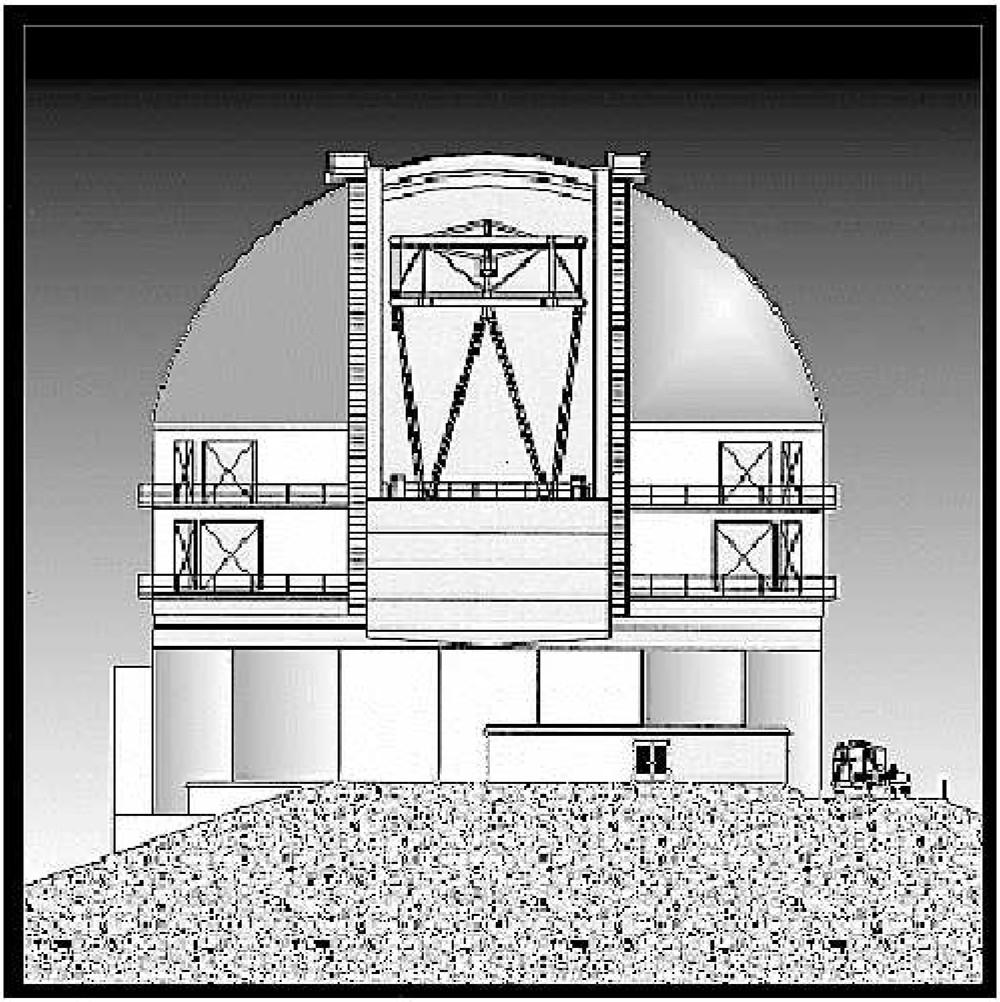
GEMINI
8-M Telescopes
Project

RPT-O-G0025

Conceptual Design of the Primary Mirror Cell Assembly



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1. EXECUTIVE SUMMARY

This report describes the design concepts for the primary mirror assembly. In some cases these concepts are innovative, but most design features have been adopted from previous large telescope designs, and are innovative only in the way they have been combined. None of the technology required is experimental.

Subsequent reports will discuss the design features in detail, and will describe the extensive analysis that has been done to optimize the designs and predict the performance of the system. This report provides an overview, explaining why design choices were made, and describing how the components of the assembly work together as a system. Comparisons are made to more traditional designs, with explanations of departures from traditional designs that are required to meet the stringent Gemini performance specifications.

Figure 1 is a center cross section view of the mirror cell assembly showing the main components. The cell structure is a steel weldment designed to have high stiffness and resonant frequencies. The mirror supports are attached to the cell at locally stiff points in the structure. The ribs of the structure are far enough apart (about 70 cm) to allow personnel access for maintenance. Mounted to the inside diameter of the cell are instrument safety covers and the attachment mechanism for the primary baffles. Deployable optical correctors will sometimes be installed inside the central hole of the cell, at a level just above the Cassegrain rotator bearing.

The mirror support system consists of an air pressure support plus axial and lateral distributed hydraulic systems. An active pneumatic actuator is also associated with each axial and lateral support. Axial support mechanisms can be replaced from below the mirror, and lateral support mechanisms can be replaced by personnel at the outer edge of the cell, without removing the mirror from the cell. The axial support mechanisms are not bonded to the mirror, to facilitate mirror removal for recoating. An ambient temperature cooling water loop will remove heat generated by the active optics actuators.

