Gemini Primary Mirror Cell Design

E. W. Huang
Gemini Telescopes Project, 950 N. Cherry Ave., Tucson AZ 85719

Gemini Preprint #10

"Gemini Primary Mirror Cell Design"

Eugene W. Huang

Gemini 8-m Telescope Project
P.O.Box 26732, Tucson, AZ 85726

ABSTRACT

To minimize the wind buffeting effect on the primary mirror figure, the Gemini primary mirror cell is designed to provide additional mirror stiffness by coupling the mirror to the cell structure through a six-zone hydraulic support system. Therefore the cell structure is designed as though it were a light weight mirror for minimum top surface distortion. This paper describes the design requirements, the design features, and the detail predicted performance of this cell structure, particularly the effects on the primary mirror figure.

As the cell structure supports the primary mirror with a six-zone hydraulic system, the mirror is coupled to the cell structure with three degree of freedom overconstraints. These overconstraints induces the possible distortion on the mirror figure due to the cell deformation. This paper presents a solution to eliminate this effect by supporting the mirror cell on the telescope structure through four bipods. The locations of the bipods are so arranged that the cell deformation will not distort the mirror figure as the telescope rotates from zenith to horizon pointing.

Keywords: mirror support, mirror cell, cell structure, telescope structure, wind buffeting.

1. INTRODUCTION

When a mirror is supported on a mirror cell, it is desirable to support the mirror on the cell structure through a kinematic support system for avoiding the mirror being distorted by the cell deformation. However, the primary mirror selected for the Gemini 8-m telescope is a thin meniscus mirror having 8.1 meter diameter and 0.2 meter thick. When such a mirror is supported on a kinematic hydraulic whiffletree, the mirror will deform significantly under the wind load which is flushing over the mirror for eliminating the mirror seeing.[1,2] To minimize this wind buffeting effect, the Gemini primary mirror cell is designed to provide additional mirror stiffness by coupling the mirror to the cell structure through a six-zone hydraulic support system.

As the cell structure supports the primary mirror with a six-zone hydraulic system, the mirror is coupled to the cell structure with three degree of freedom overconstraints. This increases the mirror's two astigmatic and one trefoil modal frequencies from 15.2, 15.2, and 40.5 Hz to 30.7, 30.7, and 48.5 Hz respectively[2], and thus reduces the wind buffeting effect on the primary mirror by a factor of four. However, these overconstraints induce distortions on the mirror figure due to the cell deformation. A lots of effort have been made to minimize this effect in the course of the cell design. At the end a solution is found to eliminate this effect by supporting the mirror cell on the telescope structure through four bipods. The locations of the bipods are so arranged that the cell deformation will not distort the mirror figure as the telescope rotates from zenith to horizon pointing.

Besides the above innovative design features, the cell structure is designed to support a 13,000 Kg Cassegrain instrument cluster and a 4 meter long baffle. In addition the cell has a unique configuration that provides convenient personnel access ways from the telescope platform to everywhere inside the cell for component services.

The design requirements, the design features, and the detail predicted performance of this cell structure, particularly the effects on the primary mirror figure are presented in the following sections.

2. DESIGN REQUIREMENTS

The Gemini primary mirror cell structure (Figure 1) is designed to support not only the primary mirror but also the primary baffle and the Cassegrain instruments. The design requirements for each of them and other general requirements, including the environmental conditions, are described as follows:

