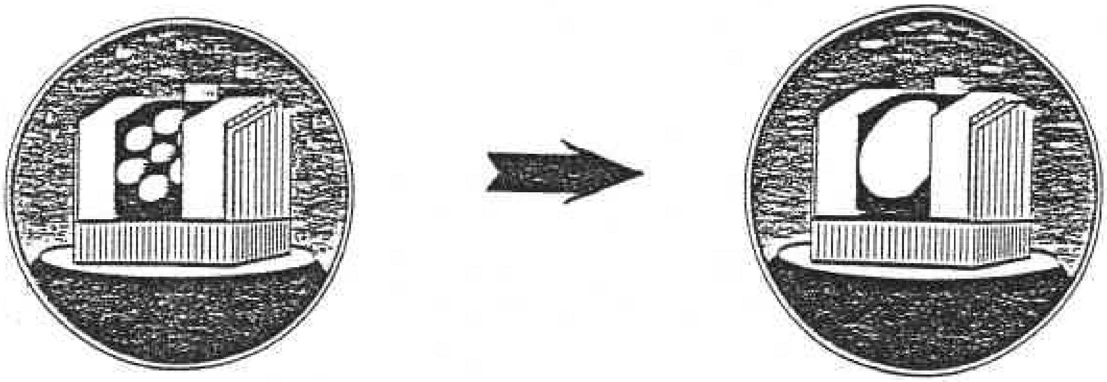
# 6.5 METER TELESCOPE



**MMTO Conversion Technical Memorandum #96-2**

The Position Actuators of the 6.5m Borosilicate Honeycomb Primary Mirrors

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# THE POSITION ACTUATORS OF THE 6.5m BOROSILICATE HONEYCOMB PRIMARY MIRRORS

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## ABSTRACT

In order to collect as much information as possible from the universe, the latest generation of astronomical telescopes have exceptionally large diameter primary mirrors. This dramatic increase in mirror diameter, and corresponding increase in weight, has placed ever increasing demands on the technical performance of the mirror support system.
In this paper the authors discuss the mechanical design, fabrication, and testing of the six servo controlled position-actuators that mechanically link the 6.5 m honeycomb mirror to six rigidly reinforced locations in the Multiple Mirror Telescope Conversion mirror cell. During telescope operation, these adjustable length actuators precisely control the six degrees of freedom of motion of the mirror. Each actuator has a high axial stiffness to assure that the natural frequency of the mirror does not degrade the optical performance of the telescope. Flexures are provided on each end of the actuators to minimize any moments applied to the attachment of the actuator to the mirror. These actuators provide a precise measurement of the external forces applied to the mirror, such as wind loads, for the control of the pneumatic force system that supports the weight of the mirror. The total length of each actuator can be measured to sub-micron resolution upon request. Each actuator has a reliable fail-safe system that limits the compressive and tensile forces that can be applied to the mirror. The position-actuators meet all of the above technical specifications in both tension and compression.

**Keywords:** astronomical telescope, primary mirror, force actuators, position-actuators, natural frequency

## 1. M1 MIRROR SUPPORT SYSTEMS

The 6.5m borosilicate primary mirror of the MMT (Multiple Mirror Telescope) Conversion adopts a mirror support system able to hold itself in the telescope structure in such a way that the forces of gravity, wind and telescope acceleration do not distort the optical surface of the mirror. The high rigidity of the glass mirror blank affect significantly the shape of the reflecting surface of all existing optical/infrared telescopes, and to avoid bending the mirror, it is necessary to float the blank against the force of gravity as the telescope changes its orientation. This technique is known as 'astatic floatation'. In order to prevent the glass bending under gravity, the weight of the mirror is supported by many distributed reaction forces able to follow the orientation of the mirror from horizon to zenith pointing. The distributed force actuators essentially apply forces to counteract gravity maintaining the mirror figure even as the telescope structure bends or flexes under the load. The position of the mirror in space is then determinated by six fixed points, able to locate precisely the floating mirror without carrying any load. The six fixed points, called 'hardpoints', typically work at zero force but they are able to measure any external load applied to the mirror by the gravity, the wind or by the telescope drivers. This information is necessary to change in real time the counterbalance forces applied by the force actuators driven by the mirror support control

