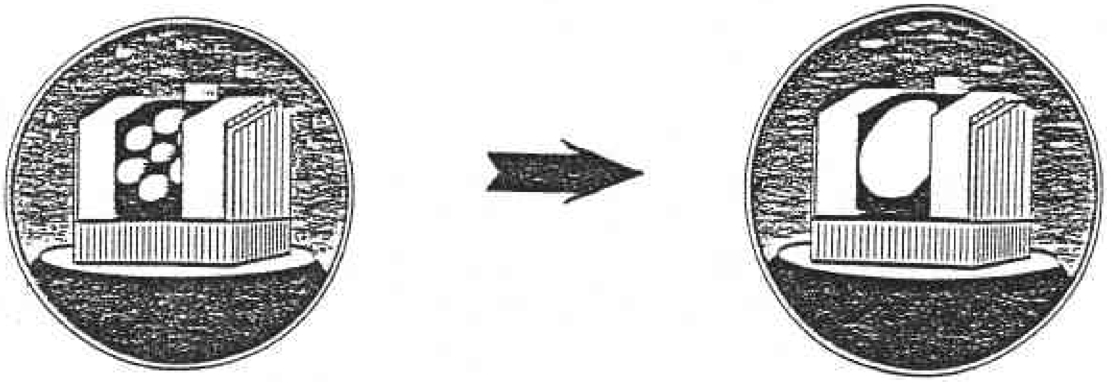
6.5 METER TELESCOPE



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

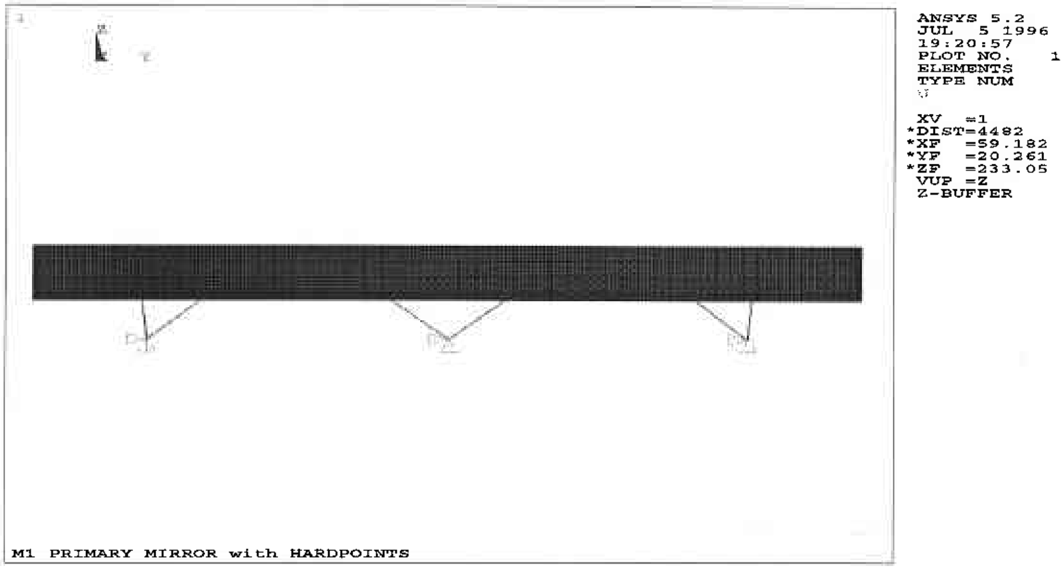

FIG.1 - M1 MIRROR FEA MODEL

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]	σ_{all}	=	maximum allowable stress	[N/mm^2]
a	=	width of the foil	[mm]				