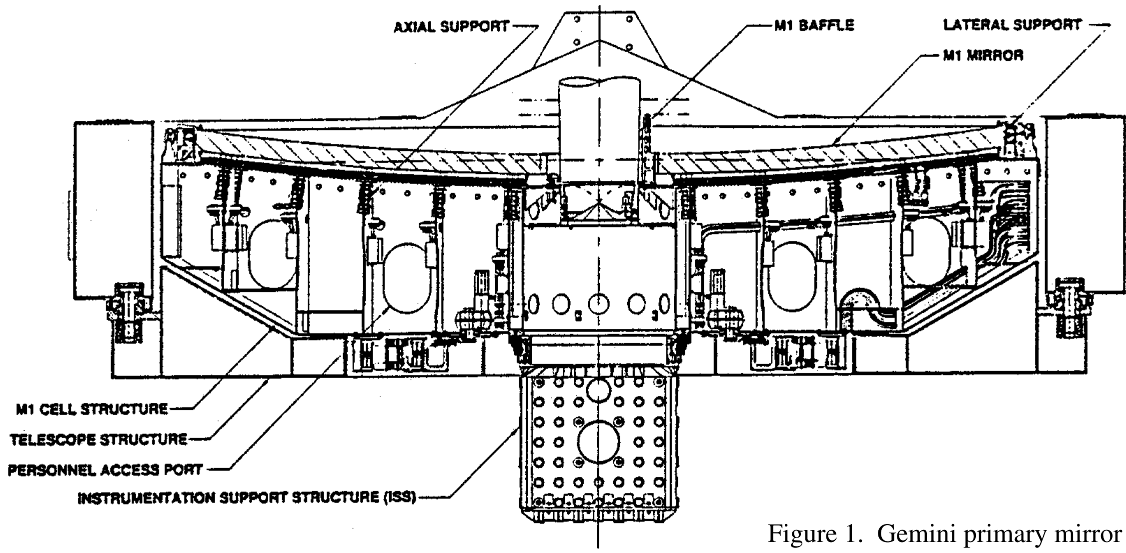
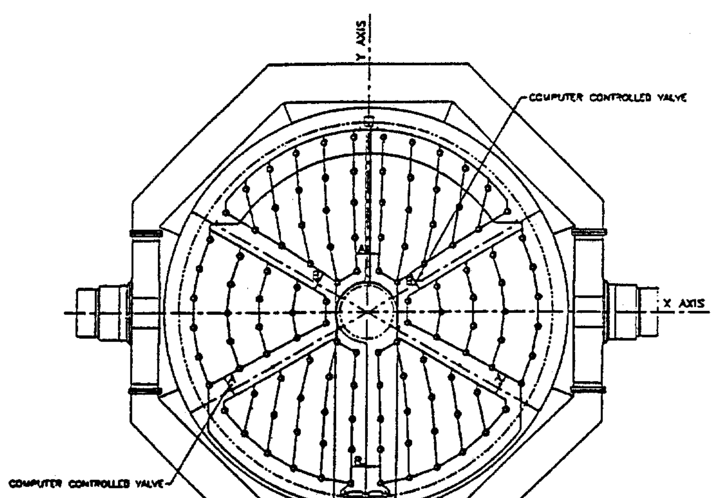


Figure 1. Gemini primary mirror assembly

2.1 Primary mirror requirements

The primary mirror is supported on the cell structure with a uniform air pressure support, 120 axial supports, and 84 lateral supports[3]. The 120 axial supports are distributed on the cell structure as shown in Figure 2. Each axial support consists of a passive hydraulic cylinder and an active pneumatic actuator. The hydraulic cylinders are so connected hydraulically that the 120 axial supports form either a 3-zone kinematic support or a 6-zone overconstrained support.



ZONE CONFIGURATION		
CONFIG	VALVE A	VALVE B
3 ZONE A	OPEN	CLOSED
3 ZONE B	CLOSED	OPEN
6 ZONE	CLOSED	CLOSED

Figure 2. A schematic of the 3-zone / 6-zone hydraulic support system

When the mirror is supported on the 3-zone kinematic support, the mirror deformation under an uneven wind is dominated by the astigmatism[2] (Figure 3). Therefore the 6-zone support is introduced to increase the stiffness of the mirror astigmatic modes and to reduce the mirror deformation. Furthermore, the analysis shows that the amount of reduction on the mirror deformation with a 6-zone support system depends on the support stiffness. The stiffer the support is, the less the mirror deforms. The relationship is shown in Figure 4. It is seen that in order to meet the 70 nm r.m.s. surface error budget the support stiffness must be higher than 7 KN/mm. The support stiffness is contributed mainly from the hydraulic cylinder and the load cell that have a combined stiffness approximate 10 KN/mm. As a goal all the structural components of the support unit including the cell structure is designed to have a stiffness at least ten times higher than this.