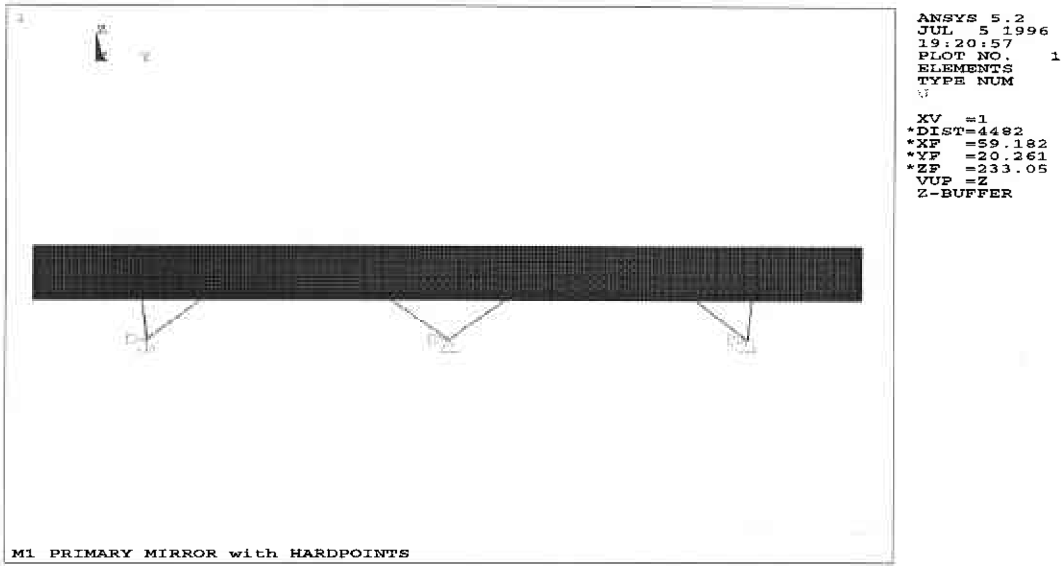


**FIG.1 - M1 MIRROR FEA MODEL**

system. In this configuration we have two actuator systems applied on the primary mirror backplate: the force-actuators system, able to support the mirror in any orientation and able to control the shape of the reflecting surface by the active optics concept, and the position-actuators system, responsible of the location of the mirror in the 3D space. In this paper we describe in details the conceptual design of the latter actuators, the mechanics and the experimental tests carried out during the development phase.

## 2. THE REQUIRED PERFORMANCES OF THE POSITION ACTUATORS

The main functions of the position-actuators, as briefly described in the previous paragraph, is to determine the spatial location of the primary mirror without applying forces on its backplate; to do that it is necessary to measure on real time the load applied on each hardpoints and to forward this information to the force actuators able to counterbalance the mirror mass applying the needed forces in the proper distributed pattern.
The hardpoints have also other functions that we considered in the mechanical development:

- due to the relatively low stiffness of the force actuators ( essentially pneumatic actuators) the stiffness of the hardpoints dominates the solid body resonant frequency of the mirror support, so it is necessary to guarantee a very high axial stiffness of each hardpoint in order to increase as much as possible the eigenfrequencies of the primary mirror.
- to maximize the resistance to wind forces bending the mirror and to reach the higher eigenfrequency the hardpoints layout has been studied carefully: they have been located equally spaced around the mirror ( 120 degrees ) at roughly 70% of the radius from the centre of the mirror. The hardpoints also need to be supported on stiff locations in the mirror cell structure to avoid loosing rigidity on their connection points. The final layout of the hardpoints has essentially a hexapod configuration as shown in FIG. 1.
- assuming a pressure failure in the force actuator system it is necessary to avoid that the mirror weight is placing suddenly on the hardpoints, reaching too high stress value on the mirror backplate connections, before the mirror was hold by the rubber earthquake pads: the hardpoints are then required to breakaway from their normal positions when the force on them exceeds a fixed value both push and pull direction.
- in order to be able to align the primary mirror on the optical path of the telescope the position actuators need also to change and to measure their own length very precisely. To do that they need to be motorized and encoded to have all the six dregrees of freedom of the mirror under remote control respect to the cell: this adjustment is needed to compensate the primary mirror position to keep the wide field focal plane aligned. perpendicular to the axis of the instrument rotator. Also look-up tables could be used to adjust the mirror position compensating the deflection of the mirror cell and of the telescope structure.