The thermal control system consists of a radiation plate located behind the mirror, and a coating heating system on the mirror front surface. Insulation behind the radiation plate and around the cooling water pipes will minimize the effect of the thermal control system on the temperature of the cell structure.

The entire cell assembly along with the mirror cell cradle can be lowered onto a handling cart to remove the cell from the telescope.

2. INTRODUCTION

The Gemini Project has ambitious goals. The two telescopes will be among the largest optical telescopes in the world. They are specified to deliver image quality better than any existing ground-based telescope of any size. The two telescopes are to be highly versatile, carrying many different instruments that can be engaged within minutes during the night. The wavelength range for observations is to be from 0.3 to 30 microns. In IR mode, the emissivity of the two telescopes is to be lower than existing IR-only telescopes. And, the two telescopes are to be built for a fixed cost--which is to be a lower cost per telescope than similar 8-meter projects, for example, Subaru and the VLT.

The challenge for the project team is to produce telescopes that are outstanding performers, to build them on schedule, and at a controlled cost. The best chance for accomplishing this is by developing designs that are innovative, but not experimental. We cannot afford to pioneer new technologies, but we must reexamine all aspects of the telescope design and not simply follow traditional approaches. We must also take advantage of the innovative work that has been done on other telescope projects.

This report describes the conceptual design of the Gemini primary mirror assemblies. Section 3 describes the key problems to be solved. Sections 4 and 5 highlight the two most difficult of these, mirror seeing and wind buffeting, and describe the system design features incorporated to solve them. Section 6 describes the optical requirements. Sections 7 through 10 describe specific aspects of the primary mirror assembly and its subsystems. Section 11 describes interfaces with other parts of the telescopes. Section 12 summarizes the design features discussed in the earlier sections, and provides a list of open questions to be answered by later Gemini technical reports.

The functional requirements for the primary mirror assembly are detailed in the *Functional Requirements for the Primary Mirror Assembly* .

3. THE PARETO PRINCIPLE APPLIED TO LARGE TELESCOPES

J. M. Juran developed the Pareto Principle to describe the effect of manufacturing problems on product quality. It states there are a "vital few" factors that contribute the majority of the defects, and a "trivial many" factors that altogether result in only a small fraction of the errors. This principle points out the need to concentrate first on solving the most significant problems and to proceed with solutions to lesser problems only after the major ones have been eliminated.

It is interesting to apply this principle to the optical performance of large telescopes. Seventy-five years ago, large telescopes weren't close to being diffraction limited. During the past 75 years many of the "vital few" problems limiting their performance have been solved. Major advances in optical testing techniques now make it possible to produce diffraction limited optical surfaces on large mirrors. The warping of mirrors exposed to uneven temperatures has been nearly eliminated by use of "zero expansion" materials having a coefficient of thermal expansion (CTE) less than 50 parts per billion. Alignment problems caused by distortion of telescope structures under gravity can now be predicted, and therefore controlled. using finite-element analysis methods. Similarly, the design of mirror support systems has been significantly improved by finite-element methods, and now active optics techniques make it possible to re-optimize the mirror support in the telescope. Tracking performance has been much improved by modern servo systems, improved drive designs, and the use of relatively smaller telescope structures resulting from fast primary mirror focal ratios. Even the performance of the atmosphere has been improved in the sense that a few superb sites have been discovered where free atmosphere seeing conditions are available much of the time.

As we have worked through the list of factors limiting image quality, solving the "vital few" problems one by one, we have moved from 2 arc second images, to 1 arc second images, to 1/2 arc second images. We are now into the territory of the "trivial many" problems that must be solved in order to accomplish any further improvements. It is time to start over, to reexamine the list of performance limiting factors in order to identify the most significant problems remaining. This will produce a new list of the vital few problems, as of today. The solution of these problems should receive the greatest emphasis in the design of the primary mirror assembly. Our analysis shows that, at present, the vital few problems are:

- Atmospheric seeing
- Diffraction
- Local seeing
- Wind buffeting

Of the remaining limitations on image quality, the largest single factor is the atmosphere itself. Adaptive optics systems now show promise of canceling out wavefront errors introduced by the atmosphere. Although the primary mirror assembly is not really part of the adaptive optics system, it must be designed so as not to limit future adaptive optics performance. Therefore, the Gemini Science Requirements Document states,

> "The primary and secondary mirrors must have optical quality which allows the tilt-correcting and adaptive optics systems to reach Strehl ratios of 0.5 at 1.6 microns and 0.2 at 0.79 microns. This particularly requires that the mirrors must be smooth on the spatial scales which cannot be corrected by the primary active optics. "

This imposes limits on high spatial frequency errors introduced by polishing and by the mirror support. Control of high frequency support errors will be discussed in Section 8.

Diffraction is the largest term in the Gemini error budget, at a wavelength of 2.2 microns. Of course, diffraction can only be reduced by building a telescope with larger aperture.

