance is shown in Figure 3-4. It starts from zero when the disturbance frequency is zero, and reaches a maximum around the bandwidth of the control system. Above the bandwidth frequency, the control system itself becomes ineffective, however the inertia forces (mirror mass in this case) provide an increasing attenuation with frequency (response proportional to $1/f^2$). The closed-loop error function plotted in Figure 3-4 corresponds to a 30 Hz control bandwidth and 10 N/µm for the rigid system stiffness. The complete control system schematic, which includes the offloading system, is shown in Figure 3-5.

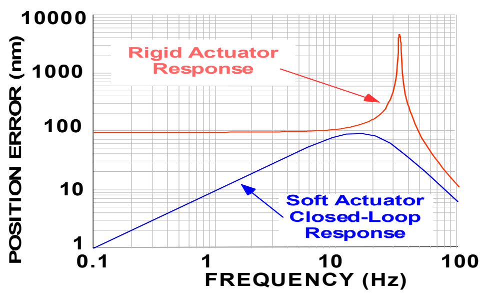

Figure 3-4 PID Controller vs. Rigid Actuator Response

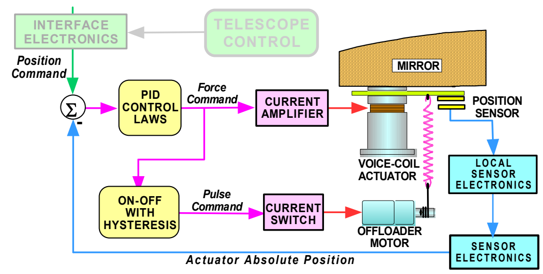

Figure 3-5 Overall Mirror Actuator Control system

### 3.3 Electronics Design

The electronic architecture for the CSPA comprises two separate control systems, one for the output position of the actuator, and one for the active/passive offload system. There is also a separate set of electronics for the capacitive position sensor that monitors the actuator position.

The sensor electronics are separated into a local electronics board that contains preamplifiers and is located immediately adjacent to the sensor capacitor plates and the remainder of the sensor electronics that is located on the same board as the rest of the control system. This arrangement greatly enhances the sensor signal to noise ratio and minimizes stray capacitance.

The actuator control electronics consist of the PID actuator position controller, the on-off offloader controller, power amplifiers for the voice coil and offload motors, and the position command decoding electronics. The control of the offloading system is accomplished by simple on/off logic. The average current in the voice-coil actuator is sampled at some low rate (10-100 /s) and compared to a bipolar threshold. When the threshold is exceeded, a pulse is sent to a current switch that drives the motor in one direction or the other for a very short time. This method is extremely efficient and requires a minimum of electronic parts.

All of the control system electronics, with the exception of the position command-decoding chip, are analog. They can be implemented using conventional off-the-shelf components on a printed circuit board. In the prototype design, the electronics is packaged inside the actuator housing thereby eliminating cabling and connectors that add cost and reduce reliability.

### 3.4 Performance Evaluation

This section presents the predicted performance of the prototype CPMA using the following parameters

| Parameter | Symbol | Value | Units |
|---|---|---|---|
| Mirror mass (per actuator) | $M_m$ | 25 | Kg |
| Actuator mass (includes whiffletree) | M | 5 | Kg |
| Counterweight mass (minimum) | $M_c$ | 1 | Kg |
| Position sensor noise equivalent | $n_u$ | 3.6 | nm |
| Actuator power constant | $C_p$ | 5 | $N/W^{-1/2}$ |
| Offloader spring constant | $K_s$ | 490 | N/m |