- the hardpoints have not to apply any stray forces and moments on the backplate of the mirror due to their own weight: in other words they need to be balanced by a simple but effective device able to remove, in all directions, these loads. A simple and used mechanism is an astatic lever applying counterbalance forces to zeroize these loads on the connection points of the mirror backplate.

All these requirements have been translated in technical requirements and solutions according to the scientific target of the telescope and to the M1 honeycomb mirror mass. The current conceptual approach is valid for every honeycomb mirror starting from the 3.5 m up to the 8.4 m of the LBT (Large Binocular Telescope now under final design), but, for each telescope, we need different specification. Talking about the MMT Conversion the new honeycomb mirror, substituting the previous six 1.8 m mirrors, is currently under final optic processing and the new telescope configuration will be ready for the first-light within 1997. The main 6.5 m mirror data have been summarised in TAB. 1.

According to the main features of the 'fixed points', in TAB. 2 we summarised the technical specification required for the hardpoint mechanisms employed in the MMT Conversion project.

**TAB.1 - 6.5 m mirror main data**

| | | | |
|---|---|---|---|
| Outer Faceplate Diameter: | 6512 mm | Clear Optical Aperture: | 6502 mm |
| Faceplate Hole Diameter: | 889 mm | Focal length (F/1.25, parabola): | 8128 mm |
| Outer thickness: | 711.2 mm | Approximate Mass: | 8249 kg |
| Inner thickness: | 391.2 mm | Material: Borosilicate Honeycomb (density 2.24 kg/dm$^3$) | |

## 3. DESCRIPTION OF THE HARDPOINT MECHANISMS

The main function of the primary mirror cell of the MMT Conversion, directly connected to the telescope structure, is to support the M1 mirror, as described in the previous paragraphs, and also to absolve other functions in order to support the Cassegrain instrumentation, supplying accurate thermal control of the borosilicate honeycomb mirror, acting as vacuum shell during the aluminising phase, providing maintenance access to the mirror support mechanisms. As easily comprehensible all these capabilities need many other devices and mechanisms located in the mirror cell volume where the space available for the hardpoints becomes not very large. The maximum axial dimension open to each hardpoint is then 900 mm starting from the bottom of the mirror cell up to the backplate of the M1 mirror. This further restriction requires the designer to optimize all the mechanisms both from a functional point of view and from a dimensional one. According to the technical requirements of the actuator described in paragraph 2 we can then divided the hardpoint in five main sub-assemblies referring to FIG. 2.

### 3.1 End-supports and Fast Connections

In order to decouple the body of the hardpoint from the mirror cell structure, on the lower side, and the body of the hardpoint from the M1 mirror backplate, on the upper side, we introduced, on both ends of the hardpoints, a flexural joint with two thin foils at 90 degrees each other (see FIG. 3). These foils have to meet the following requirements:

- having an high axial stiffness;
- having a low stress due to the maximum bending load;
- reducing the stray moments due to the maximum bending load.

For a constant thickness foil (as in our case) this means:

$(E\, s\, a) / l > K_{min}$ ;  $(E\, s\, \alpha) / 2\, l < \sigma_{all}$ ;  $(E\, s\, a\, \alpha) / 12\, l < C_{max}$

| where: | | | | | | |
|---|---|---|---|---|---|---|
| $E$ | = | Young modulus | [ N/mm$^2$ ] | $K_{min}$ | = | axial stiffness [ N/mm ] |
| $s$ | = | thickness of the foil | [ mm ] | $C_{max}$ | = | maximum stray moment [ N mm ] |
| $l$ | = | length of the foil | [ mm ] | $\sigma_{all}$ | = | maximum allowable stress [ N/mm$^2$ ] |
| $a$ | = | width of the foil | [ mm ] | | | |