The vital problems that most directly affect the design of the primary mirror assembly are local seeing, particularly mirror seeing, and wind buffeting of the primary mirror. These two problems, and the design features proposed to solve them, are described in Sections 4 and 5.

In addition, there are a number of other error factors to be considered (the so-called trivial many) that are only slightly less significant than mirror seeing and wind buffeting. These include: optical design, optical alignment, polishing residuals, mirror support, and mirror thermal distortion. Allowable limits for these error factors are defined in the Gemini System Error Budget Plan and features of the primary mirror assembly design that help control these effects are described in Sections 6 through 11.

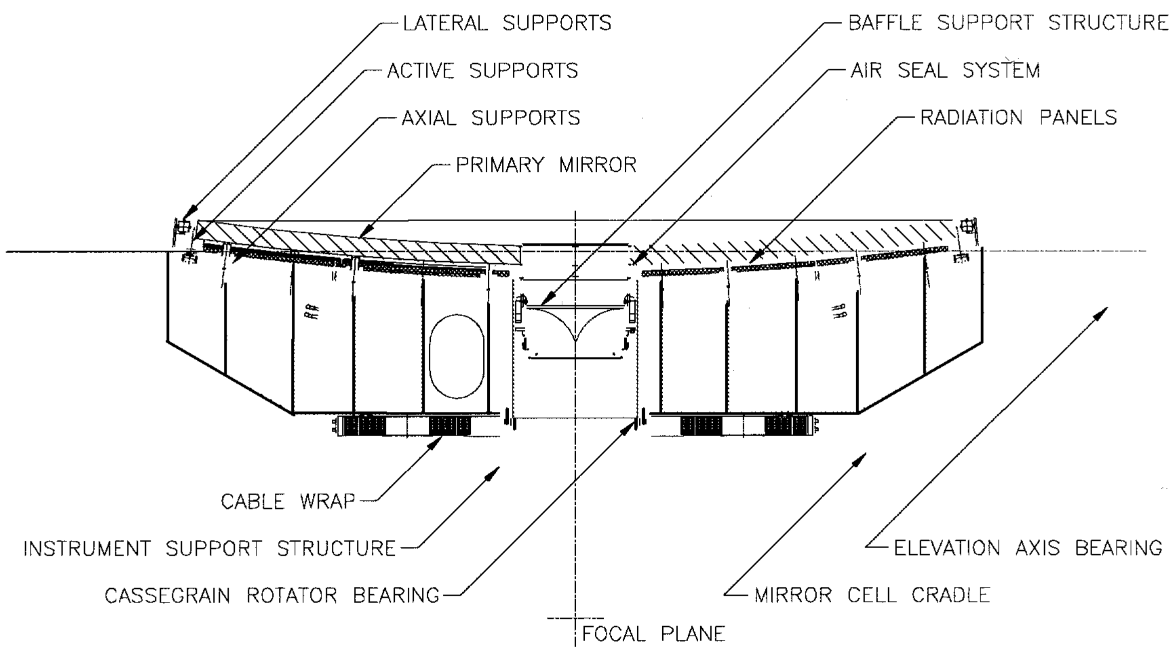

Figure 1. A cross section of the Gemini primary mirror cell assembly, showing the principal subsystems.

Please note that categorization of design problems as part of the vital few or trivial many is specific to the type of telescope proposed. For large telescopes having other designs, for example, telescopes with a different primary f/ratio, or with a different type of primary mirror, the list of vital few problems would be different, and would include some of the problems that are considered part of the trivial many for the Gemini design.

4. MIRROR SEEING

At one time many astronomers believed that atmospheric seeing limited the resolution of large telescopes to about one arc second, but over the last two or three decades it has become apparent that local seeing, produced by pavement, domes, and even the telescope itself, can be as significant as the effects of the rest of the atmosphere. One type of local seeing that is receiving particular attention is mirror seeing caused by the primary mirror. When the mirror is at a different temperature than the air? heat transfer between the mirror and air results in refractive index differences in the air that produce wavefront aberrations. In some cases these refractive index patterns are relatively stationary, and can be corrected by active optics, but in general they change and move over time periods of a few seconds.

Most telescopes with thick glass mirrors suffer from noticeable amounts of mirror seeing. As the air cools at sunset, the thick mirror cannot cool as rapidly. After sunset the air cools more slowly, but if the mirror is already significantly warmer than the air, it will remain so throughout the night unless its temperature is actively controlled in some way. The temperature of the mirror surface is cooled by contact with the air (it is this heat transfer, after all, that produces the mirror seeing) but the core of the mirror substrate continues to conduct heat to the surface, maintaining its temperature at an intermediate point between the temperature of the air and that of the mirror core.