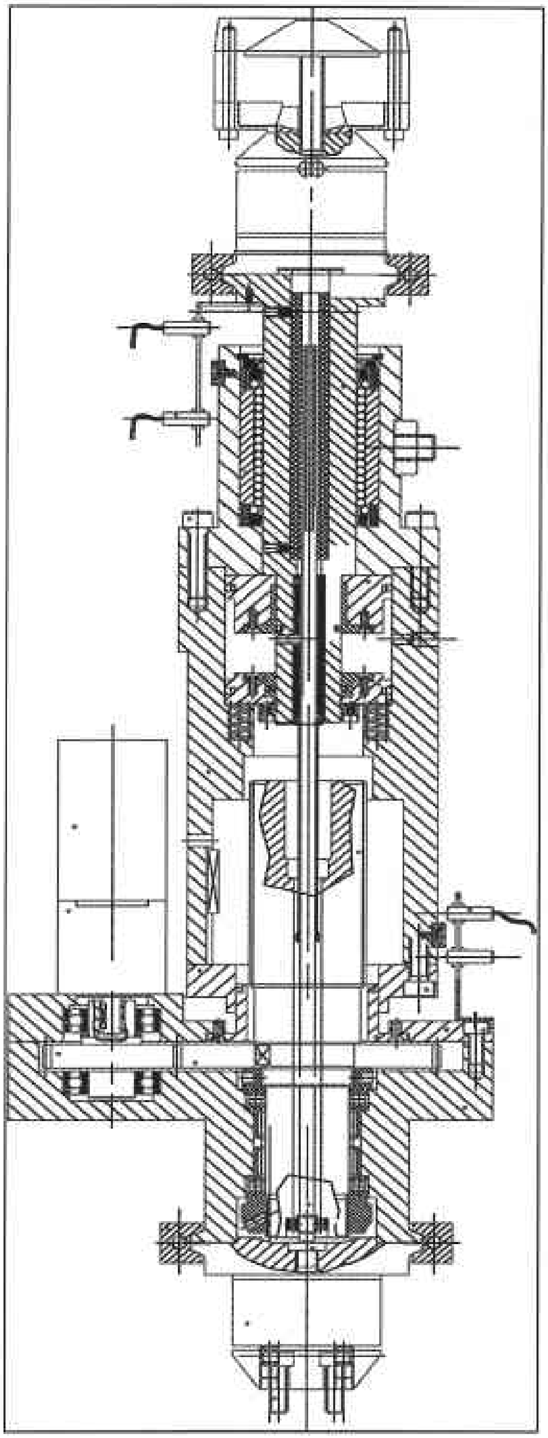

FIG.2 Position Actuator Assembly

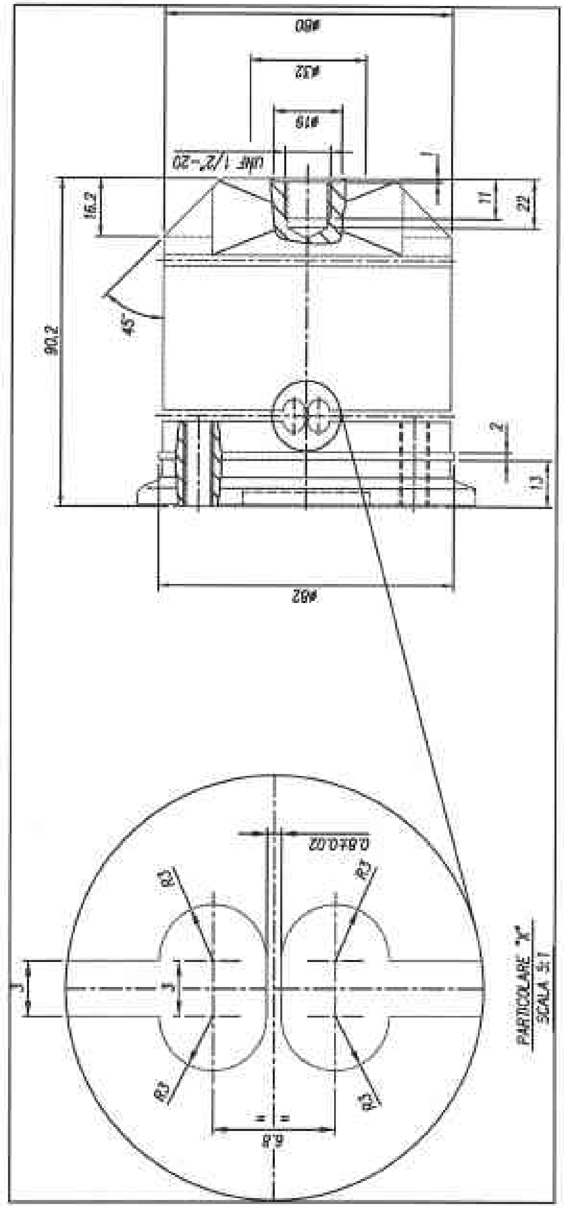

FIG.3 - Flexural Joint

assuming the material data for steel as E = 206,000 N/mm^2 and σ_{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.