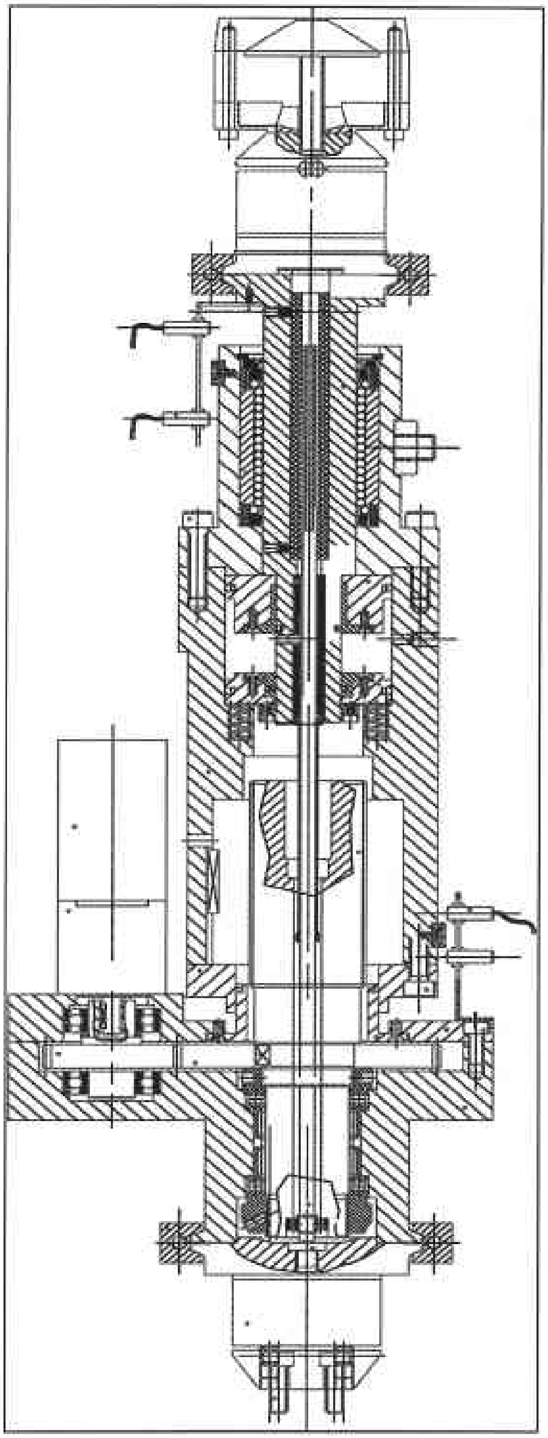

**FIG.2 Position Actuator Assembly**

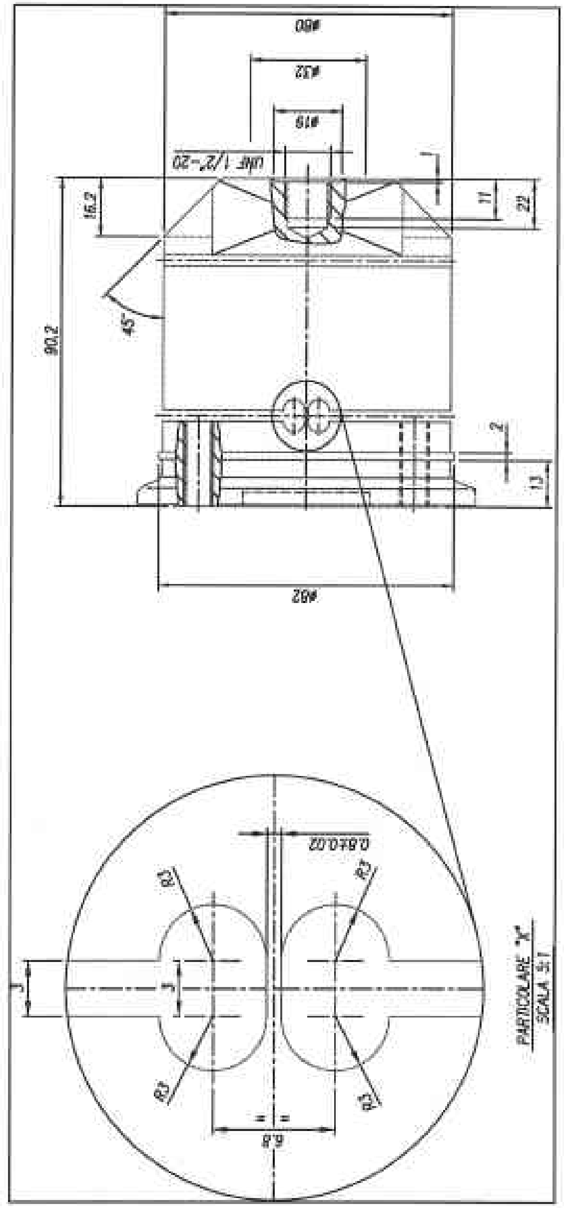

**FIG.3 - Flexural Joint**

assuming the material data for steel as E = 206,000 N/mm$^2$ and $\sigma_{all}$= 400 N/mm$^2$ we can fix the other geometric parameters as depicted in FIG. 3 (see detail). These flexural joints are directly bolded to the mirror cell and to the M1 mirror backplate (by a glass wedge interface) to guarantee precise references with high stiffness to the hardpoints. In order to remove quickly the hardpoints from their locations (for maintenance or other purposes) two fast connections have been introduced, one each side of the hardpoint. They are essentially composed with two flanges V shaped well tighten by two semi-anular collars: if both frontal and V surfaces of the connections are very accurate machined, with strict geometric and dimensional tolerances and grounded finishing, we can avoid to loose axial stiffness in these flanges (see FIG.4). This is a general rule for all the frontal connections in the hardpoint components: anywhere we have two flat surfaces, connected in a frontal way, we must guarantee very accurate contacts between them in order to maintain high the total axial stiffness as required in the specifications. Of course the cost of the components increases but, as we experimental verified, this is mandatory to realize a stiff actuator.

### 3.2 Motor Drive and Roller Screw

One of the positioning capability of the hardpoint, shown in TAB.2, is the axial motion required (range: + / - 10 mm) to align the M1 mirror by a remote control. We need an expandable device with two essential features: high axial stiffness and very accurate precision for the axial movement.

| TAB.2 - HARDPOINT SPECIFICATIONS FOR 6.5m HONEYCOMB MIRRORS | | |
|---|---|---|
| Number of hardpoints: 6, to constrain solid body motions | | |
| Support Stiffness (not including the mirror or the cell) | : | 80,000 N/mm |
| Force Measurement | range : | -300 N to +300 N |
| | resolution: | 0.5 N |