Several studies have shown that the effects of mirror seeing are not as pronounced for a mirror cooler than the air as for a mirror that is warmer[1,2,3,4]. Based on these and other[5,6,7,8] studies, the Gemini Project Scientist has concluded that, for conditions with a moderate amount of ventilation by the wind, mirror seeing will be within the levels set by the Gemini error budget if the mirror temperature is no more than 0.2 degree C warmer than the air? and no more than 0.6 degree cooler. More details about this requirement are given in the document *Functional Requirements for the Primary Mirror Assembly* .

Two aspects of the problem are important to keep in mind: first, the temperature that is important is the temperature of the mirror surface, rather than the temperature of the mirror core; second, any mirror thermal control strategy should aim for the mirror to be slightly cooler than the air, rather than slightly warmer.

The air temperature at the Gemini sites declines slowly after sunset. For example, the average rate of temperature change during the night on Mauna Kea is -0.12 degree C per hour. In order for the mirror front surface temperature to decline 0.12 degree C per hour, <u>if there were no heat transfer with the air</u>, the back surface of the mirror would need to be 1.7 degrees cooler than the front. To accomplish this, 11 watts per square meter must be continuously removed from the back of the mirror. This is? in fact, not difficult to accomplish. It is fairly easy to follow the average night-time air temperature decline by cooling only from the back of the mirror. even if heat transfer at the front surface is neglected.

Of course, the night-time air temperature does not follow a perfectly linear ramp. There are variations both plus and minus from the average slope. However, when the air temperature varies from the temperature being followed by the mirror surface, there is air-to-glass heat transfer that keeps the glass surface temperature intermediate between the air temperature and the internal glass temperature. Experience at the Hiltner Telescope on Kitt Peak[9] indicates that during observing, the mirror surface temperature is typically midway between the air temperature and the glass core temperature, owing to the good ventilation characteristics of the telescope enclosure.

The baseline Gemini thermal control strategy is therefore to start the night with the mirror surface in equilibrium with the air, lower the temperature of the back surface of the mirror to create a heat flow that will pull the front surface temperature downward at a rate equal to the average temperature decline of the air, and then let the mirror front surface partially follow the small minute-by-minute variations of the air temperature by virtue of natural wind ventilation. In addition, we are also working on an enhancement to this approach that will increase our ability to control the front surface temperature on much shorter time scales. The proposed Gemini thermal control design is described in Section 9.

5. WIND BUFFETING

Modern large telescope designs have moved towards use of lightweight mirror substrates to reduce thermal inertia, total moving mass, gravity distortion of the telescope structure, and cost. As primary mirrors get larger and relatively lighter, they become more susceptible to wind buffeting. Wind buffeting effects include both rigid body motion and elastic deformation of the mirror, at frequencies ranging from quasi-static up into the acoustic range.

Rigid body motion can produce focus and alignment errors and image motion. All of these enlarge the integrated image spot size.

Elastic deformation is caused by the combined effects of wind pressure and the force reactions that are produced in response to the wind pressure. It is important to remember that under wind loading, the shape of the mirror deformation is determined by the type of defining system used to resist mirror movement.

Vibration of the mirror can also be a problem, either from rigid body oscillation on the support system or from elastic modes of mirror bending. If system resonant frequencies were low enough to be excited by the wind, and if damping in the mirror support system were low, a large amplitude of motion could build up at resonant frequencies.

Many existing telescope enclosures were designed to minimize the exposure of the telescope to the wind. However, it is now realized that natural ventilation can significantly reduce the effects of local seeing, therefore it is an advantage if the telescope can function properly with wind ventilation of several meters per second inside the enclosure .

To minimize rigid body motion of the mirror and keep resonant frequencies high, the mirror cell and defining system must be very stiff. For example? to meet the Gemini error budget for wind-induced focus error, the mirror cell and defining system must have a stiffness against mirror piston movement of the order of six hundred Newtons per micron . As it turns out, approximately this level of stiffness is also required to maintain proper spacing between the primary and secondary mirrors under the effects of changing gravity orientation.

To meet this requirement the Gemini design uses a very stiff welded steel plate structure for the mirror cell. This structure is shown in Figure 2, and is described further in Section 10. Note that the supports are located directly over the stiffening ribs in order to maximize local stiffness. The cell structure is 1.5 meters thick at the center and it weighs over 40 tons. Such a massive structure can be used because it is close to the telescope elevation axis, therefore it does not have a large effect on the telescope balance. Figure 3 illustrates the Gemini telescope structure, showing the location of the primary mirror cell.

The defining system is also designed to be stiff; to be compatible with overall system stiffness requirements, the defining system stiffness must be approximately 1000 N per micron. In many traditional mirror support designs, the mirror's position and orientation are defined by three "hard points" behind the mirror, but most of these designs have not attempted to achieve a stiffness of 1000 N per micron. For example, if we placed the mirror directly on three 2-inch diameter steel balls, with the balls supported on hardened steel pads, the total stiffness to piston movement would be about 100 N per micron approximately an order of magnitude lower than required.