4. ACTUATOR EXPERIMENTAL TESTS

4.1 Test-bench and set-up of the measurements

The complete final tests of the actuators control system will be performed only in the actual M1 mirror cell, but extended tests on all the actuators have been carried out in the workshop and in the mechanical lab. Each actuator has been tested in horizontal position according to the invariance of its performances by the gravity direction and to the easiness of applying loads both in compression and in tensile direction with this arrangement. A very stiff test-bench has been designed and manufactured for this tests to be able to check the mechanical properties of the actuator without introducing errors due to the elasticity of the bench. As shown in FIG. 7 two lateral vertical stiff rods are connected by a solid horizontal body: on both of them the actuator under test, or a section of it, is connectable to the test-bench by simply bolted joints. On the central area of the bench a moving trolley is located to fix the other end of the actuator. The trolley is supported by very low friction linear ball guides to allow the hardpoint axial movement without introducing relevant friction forces. To test the hardpoint by low load the trolley is supported by flexural foils to avoid any friction effect. A scheme of the test-bench with the hardpoint under test is depicted in FIG. 7. By the pulleys, on the external sides of the test-bench, we can apply on the actuator tensile or compressive static forces starting from few newtons up to hundred of newtows to test the breakaway device.

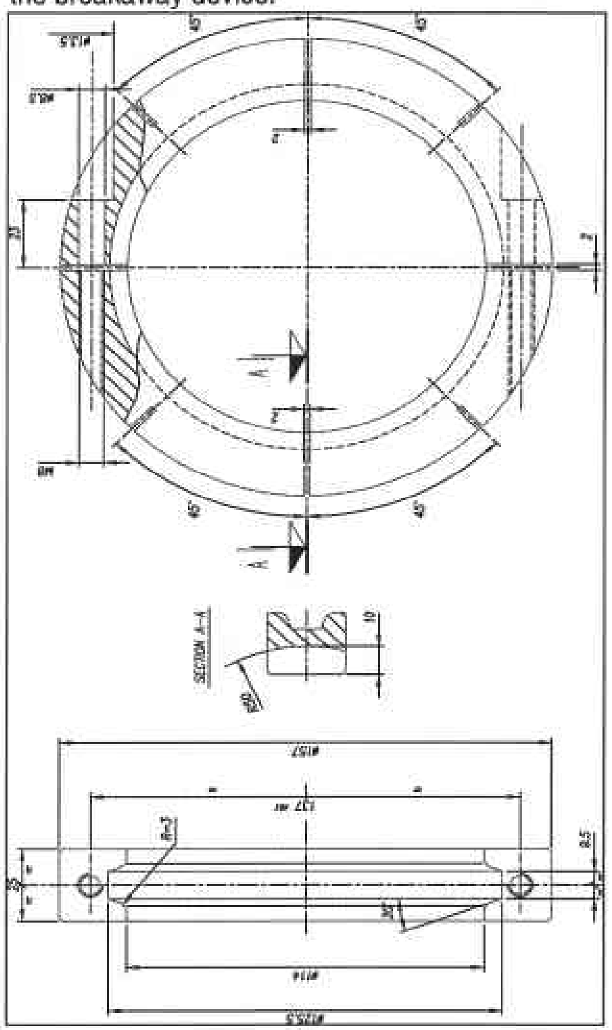

FIG.4 - V Fast Cennections

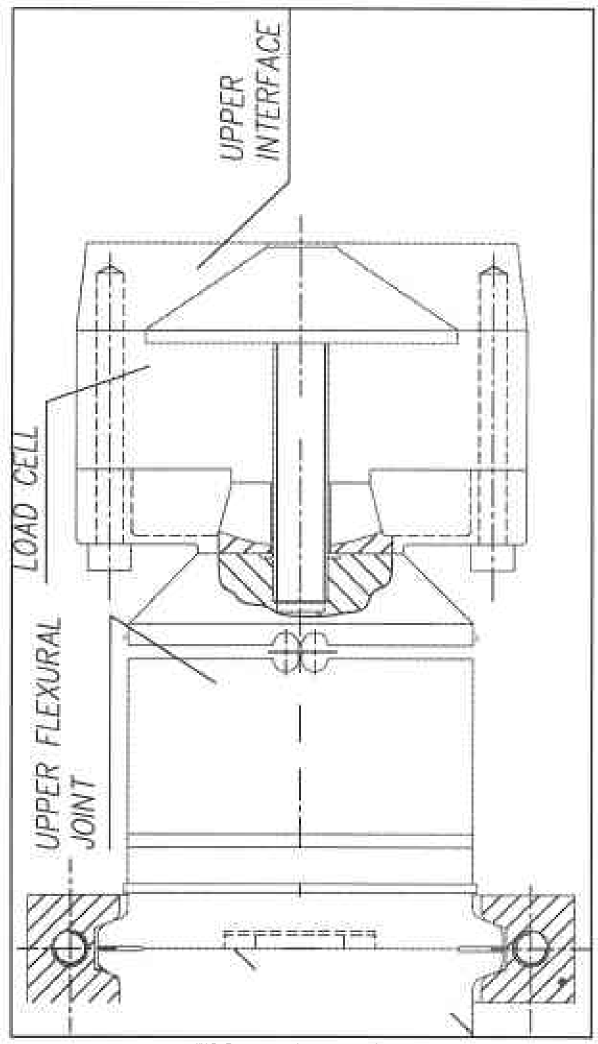

FIG. 5 - Load Cell

The hardpoint has been positioned and tested with all the components assembled as it will be located in the mirror cell. Also the motor, the load cell, the LVDT, the upper and lower flexural joints were on and correctly operative.

With this set-up the loads applied on the actuator have been measured directly by the load cell, after its off-line calibration procedure. The coarse length variation produced by the motor has been sensed by the LVDT with the proper resolution. In order to check carefully the axial stiffness of the hardpoint, in the required load range, we needed to use an external sensor measuring accurate length variation of the actuator separately from the stiffness of the test-bench. To perform this measurement we applied, on both lateral sides of the central trolley, a stiff steel bar on which we fixed, by a magnetic base, an inductive contact sensor: we applied one sensor each side of the actuator to minimize the errors due to any unsymmetry of the load direction. By this inductive sensor we were able to measure the total variation of the length of the actuator under load with a resolution up to 0.01 μm. During the tests we had to paid attention basically on two main effects: the thermal deviation of the measurements due to all the components involved in the mechanical chain and the complicated mechanical zero set-point of the inductive sensor at this so low resolution.

4.2 The Acceptance tests of the actuators