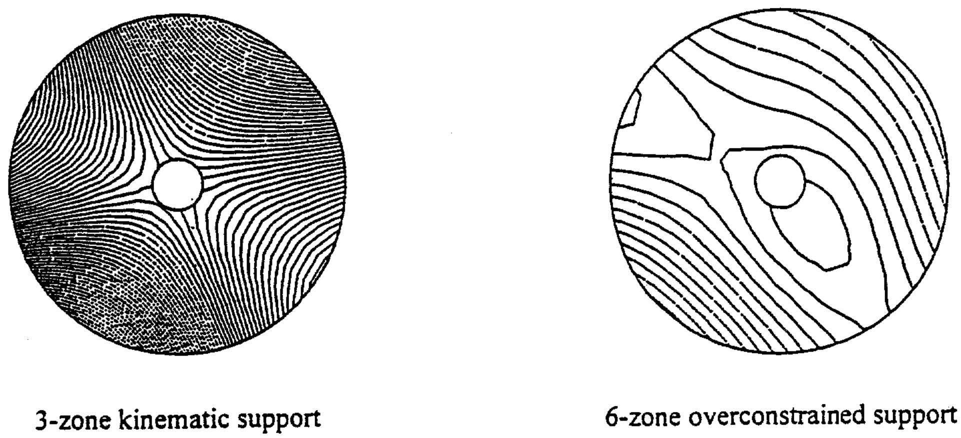


Figure 3. Deformation contour of the mirror surface under a typical uneven wind load

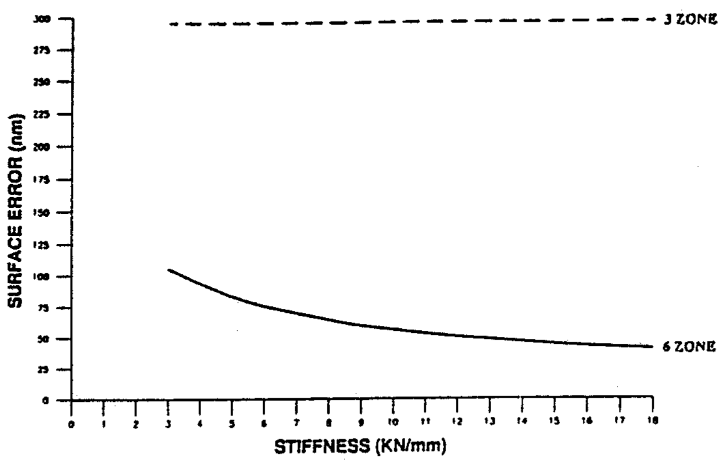


Figure 4. Effect of the support stiffness on the mirror response to uneven wind load

Based on the above analysis the cell structure is designed to the following requirements for supporting the primary mirror.

- o The first bending mode of the cell structure including the primary mirror and Cassegrain instruments shall have a frequency higher than 35 Hz.
- o The local stiffness of each axial support mount shall be larger than 100 KN/mm.
- o The effect of cell deformation on the primary mirror figure in the 6-zone hydraulic support system shall be less than 4 nm r.m.s per minute.
- o The dimensional tolerances of the support mounts shall meet the requirements as summarized in Table 1.

2.2 Cassegrain instrument requirements

The Cassegrain instruments are supported on the cell structure with an instrument support structure (ISS). The ISS rotates with respect to the cell structure with a circle bearing that attaches to the bottom inner ring of the cell structure. To maintain the optical alignment and to ensure the proper function of the bearing as recommended by the bearing manufacturer, the cell structure is designed to the following requirements:

- o The gravity rotation of the Cassegrain axis due to the cell deformation shall be less than 7 arc-seconds for a total instrument cluster weight of 13 tons.
- o The local deformation of the Cass. rotator bearing mounting surface shall be less than 0.25 mm over 90 degrees.
- o The dimensional tolerance of the bearing mounting surface shall meet the requirement as summarized in Table 1.