However, there are alternative defining systems that spread the load over more than three points. For example, the 3.5-meter telescope at Calar Alto, designed by Zeiss, has a different type of defining system based on what is sometimes called a "hydraulic whiffletree" because it is in some ways the hydraulic analog of a mechanical whiffletree support. We prefer to call this a "distributed defining" system, because the defining function is distributed over all the support points. When the wind exerts pressure on the surface of the mirror, it produces reactions at all of the support points instead of at just 3 hard points. This provides a large effective spring constant, while minimizing contact forces exerted on the mirror. For the Gemini design, having 120 hydraulic cylinders in the defining system, the required stiffness per mechanism is only about 8 N per micron, an achievable value for the type of hydraulic cylinder that will be used. For more information on hydraulic mirror support systems, see the Gemini technical report *on Primary Mirror Forces From a Distributed Hydraulic "Axial " Support System* .

The Gemini baseline design uses a hydraulic distributed defining system for the primary mirror. Specific design innovations have been developed to enhance the resistance of this system to uneven wind loads by coupling the system to the stiffness of the mirror cell. The design of the Gemini support and defining system is described in Section 8.

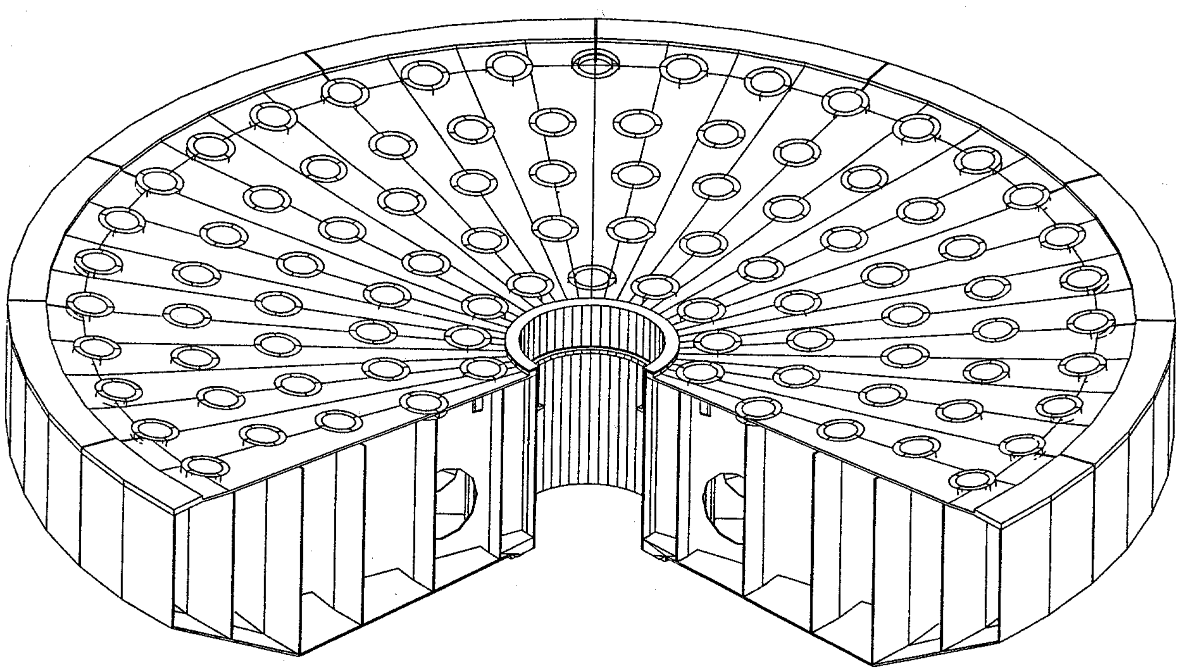

Figure 2. A conceptual sketch showing a cut-away view of the Gemini primary mirror cell