Before manufacturing the final version of the actuators we tested a prototype on which we studied different solutions for some of the above mechanisms, and on which we performed on each sub-assembly in order to measure, for eachone, stiffness, backlash and the other main parameters. After this optimization phase we analyzed the performances of the whole hardpoint. The final acceptance tests will be then perform on each hardpoint completely assembled. In Appendix 1 the whole sequence for tests is summarized.
From a mechanical point of view we had to solve an indesiderable behaviour of the load cell after the prototype tests. This kind of device is very stiff and not very heavy, so we can use it directly axially mounted, but it is also necessary to have no stray effect due to non-axial loads. We tested the load cell applying bending, lateral and torsion loads and we verified a total invariance of the measured load except for the latter one. To avoid this effect we introduced a torsion decoupling flange between the upper joint interface and the load cell maintaining the same axial stiffness.

4.3 Experimental Test Results

The final acceptance tests on the M1 mirror actuators for the MMT Conversion project shown a good reproducibility of all the features required in the technical specifications. In APPENDIX 2 we shown in detail all the main final measurements carried out on the six hardpoints. We can note essentially the following comments:
- due to the restricted space in the M1 mirror volume we needed to design a relatively small actuator having an axial stiffness as much as possible with reduced sections; to solve this problem we needed to have very accurated mechanical tolerances and well ground surfaces especially for all the frontal interfaces from which depends basically the high axial stiffness;
- the actuators shown a good reproducibility for the compressive and tensile stiffness especially at low load, around which, they usually work; around zero-load we don't detect any slipping effect and the load inversion has been regularly measured;
- as depicted in APPENDIX 2 - FIG. 8 we have a maximum deviation (about 30%) of the stiffness values measured at low load between the hardpoint n.2 and the n.1/n.5; although all of them are stiff enough to well satisfy the requirements (minimum required 80,000 N/mm), this deviation is due to the following reasons:
  - loading the hardpoint by few newtons we measure axial displacements of hundredth of micron, that is the limit capability of the inductive sensor resolution. In other word at very low load (few newtons) we are in the sensor noise with a high increasing of the error. To better analyze this range of load we are developing a capacitive sensor with very high resolution (tens of nanometers) able to measure also dynamic load with a bandwidth of several kHz.
  - the mechanical connections in the actuator needed very high tolerance and very precise clamping. Especially for the end joints and for the collar connections we need high tightening torque in a strict tolerance range: small different values define different axial stiffness.
- all the other requirements are well satisfied and surely in the technical specification ranges.