Table 1. Summary of Critical Dimension Tolerance Requirements

Mount	Position	Flatness	Other
Bottom Inner Ring (Cass. rot. mount)	datum	0.025 mm	0.1 mm outer diameter roundness
Top Outer Ring (lateral sup. mount)	± 1 mm	0.25 mm	parallel to datum within 0.5 mm
Top Inner Ring (baffle mount)	± 1 mm	0.5 mm	parallel to datum within 0.5 mm
Axial Support Rings (axial sup. mount)	± 1 mm	0.1 mm	tilt within ± 0.2 degree
Bipod Feet (telescope structure interface)	± 1 mm	0.1 mm	co-place within ± 1 mm

2.3 Primary baffle requirements

The primary baffle is mounted on the top inner ring of the cell structure. For proper baffle alignment the mounting ring shall have a dimensional tolerance as summarized in Table 1. Also the effect on the primary mirror figure due to the wind load acting on the baffle surface shall be less than 2 nm r.m.s.

2.4 Other general requirements

- For minimizing the histeresis of the structure, the stress in the cell structure shall be less than the precision elastic limit of the material (13,000 psi for A-36 steel).
- For telescope balance the weight of the cell structure shall be 42 − 2 tons.
- For component maintenance the cell structure must allow personnel access to everywhere inside the cell.
- For meeting the envelope requirement the size of the cell structure shall be 9 meter in diameter and 1.5 meter high at center.

2.5 Environmental conditions

The environmental conditions that are used in the cell structure design are:

a) Operating condition	
Gravity orientation	0 to 75 degree zenith angle
Ambient temperature	-5 to 20 C
Wind velocity over primary	0 to 5 meter/second
Tracking velocity-alt	0.24 degree/minute maximum
b) Earthquake load	2.7 g in X, 1.5 g in Y, 1.4 g in Z
c) Transportation load	5 g shock in all axes

3. KEY DESIGN FEATURES

For meeting all the requirements described above the cell structure is designed as through it were a light weight mirror for minimum top surface distortion. The design is shown in Figure 5. The structure is a single piece eggcrate type welded steel structure. The top surface of the cell structure provides 120 mounting rings for the primary mirror axial supports. Circumferential ribs are located under each axial support mount to ensure the local stiffness larger than 100 KN/mm. The lower end of the center hub that consists of inner and outer cylindrical shells with radial ribs provides an ideal mounting surface for the Cassegrain rotator bearing as recommended by the bearing manufacturer (Figure 6). The 65 mm thick mounting ring is selected so that the local deformation is less than 0.25 mm over 90 degrees.

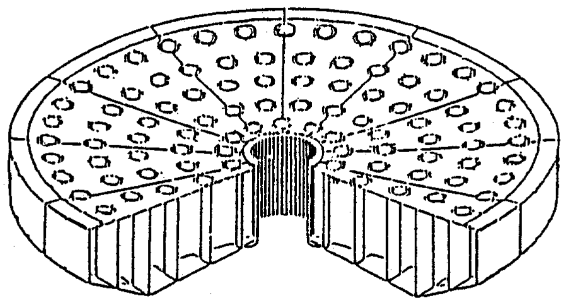
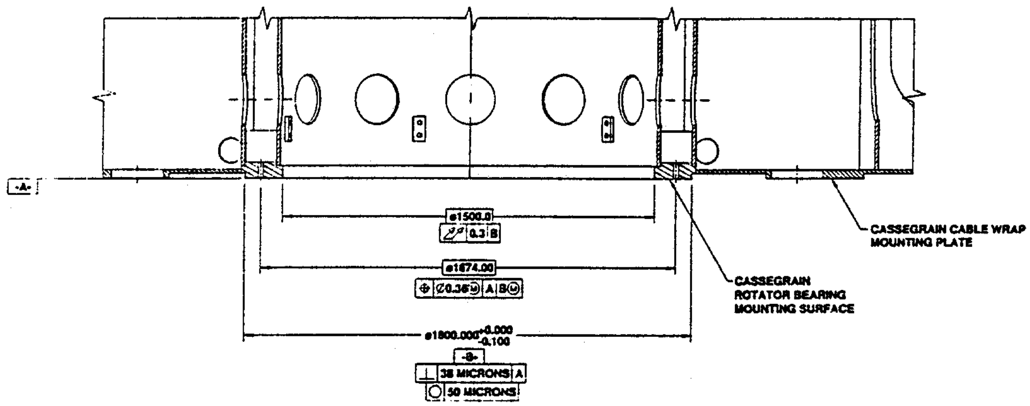