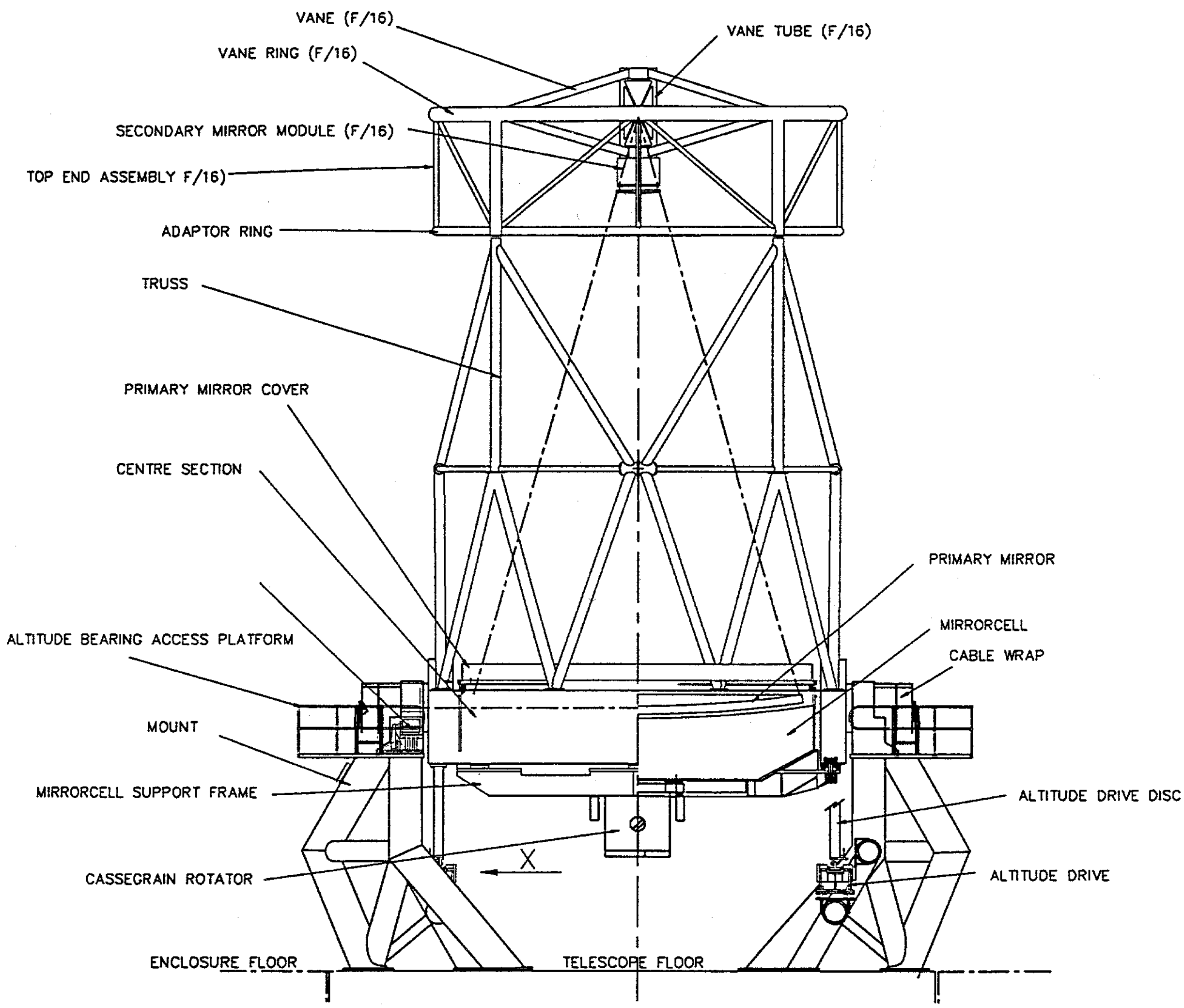

Figure 3. The Gemini Telescope structure, showing the location of the primary mirror cell.

6. OPTICAL DESIGN AND SPECIFICATIONS

6.1. Optical Design

In designing a two-mirror optical system, one of the fundamental choices to be made is whether to use a classical Cassegrain design, or an aplanatic Ritchey-Chrétien design. The classical Cassegrain has the advantage that, since the primary is a paraboloid, any number of different secondary mirror f/ratios and back focal distances can be used. Gemini plans to use both f/16 and f/6 secondary mirrors, and additional options might be developed in the future. The Ritchey-Chrétien has the advantage that both spherical aberration and coma are corrected, so the images over a wide field of view are sharper, and more symmetrical, than those from the classical Cassegrain.

However, it must be optimized for a single focal ratio. If a second focal ratio is implemented, and if that focal ratio is longer than the optimized one, the images will be worse than for a classical Cassegrain. However, we are in the fortunate position of being able to have both a classical Cassegrain and a Ritchey-Chrétien, by virtue of the meniscus mirror and its active optics system. The focal ratio for the baseline telescope configuration in both hemispheres is f/16? and the optical design of the telescope is optimized as a Ritchey-Chrétien at f/16. The conic constant of the primary mirror is -1.00376. The departure of this hyperboloid surface from that of the best-fit paraboloid is about 1.2 microns peak-to-valley. The active optics system can change the mirror figure to a paraboloid with no more than 75 newtons of force at any actuator? and with a residual distortion of about 2 nm RMS. It would also be possible to change the conic constant up to a value of about -1.01 in order to optimize a Ritchey-Chrétien system at a faster f/ratio. This active optics capability will allow excellent versatility in future optical design options? while optimizing the bare-focus performance of the baseline design? an essential feature for an IR optimized telescope required to deliver superb image quality.

6.2. Optical Specifications

The Gemini Science Requirements specify image quality in terms of energy concentration for specific telescope configurations and wavelengths. In translating these requirements into fabrication specifications it is necessary to consider the spatial frequency of surface errors.