5. CONCLUSION

The development of the prototype of this actuator and then the manufacturing of the six hardpoints for the M1 6.5 m mirror of the MMT Conversion Project has been a useful test for solving several mechanical problems both in

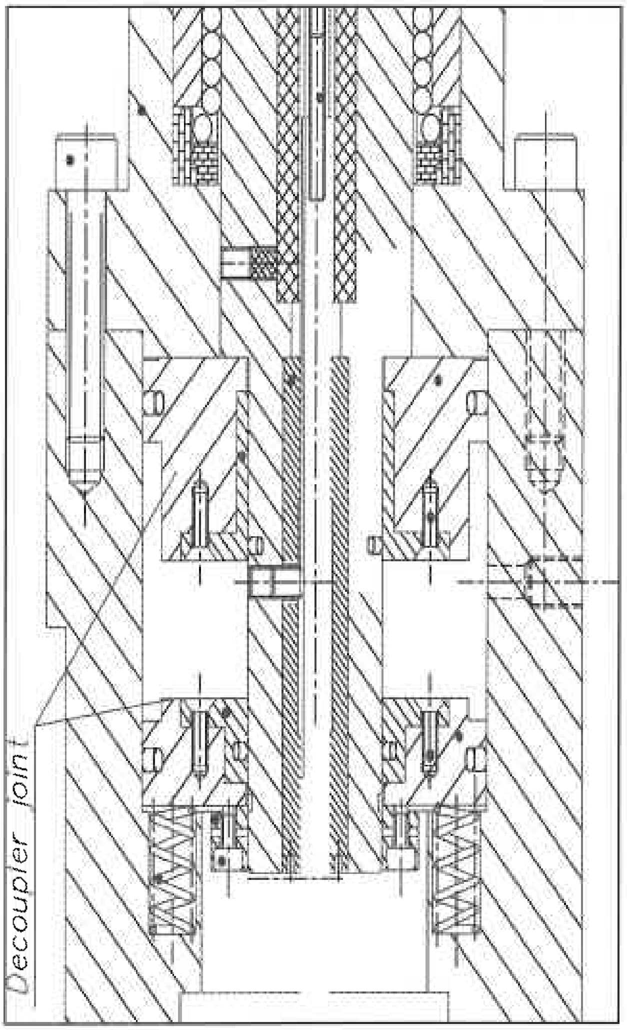

FIG. 6 - Pneumatic Decoupler

the design and in the measuring phase. According to the high axial stiffness required for this actuator, usually working at low load, we tried to reach the best results restraining as much as possible the cost of the components. Also for the test measurements we used relatively cheap sensors but able to guarantee the required resolution in the displacement measurements. The current mechanical solution performs the best static result that we are able to reach in this configuration. In the future the next generation of 8 m M1 mirrors for ground-based telescope (as LBT), where the mass of the mirror grows up to 15 tons, need to have higher axial stiffness for these position-actuators in order to maintain up the natural frequencies of the M1 mirror located in the mirror cell. To do that we need to introduce in the actuator an active control system able to increase its axial static stiffness at least by a factor of three.