Figure 5. Gemini primary mirror cell



structure with a cut away view

Figure 6. Mounting ring for the Cassegrain rotator bearing

The radial ribs are extended from the center hub to pick up all the circumferential ribs and supported on four bipods at approximate 0.6 radius (Figure 7) to achieve the first bending frequency of the entire cell structure higher than 35 Hz. The size and location of the bipod are carefully determined so that the effect of cell deformation on the primary mirror figure is minimum for the gravity change, the thermal gradient, and the external telescope structure distortion. The details of this optimization are given in the next section.

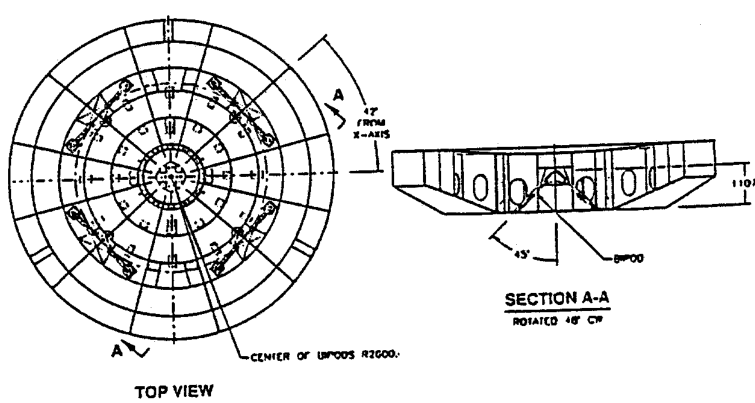


Figure 7. Cell structure cross sections and bipod locations

Finally there are 76 personnel access holes that allow maintenance staff to reach everywhere inside the cell structure for component services. Figure 8 shows the arrangement of the access holes. Here the safety issue plays an important role for the determination of the location and the size of the hole.

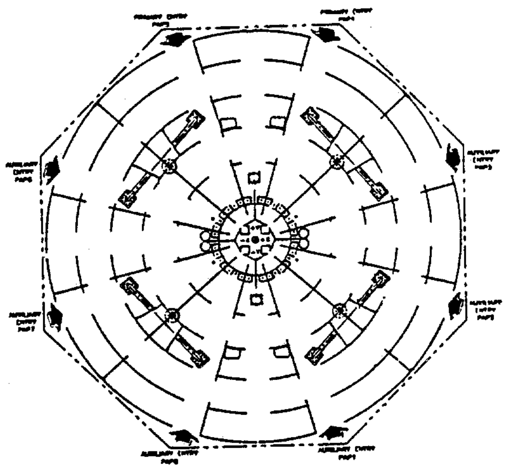


Figure 8. Cross section of cell structure showing personnel access hole arrangement

4. PREDICTED PERFORMANCE

4.1 Resonance frequency and mode shape

I-DEAS finite element analysis software[4] is used to model the cell structure including the bipods and the weight of the primary mirror and Cassegrain instruments. The predicted resonance frequency and mode shape are summarized in Table 2. The first cell rigid body mode that is dominated by the bipod stiffness is 13 Hz. The first cell bending mode is 35 Hz and its mode shape is astigmatic as shown in Figure 9.

Table 2. Resonance Frequency and Mode Shape

Mode	Frequency (Hz)	Mode Shape
1	13	rigid cell decenter
2	16	rigid cell decenter
3	18	rigid cell clocking
4	19	rigid cell tilt
5	20	rigid cell tilt
6	23	rigid cell defocus
7	35	astigmatism (Figures 9)
8	43	astigmatism
9	68	coma
10	71	coma