Errors on the surface of a nearly diffraction limited mirror will primarily scatter light into the first diffraction order corresponding to the spatial frequency of the error. For a mirror of diameter D with a figure error of characteristic spatial period d and spatial frequency f, where f = D/d, the angle that light of wavelength λ is diffracted away from the center of the point spread function is:

$$\theta = \frac{\lambda}{d} = \mathrm{f}\frac{\lambda}{D}$$

This can be compared to the angular size of the diffraction-limited point spread function (PSF). For a telescope with a central obscuration ratio of 0.15? the radius of the first dark ring is:

$$\alpha = 1.19\frac{\lambda}{D}$$

Surface figure errors can be classified by their effect on the point spread function. Low spatial frequency errors, where f is less than about 5 cycles per aperture, scatter light close to the central core of the PSF; they essentially broaden the base of the PSF. The Seidel aberrations are examples of low frequency errors. Wavefront errors caused by misalignment fall into this category. In general, errors in this range can be controlled by the active optics system.

Mid spatial frequency errors, where f is between about 5 and the highest frequency of error that can be produced by the polishing tool (perhaps 200), scatter light widely. The wings produced by these errors can extend for several arc seconds. These errors generally cannot be controlled by active optics, yet they can be large enough to scatter a significant fraction of the light.

High spatial frequency errors, where f > 200, tend to be controlled by the natural smoothing action of the polishing tool. These errors can be characterized as surface roughness. Light will be scattered from these errors very widely (up to several degrees), but for properly polished surfaces the amplitude of the scattered light is normally very small for visible and IR wavelengths.

For a nearly diffraction limited optical surface with an RMS surface error 6, the amount of energy scattered is proportional to σ^2/λ^2. A given amplitude of surface error will have significantly different effects at different wavelengths. As λ increases, the scatter angle increases, and the amount of scattered energy rapidly decreases. For example, the amount of light scattered out of the central core of the image by a 10 nm RMS surface error will be 0.3% at a wavelength of 2.2 microns, but will be about 18% at a wavelength of 0.3 micron. The distance this energy scatters away from the central core is determined by the spatial frequency content of the surface errors. Therefore, when evaluating the performance of the Gemini telescopes, we must calculate the effects of the predicted errors on the image energy concentration, at specified wavelengths, rather than simply determining the RMS error of the surface.

The polishing specifications for the primary mirrors have been set to not only meet the tight image quality requirements at 2.2 microns, but also to ensure the optical surface is smooth enough to be compatible with future adaptive optics work at shorter wavelengths. Active optics correction of low frequency surface errors by the polisher will be allowed. This will let the polisher concentrate on producing a mirror that is smooth at high spatial frequencies.

The conic constant of the mirror can also be adjusted using the active optics system. If no active optics compensation were possible, the conic constant would need to be controlled within 55 parts per million to meet the error budget. Because of the active optics capability, variations or uncertainty of greater than 5 parts per thousand can be allowed.

The polishing specifications are summarized in the *Primary Mirror Polishing Specification* .

7. Primary Mirror Substrate

The Gemini primary mirrors will be monolithic menisci made from Corning ULE™ titanium-doped fused silica. The mirror blanks will be formed by fusing boules of ULE™ together into stacks, cutting the stacks into hex-shaped pieces? fusing the hexes into a monolithic flat disk, and then slumping the disk into the meniscus shape. Figure 4 illustrates the arrangement of hexes into the full disk.

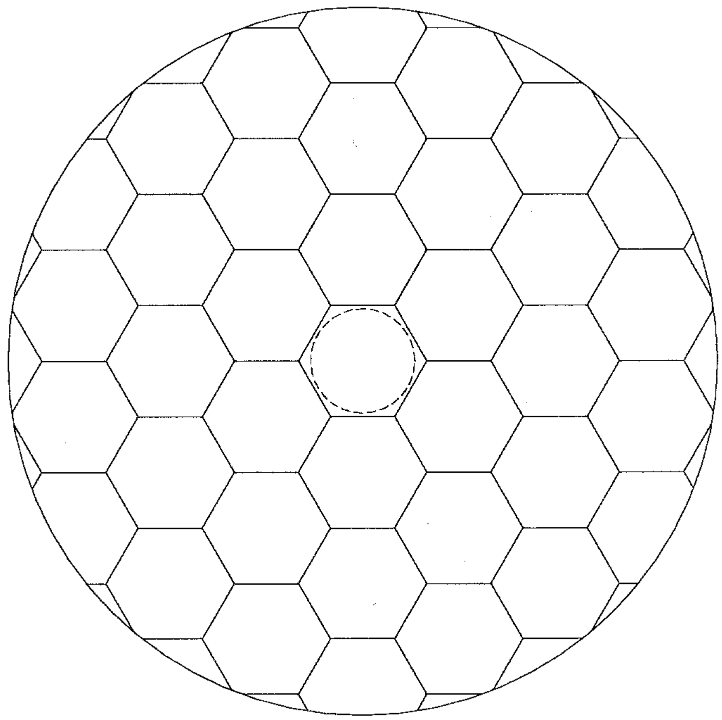

Figure 4. Arrangement of ULE™ hexes into a mirror blank.