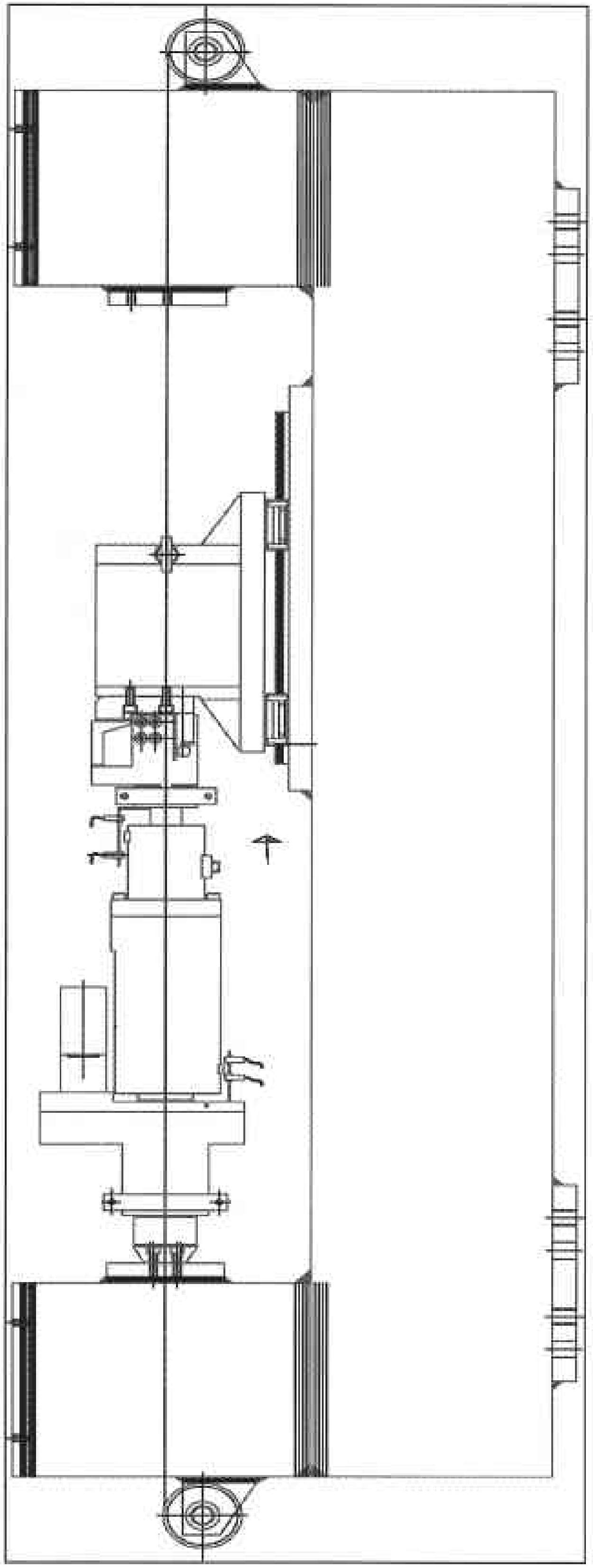

FIG.7 - Test-bench and hardpoint

We are now in the process to test a new actuator prototype with piezoceramic elements able to control the dynamic behaviour of the 8m M1 mirrors in the range 0 - 50 Hz where the stray loads applied to the mirror can have destructive effect on the quality of the optical images, coming from the universe, especially for interferometer applications.

6. ACKNOWLEDGEMENT

The authors wish to thank all the people, coming from different Institutes and Observatories, involved in the design reviews of the actuator in last two year. Special thanks to Mr. Tomelleri, who designed most of the technical solutions applied on the actuator, for his competence and patience in solving the innumerable requests of the project team.

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<u>APPENDIX 1</u>

a) PRELIMINARY INDIVIDUAL GROUP TESTS.

a1) Upper Foiled Join:

- without load cell: axial stiffness of each foil; axial stiffness of the whole join; flexural stiffness of the foil
- with load cell: axial stiffness; linearity; hysteresis; repeatability; resolution
  (temperature effects on the load cell checked in the range: -25 +35 °C)

a2) Antirotation Coupling Star 1 D.50: axial friction resistance vs. preload; radial backlash vs. preload.

a3) Pneumatic Breakaway Decoupler: axial stiffness vs air pressure; pulling and pushing maximum load

a4) Roller Screw Rollvis RVR 250/63.2.R: dynamic friction; radial and axial backlash; axial stiffness

a5) Lower Foiled Joint: axial stiffness of each foil; axial stiffness of the whole join

b) TESTS DURING ASSEMBLING GROUPS .

b1) Lower Foil Joint + Motor And Gear Drive: axial stiffness; combined bearing axial stiffness

b2) Lower Foil Joint + Motor And Gear Drive + Roller Screw: axial stiffness; gear drive backlash; axial travel resolution

b3) B2 + PNEUMATIC DECOUPLER: axial stiffness; breakaway maximum load

b4) B3 + ANTIROTATION COUPLING: axial stiffness; axial backlash

b5) B4 + UPPER FOIL JOINT (whole hardpoint: see item D)

C) MOTOR AND LVDT TESTS
Separate tests will be performed on the motor, electronic drivers and LVDT device before assembling them on the hardpoint and taking into account performances depending also on temperature in the range -25 +35 °C.