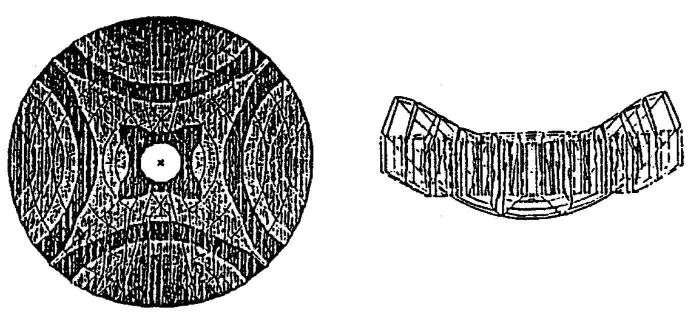


Figure 9. The mode shape of the cell structure first bending mode

4.2 Effect of cell deformation on the primary mirror figure

4.2.1 Gravity effect

When the primary is supported on the 6-zone hydraulic system, the mirror is coupled to the cell structure in two astigmatic modes and one trefoil mode (Figure 10). Because of this coupling, cell deformation will cause the pressures in the six hydraulic zones to change, producing proportional changes in the forces on the mirror. The force patterns on mirror that correspond to the three coupling modes are shown in Figure 11. Note that in each hydraulic zone the support cylinders are interconnected hydraulically. The zone average displacement, the average displacement of the support mounts in each zone, determines the force change on the mirror. Therefore it is desirable to avoid the cell deformation that produces the zone average displacement pattern as same as these three force patterns or the combination of these patterns. Efforts of locating the four bipods to avoid these deformation patterns and to minimize the distortion on the primary mirror figure are described below.

Figure 10. Three coupling modes in the 6-zone hydraulic support system

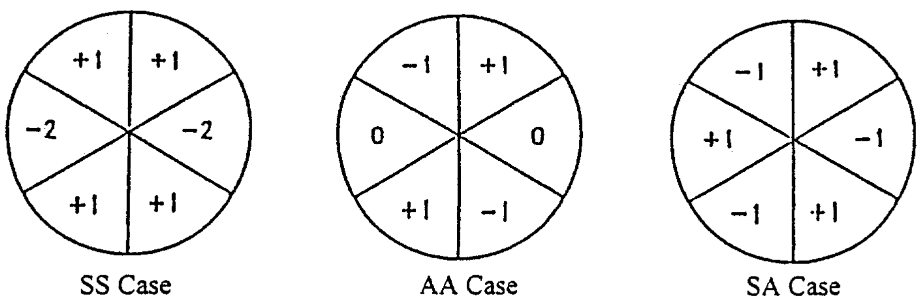


Figure 11. Patterns of force variation on mirror in the 6-zone hydraulic support system

At horizon pointing the deformation shape of the cell top surface (Figure 12) is always anti-symmetric with respect to the elevation (X) axis as the cell structure is supported on four bipods which are symmetric with respect to X and Y axes. Therefore the cell deformation will only tilt the mirror and not distort the mirror. Also at horizon pointing, the cell top surface deformation varies with the bipod location along the z axis. The optimization curve is shown in Figure 13.