Table 3-1 Baseline design parameters

The major challenge for the CSPA is to properly handle external disturbances, the largest of which is the residual wind blowing on the primary through the dome aperture. Although no direct data is available at this time to quantify the effect of the wind, results obtained for the Keck telescope are used in this paper to obtain an estimate of the mirror position error. With 3 actuators per segments, each actuator only sees one third of the disturbance, and only one third of the mass of the mirror/whiffletree system. Given the control system described in section 3 one can calculate the response of the mirror as a function of frequency. In the actual CELT design, the whiffletree is not rigid but behaves like a spring between the mirror and the actuator. The dynamic analysis of the system shows that a higher actuator bandwidth (~100 Hz) is required to provide the necessary damping of the system. Because of the whiffletree flexibility, the quasi-static response of the system stays around 100 nm/N, even for a perfectly controlled actuator. Figure 3-7 shows the position response of both actuator and mirror to a 1N disturbance as a function of frequency. The whiffletree stiffness is about 10 N/µm (corresponding to about 60 Hz resonance of the mirror/whiffletree system). The parameters used are those of table 3-1.

The damping effect of the actuator can be seen clearly (blue curve). The analysis also shows that the response at low frequency is dominated by the compliance of the whiffletree. The position response of the actuator alone (purple) is quite modified when the mirror/whiffletree system is attached, and shows the two resonant modes of that system appearing as zeros (green curve).

A wind spectrum was derived from actual wind velocity measurements[2] on Mauna Kea in 1992. The static pressure on a 1 $m^2$ segment results in about 1N of force. Using this wind model, and the frequency responses of figure 3-7, one can estimate the response of the mirror and the actuator output shaft position at various frequencies. A plot of these results is shown in Figure 3-8. The actuator position clearly meets or exceeds the requirements by staying below 2 nm. The mirror position however is again driven by the flexibility of the whiffletree system, but the effect of the spring/mass resonance at 60 Hz is properly damped and does not significantly affect the overall performance.

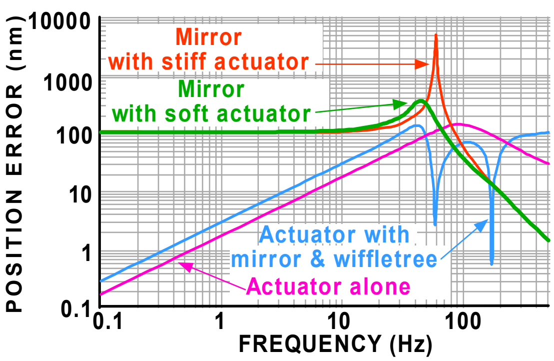


Figure 3-7 Responses to External Disturbance -Flexible Whiffletree Case

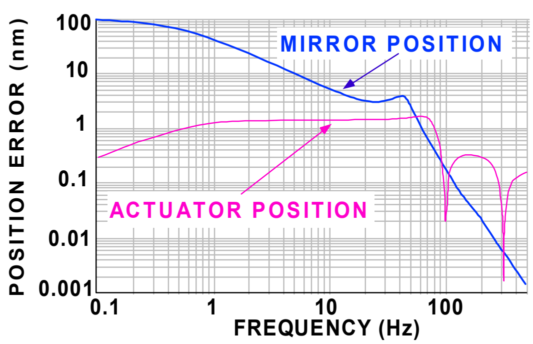


Figure 3-8 Estimated responses to Wind

### 3.4.2 Performance under offloading conditions

When the offloader motor is activated, it essentially compresses or expands the offloading spring by a small amount resulting in a perturbation force on the mirror. In the prototype design, the motor minimum motion produces a displacement of about 63 µm, resulting in a perturbation force of 0.03 N. Dynamic analysis in closed-loop conditions shows that corresponding mirror motion stays below 2 nm.

### 3.4.3 Performance under tracking conditions

During normal operations, the telescope may be required to slew almost from horizon to Zenith to horizon (tracking). This means that the gravity load on the actuator can go from 0 to 114 N in 6 hours. Under these conditions the offloader spring must produce 30 N corresponding to a maximum extension of 60 mm and a tracking rate of $V_{track}$ of 2.78 µm/s.

Because of the deformation of the telescope under gravity loads, it is expected that the mirror will move a maximum of 1.2mm from Zenith to horizon, thus a tracking rate of about 57 nm/s. While the PID controller perfectly eliminates static errors, it only reduces tracking errors by some amount. The exact form of the position error (hang-off) due to the tracking rate $V_{track}$ is given by Vtrack / (2 $\omega_b$), where $\omega_b$ is the controller bandwidth (about 630 rad/s). The estimated hang-off error is thus less than about 0.03 nm.