D) FINAL TESTS ON THE WHOLE HARDPOINT ASSEMBLED (mechanical and electronic tests): total axial stiffness; total axial friction resistance; axial positioning capability: resolution, linearity, velocity; breakaway travel; force breakaway; force measurement

APPENDIX 2
MAIN TEST RESULTS

1. AXIAL STIFFNESS (Compressed Air = 4 bar)

- Compression load

Force [N] δ [μm] Ka[N/μm]

	N.1		N.2		N.3		N.4		N.5		N.6	
92.2	0.9	102.4	0.68	135.6	0.8	115.3	0.8	115.3	0.9	102.4	0.8	115.3
141.3	1.4	100.9	1.1	128.5	1.3	108.7	1.3	108.7	1.4	100.9	1.2	113.8
190.3	2.0	95.15	1.6	118.9	1.75	108.9	1.8	105.7	2.0	95.2	1.7	111.9
288.4	3.2	90.12	2.8	103.0	2.8	103.0	2.8	103.0	3.2	90.1	2.8	103.0

- Tensile load

Force [N] δ [μm] Ka [N/μm]

	N.1		N.2		N.3		N.4		N.5		N.6	
92.2	0.9	102.4	0.8	135.5	0.8	115.3	0.8	115.3	0.9	102.4	0.8	115.3
141.3	1.4	100.9	1.2	126.7	1.3	108.7	1.3	108.7	1.5	94.2	1.4	100.9
190.3	2.0	95.15	1.6	118.9	1.8	105.7	1.78	106.9	2.0	95.2	1.9	100.2
288.4	3.2	90.12	2.4	106.8	2.7	106.8	2.8	103.0	3.1	93.2	2.9	99.4

2. HARDPOINT FORCE BREAKAWAY (Compressed Air = 4 bar)

- Compression load (measured by its load cell)
-

Breakaway begins at F1 [N]	1040	1040	1371	1195	1094	1232
" ends at F2 [N]	2145	2145	2348	2515	2364	2405

- Tensile load (measured by the load cell)
-

Breakaway begins at F1 [N]	1500	1500	1724	1804	1565	1807
" ends at F2 [N]	1700	1700	2311	2342	2326	2347

3. FORCE MEASUREMENTS RESOLUTION: < 0.5 N

4. STRAY FORCES APPLIED BY HARDPOINT:

4.1 Disturbance by lateral force: absent
4.2 Friction Torque of the Roller Screw: < 2 N m
4.3 Maximum moment at the connection: 3500 N mm