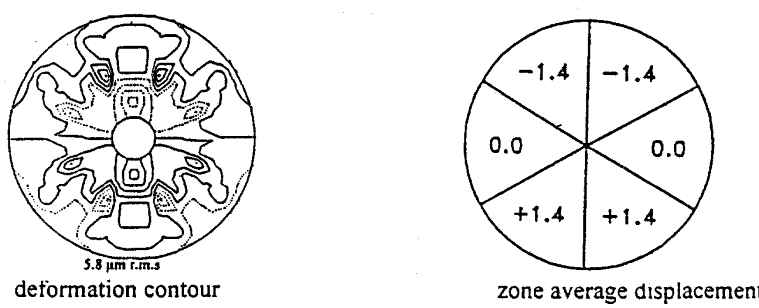


Figure 12. Cell top surface deformation at horizon pointing

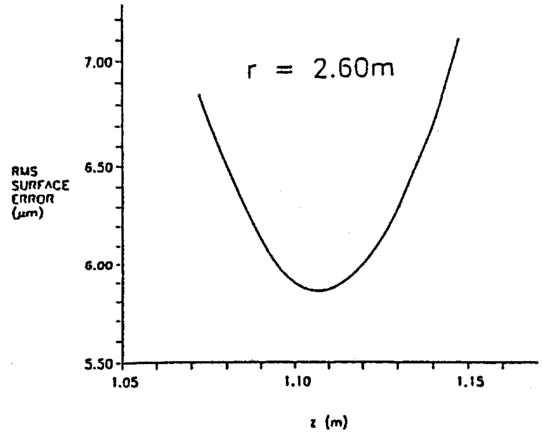


Figure 13. Cell deformation versus bipod location in Z at horizon pointing

At zenith pointing the deformation shape of the cell top surface is symmetric with respect to the X and Y axes as shown in Figure 14. The deformation is minimized by located the bipods at approximately 0.6 radius as shown in Figure 15. It is seen that all the zone average displacements are same. This cell deformation pattern will piston the mirror and will not distort the mirror. Figure 16 shows that if the bipods are not located at 42 degrees from the X axis, the zone average displacement for the two zones on the X axis is different from the other four's. This difference will induce the force variation of 'SS Case' on the mirror as shown in Figure 11 and distorts the mirror figure. Therefore an accurate determination of the bipod location for having same zone average displacements at zenith pointing is very important to the design of this cell structure.

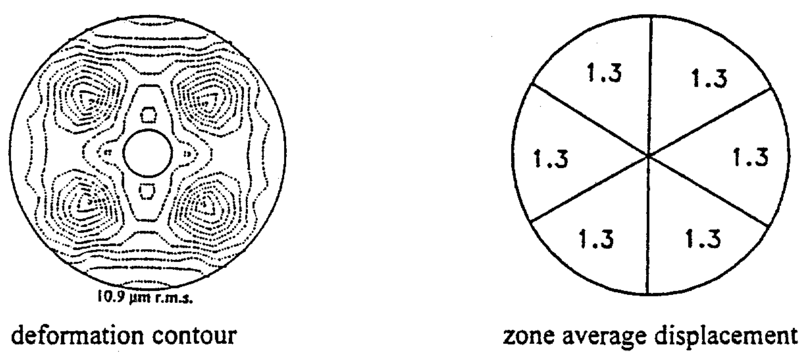


Figure 14. Cell top surface deformation at zenith pointing

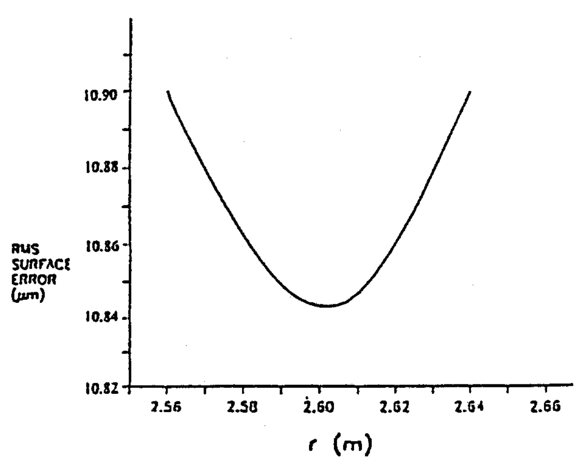


Figure 15. Cell deformation versus bipod location from center at zenith pointing

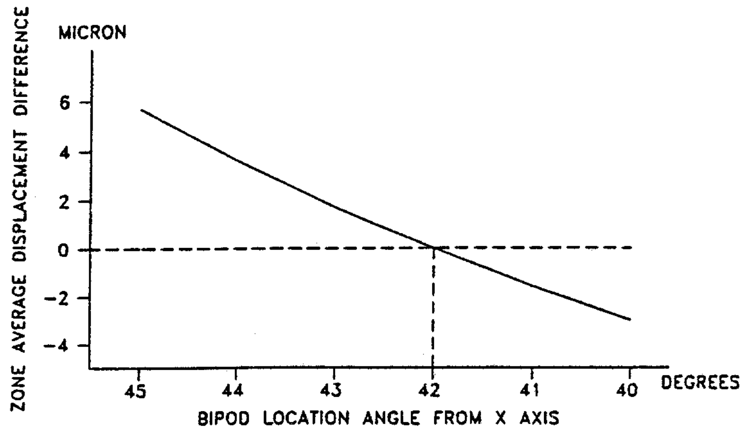


Figure 16. Zone displacement variation versus bipod location angle

The above analyses show that with proper bipod location the cell deformations at both the horizon pointing and the zenith pointing will not distort the primary mirror figure. At any other zenith angle the cell top surface deformation is a sine and cosine combination of these deformations. Therefore the primary mirror will not be distorted by the cell deformation as the telescope rotates from zenith to horizon pointing.