| | overload survival: | 3000 N |
| Force Breakaway: | | |
| maintaining the above stiffness for: | < 300 N | |
| maximum breakaway force: | < 3000 N | |
| Breakaway Travel: | +/- 17 mm along mirror optical axis | |
| | +/- 17 mm lateral / radial | |
| Unpredictable Stray Forces Applied by the Hardpoints to the Mirror: | | |
| axial force: | < 2 N | |
| lateral force: | < 10 N | |
| moment at the connections: | < 8,000 N mm | |
| Positioning Capability: | | |
| axial and lateral motion of the mirror | (range): | +/- 7 mm |
| along axis of hardpoint | (range): | +/- 10 mm |
| resolution | : | ~ 1 μm |
| linearity | : | ~ 1% |
| velocity | : | < 30 μm/s |
| control bandwidth | : | < 1 Hz |

After the M1 mirror alignment the hardpoints do not have to change their length for any reason and the motor must be switch off. In order to minimize the friction and to avoid any stick-slip effect, as usually we have with a traditional trapezoidal thread screw, we choose a preloaded roller screw with the following data:

| TAB.3 Roller Screw Data | |
|---|---|
| Type: | Rollvis RVR 250/63.2.R |
| Threaded length: | 130 mm |
| Pitch: | 2 mm |
| Precision: | 12 μm |
| Axial travel: | 12 mm |
| Number of rollers: | 12 |
| Dynamic load: | 114.3 kN |
| Static load: | 207.1 kN |
| Loadless Torque: | 1 N m |
| Axial Stiffness: | $1.5\ 10^{6}$ N/mm |
| Factory Pre-load: | 1250 N |

The Roller Screw is driven by a Step Motor ( 1.8 °/step) and a Gearheads with 50:1 reduction ratio, a further reduction is realized by two gears 2:1. The total reduction Step Motor / Roller Screw is then 100:1. With these performances the maximum axial resolution achievable, considering to avoid any backlash in the gears, is then 0.1 μm and to satisfy the 30 μm/s required for the axial velocity we need to drive the motor at 90 RPM. Further, the static torque of the motor is 0.85 Nm and the dynamic one is 0.15 Nm, corresponding to axial forces of 1,600 kN and 470 kN: these values are oversize for the loads on the hardpoint and a limiting device is then required.