### 3.4.4 Power Dissipation

The power dissipated by the voice coil actuator is very dependent on the environmental and observing conditions. If the system is absolutely quiescent, there is a residual excitation of the actuators due to sensor noise coming back from the feedback loop. In addition, there is the actuator noise itself, due to the drive electronics. Finally, if there are any dynamic perturbations to the system, the actuators have to react to the disturbance forces by applying counter forces, thus dissipating even more power.

The major contributors to actuator control forces are sensor noise (0.014 N), actuator noise (0.005 N), and wind forces (0.033 N). The total nominal power dissipated in the actuator is obtained by taking the sum squares of the preceding contributions and applying the power factor. It was found that $P_{total}$ = 0.0524 mW. For a coil resistance of 2 Ohms, the corresponding current is about 5 mA. If the DC power supply for the actuator power amplifier uses ± 15V, the power dissipated in the electronics due to actuator commands is then $P_e$ = 30 * 5 = 150 mW. The majority of the power dissipation comes from the sensor and signal processing electronics and was estimated at about 1.5 W, thus the total power is less than the requirements of < 2 W.

## 4. PROTOTYPE ACTUATOR HARDWARE

### 4.1 Electromechanical Hardware

The development of the prototype actuator was undertaken with the goals of securing the new technology, establishing the feasibility of the concept, and understanding the risk associated with reliability so that cost-effective solutions could be developed and incorporated into the final product.

The main body of actuator was built using standard aluminum tubing for the load bearing shell, Plexiglas for the protection of the mechanism, and stainless steel for the load bearing mechanisms. Special flexural levers were designed and built to achieve infinite lifetime. The body of the actuator is 8-in diameter and about 13-in height. The sensor and actuator drive electronics was integrated inside the actuator, communicating with the outside world with a 25-pin D connector. This allowed to give analog position commands and measure actual position and actuator current.

### 4.1 Preliminary Tests

To provide for a meaningful dynamic test and emulate the mirror segment expected gravity load, a 70 lb. iron weight was attached to the output shaft as shown in figure 4-1. A special tripod was built to hold the actuator and allow the system to be tested vertically (Figure 4-2).

An electronic drive test unit was built to command and monitor the position and installed below the actuator as shown in figure 4-2. The tests measured control system bandwidth (about 100 Hz) and residual position error (about 17 nm and mostly due to seismic environment at the testing site - the sensor noise itself was below 2 nm). The system power was 1.75 Watts with a ± 12 V power supply. The offloading system was also tested successfully by adding extra weights to the main one, and by removing the dummy weight entirely to emulate the telescope horizon pointing condition. Over 1.2 mm displacement of the actuator output shaft was achieved.

## 5. CONCLUSIONS

A prototype actuator has been developed for the California Extremely Large Telescope that meets all the requirements and provide additional value to the telescope operation 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 capacitive 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 and weight of the system.

Several options have been considered and the final selection consists of an actuator in which the weight of the mirror is offloaded partially by a counterweight, partially by a motor-driven spring (offloader). Because the best combination of these two means of offloading is not completely known at this time, from full counterweighting to full motor offloading, this hybrid design offers flexibility for future adjustment.

The actuator is self-contained and can be removed in its entirety from the telescope by sliding the neck part out of the telescope support structure node. The electronics and offloading system can be accessed and serviced from its bottom and side without removing the actuator from the telescope.

The prototype will be further tested to evaluate its performance and reliability, and optimize the way in which counterweighting and active offloading should be combined.

Figure 4-1 Actuator with 70 lb weight

Figure 4-2 Actuator with mount and test electronics

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2) J-N Aubrun and K. R. Lorell; Final Report on the Effect on Image Quality of Ten Meter Telescope Segment Control System Dynamics, Keck Observatory Science Office, Lawrence Berkeley Laboratory, March, 1987