4.2.2 Telescope structure effect

The cell structure is supported on the telescope structure through four bipods. The advantage of using four bipods instead of the kinematic three bipods is to eliminate the gravity distortion on the primary mirror as described above. To minimize the effect due to the telescope structure deformation and the assembled mismatch, each leg of the bipod is sized as a flexure. The effects on the primary mirror figure due to various telescope distortions have been analyzed and the results are summarized in Table 3.

Table 3. Summary of Telescope Structure Effects on the Primary Mirror Figure

Case	Description	Mirror Deformation (nm r.m.s.)
A	5% torque deviation between two elevation drives	0.6
B	5 meter/second wind load on telescope structure	0.5
C	Change per minute caused by 1 watt heat input to bipod strut	0.2
D	Change per minute caused by 1G gravity load during tracking	0.1

4.2.3 Thermal effect

In the six zone support mode the cell deformation due the thermal expansion may also distort the primary mirror figure. To minimize the effect the components that dissipate heat, e.g. electronics and actuators are insulated and cooled with cooling water. They are also properly distributed inside the cell so that the thermal induced cell deformation is either uniform or axially symmetric. Analyses are performed for several expected heat loads. The results are summarized in Table 4. As expected the effect is small because the deformation due to thermal is relatively slow.

Table 4. Summary of Thermal Effects on the Primary Mirror Figure

Case	Description	Mirror Deformation (nm r.m.s per minute)
A	Uneven cooling from wind ventilation	0.2
B	Heat from 120 actuators (120 watts)	0.3
C	Heat from distributed electronics (75 watts)	0.2
D	Heat from Cass. rotator bearing (50 watts)	0.2
E	Heat from radiation plates (-375 watts)	0.1

4.3 Summary of predicted performances versus requirements

Analyses have also been done for all the other requirements in the course of the design. The predicted performances against the design requirements are summarized in Table 5.

Table 5. Summary of Predicted Performances vs Requirements

Key Requirements	Predicted Performances
o Frequency of the first cell bending mode >35 Hz	35 Hz
o Local stiffness of axial support mount > 100 KN/mm	900 KN/mm
o Primary mirror distortion due to cell deformation < 4 nm/minute	2 nm/minute
o Cass. axis gravity rotation from cell flexure < 7 arc seconds	6 arc seconds
o Local deformation of Cass. rotator bearing mount < 0.25 mm	0.03 mm
o Primary mirror distortion due to wind load on the baffle < 2 nm	1 nm
o Maximum stress in the cell structure < 13,000 psi	4,500 psi
o Weight of the cell structure = 42 − 2 tons	40 tons

5. ACKNOWLEDGMENTS

The Gemini 8-m Telescope Project is managed by the Association of Universities for Research in Astronomy, for the National Science Foundation and the Gemini Board, under an international partnership agreement.

The author would like to express his thankfulness to Michael Sheehan and Eric Hansen for their analysis data that are used to calculate the telescope structure effect and the thermal effect on the primary mirror figure. Also thanks to Dale Circle for the drawings and to Larry Stepp for many helpful discussions.

6. REFERENCES

1. L. Noethe, M. Mornhigweg, M. Ravensbergen, M. Sarazin, G. Timmermann, L. Zago, "Pressure measurements with a dummy mirror and the implications for the design of the enclosure and the support of the primary mirror of the VLT", ESO Tech. Report VLT/TRE/ESO/12300/0348, 1991.
2. E. Huang, "Response of the primary mirror to wind loads", Gemini 8m Telescope Project Tech. Report RPT-O-G0035, 1992.
3. L. Stepp, E. Huang, M. Cho, "Gemini primary mirror support system", SPIE Vol. 2199, 1994.
4. I-DEAS finite element analysis software, Version VI, 1991, developed by Structural Dynamic Research Corporation, Milford, Ohio, USA.