### 3.3 Axial length measurement system

The location of the M1 mirror in the space is determined by the knowledge of the length of each actuator ( six actuators for six degrees of freedom ). The most reliable and cheaper solution applied in this case is a LVDT (Linear Voltage Displacement Trasducer) axially located in the actuator. Since it is important in our case to measure the total length of the actuator, considering also any possible local shortening effect due to deformations of the actuator components (the required axial length resolution is 1 μm), the external stainless steel case is rigidly fixed to the upper part of the hardpoint while the internal core is integral to a long rod directly connected to the lower part of the actuator. This solution is possible according to the restricted dimensions of the case ( 168 mm length and 22 mm in diameter). In this approach we are able to measure in real time the total length of the actuator according to the LVDT sensitivity (0.39 mV / V per 0.001"): having and input nominal voltage of 3 V we need to read about 0.046 mV for the resolution required, possible with a low noise amplifier. About the linearity we have 0.25% full scale (16 mm) that means 0.16% along 10 mm axially required.

### 3.4 Load cell and anti-torsion bush

As mentioned in the introduction the instant forces applied on each position-actuator, by external stray loads, have to be sense in order to control the force actuators able to apply a reaction forces system to correct the optical surface of the mirror and to unload the hardpoints. Also the load cell introduced in the actuator has not to decrease the stiffness of the mechanics and has to absolve to the required specification. As depicted in FIG. 5 the used load cell is located axially on the external side of the upper joint to avoid any lateral load on it. In the experimental testing phase we noted a relevant sensitivity of the load cell to the torsion torques depending on the orientation of it; then we have been forced to introduce a further flange to decoupling any torsion effect. The load cell data have been summarized on TAB.3. With a proper amplifier reading the load cell output voltage ( 1mV / V) we are able to satisfy the technical specifications in TAB. 2.

| | |
|---|---|
| Load Capacity: | 1000 lb (4450 N) |
| Non-Linearity (max): | 0.05% F.S. |
| Hysterisis (max): | 0.05% F.S. |
| Non-Repeatability (max): | 0.02% F.S. |
| Deflection at Capacity: | 0.0005" (12.7μm) |
| Temperature Range (usable): | -65 °F to +200 °F (-54 °C to 93 °C) |
| Temperature Effect on Zero(max): | 0.001% F.S./ °F |
| Temperature Effect on Output(max): | 0.002% Reading / °F |

**TAB.3 - Load Cell Specifications**

### 3.5 Breakaway device and anti-rotation bush

The most important item in the hardpoint specs is probably the guarantee to have its length decreased or increased when an external load greater than 3000 N is applied on it (this value comes from the Von Mises stress limit evaluated by many finite element analyses carried out on the M1 mirror structure). If this requirement had not to be satisfied the forces applied on the M1 mirror backplate could effect a huge increasing of the local stress in the borosilicate with serious consequences for the mirror integrity. As detailed in FIG. 6 the decoupling joint has been realized as air-pressured cell continuously supplied by the external pneumatic net ( 4 bar) and with a check valve at the cell input. On one side of the joint four springs avoid any backlash in the joint and apply to the flange about 1000 N preload.

That means that an axial force of 1000 N is needed, at least, to start the decoupling phase: after that, and according to the acting section of the pneumatic cell, a force greater than 2000 N is requested to continue the decoupling of the joint. Each hardpoint has been tested and qualified especially for this requirement. In order to assure the correct functioning of the decoupling joint, in any condition, and to prevent rusting, we employed stainless steel for the external liner and brass alloy for the mobile parts. Also low friction polytetrafluoroethylene material has been used for the circular seals, then everything well lubricated with special vacuum grease.

To support the internal shaft connected to the pneumatic decoupler and to the upper joint of the actuator, a very accurate ball bush is located inside the hardpoint body: on the external cylindrical surface of this bush a radial screw prevents any rotation.