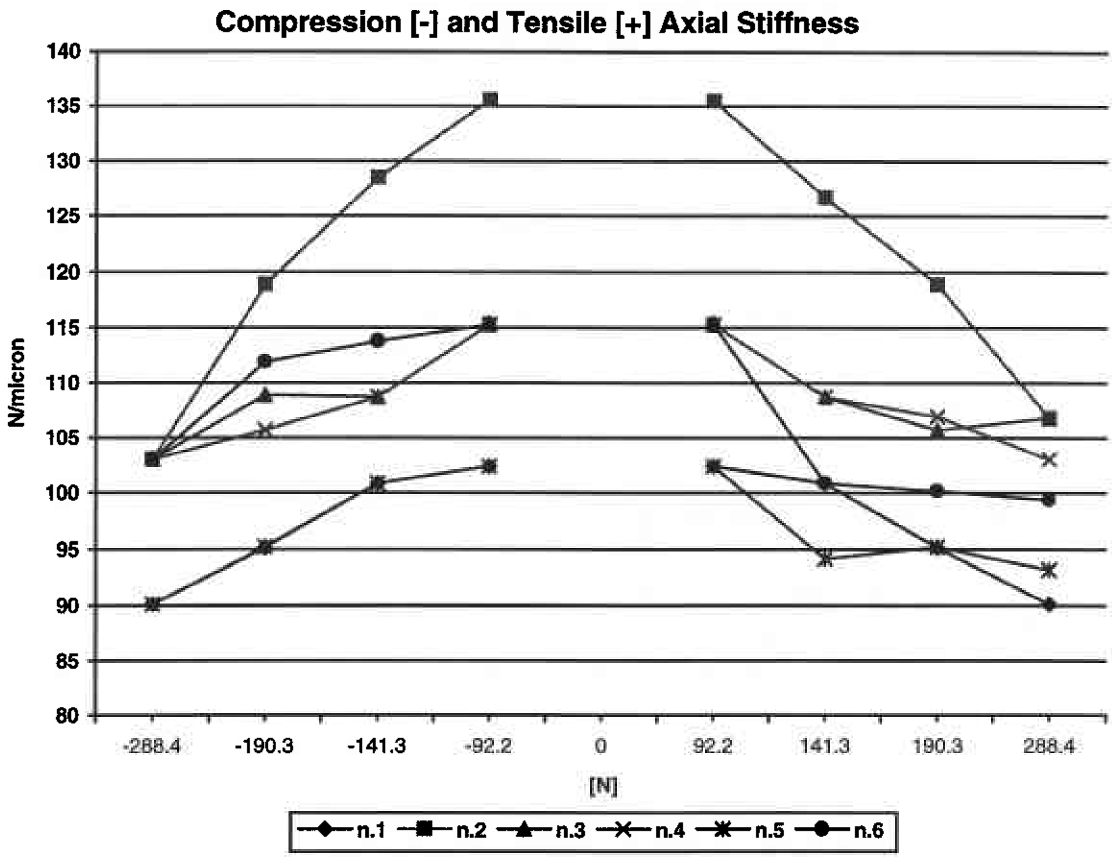

FIG. 8 - Comp. and Tensile Stiffness

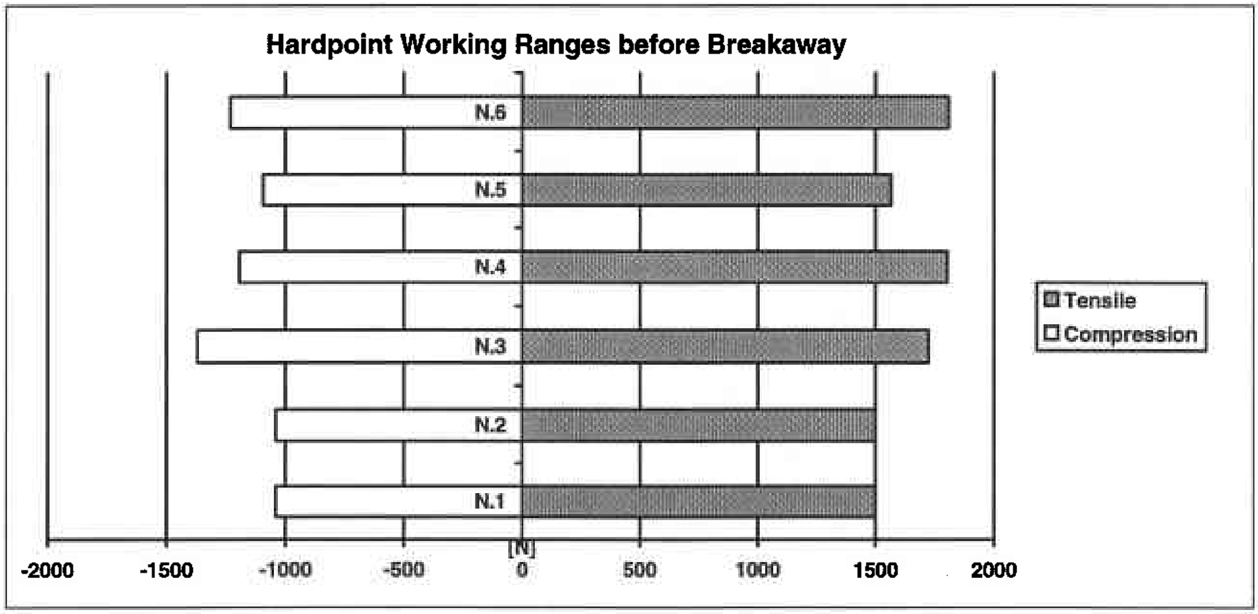

FIG.9 - Force ranges before Breakaway