The Gemini meniscus primary mirrors will have a number of performance advantages. Most of these advantages are related to: (1) the ultra low CTE of the glass; (2) the solid, monolithic structure; and (3) the uniform cross section.

As is well known in the optics industry, low CTE is an important factor for large mirrors. In the optics shop, ULE™ mirrors can be tested shortly after polishing, without allowing time for thermal stabilization, and without requiring a thermal control system built into the polishing setup. Uncertainty in optical testing results is minimized. No special thermal precautions will be required during shipping and handling. The ultra low CTE is also important in the observatory environment. Detailed finite-element studies indicate the Gemini ULE™ meniscus mirrors will meet the tight error budget requirements for thermal distortion with only minimal help from the active optics system. These studies are described in the Gemini technical report *Distortion of the Primary Mirror Due to Temperature and CTE Nonuniformity* .

The solid structure of the meniscus mirror offers several advantages. Because of the high local stiffness, and the opportunity to grind all glass surfaces, the meniscus mirror can be safely handled and transported. The mounting pads for support mechanisms can be relatively small on a meniscus mirror, and the number of support points can be relatively low, which is particularly an advantage for lateral supports. Because of the solid structure and the mirror support design described in Section 7, the Gemini mirrors will not suffer the same print-through problems that would occur in a lightweight structured mirror. This avoids possible high spatial frequency surface errors that could not be corrected by active or adaptive optics systems.

The uniform cross section also offers advantages. Because of the uniform area weight density, and the uniform curvature, excellent results can be achieved by supporting most of the mirror weight on a uniform air pressure. In addition, the locations of the mechanical supports can be optimized, so for example the mechanisms can be arranged in concentric rings to match the mirror geometry. This allows the applied forces, and therefore the mechanisms, to be designed to be identical, which has important advantages for low cost and simplified maintenance. The uniform cross section also simplifies the thermal control system design.

In summary, the properties of the ULE™ meniscus mirrors provide the advantages of: (1) resistance to mechanical stresses and thermal environments; (2) ease of polishing and testing; (3) low distortion from temperature variation; (4) simple support system and thermal control system design and fabrication; and (5) little or no print-through problems if supported on the right type of support system.

Fabrication of the Gemini primary mirror blanks is currently underway at Corning. They will meet stringent specifications for glass quality, uniformity of CTE, number of bubbles and inclusions, and dimensional accuracy. These requirements are summarized in the *Primary Mirror Blank Specification*, which is abstracted from the Work Scope of Gemini Contract No. 47990-GEM00050.

8. MIRROR SUPPORT SYSTEM

8.1. Definitions and fundamental concepts

A mirror support system must perform several functions. First, it must support the mirror's weight, at all telescope orientations, keeping the mirror figure smooth enough and accurate enough to meet the error budget. Second, the system must control the mirror's position and orientation in 6 degrees of freedom, and for the Gemini Telescopes it must be able to make accurate adjustments to the mirror position in X, Y and Z translations as well as two axes of tilt in real time under computer control. Third, the system must be able to vary the forces exerted on the mirror, allowing the mirror figure to be reoptimized while in service.

We refer to the first function, that of carrying the mirror weight, as " mirror support".

We refer to the second function, that of controlling the mirror position and orientation, as "defining"..

We refer to the control of mirror figure during use as "active optics".

The supports that carry the mirror weight at zenith-pointing are called axial supports, because the resultant of the applied forces is parallel to the optical axis. The supports that carry the mirror weight at horizon-pointing are called lateral supports, because the resultant of the applied forces is perpendicular to the optical axis. In reality, none of the individual mechanisms applies a force precisely parallel to or perpendicular to the optical axis. Some telescope designs combine both of these functions into a single system, however, they are separate in the Gemini design.

To understand the Gemini mirror support design it is necessary to understand the differences between:

- bending deflections vs. shear deflections
- global stiffness vs. local stiffness
- passive performance vs. active optics performance
- force control vs. displacement control
- symmetric deflections vs. anti-symmetric deflections
- figure errors vs. print-through
- "three-point" defining vs. distributed defining
- kinematic defining systems vs. overconstrained defining systems

Bending deflections vs. shear deflections. A mirror can be treated as a thin plate subjected to gravity and support reaction loads. Its elastic deflection in response to those loads can be categorized in terms of bending deflection and shear deflection. The normal assumptions for plate bending are:

- The deflection of the midsurface is small compared to the thickness of the plate
- Any straight line initially normal to the midsurface will remain straight and normal to the midsurface after bending
- There is no stretching or contracting of the midsurface

For plate bending, the deflection is proportional to:

$$\delta \propto \frac{Wa^4}{Et^3}$$

where W is the weight per unit area, a is the length between support points, E is the elastic modulus of the material and t is the thickness of the plate. Note that the resistance of a plate to self-weight bending varies as the square of its thickness.

In general the second assumption stated above is not completely true; straight lines normal to the midsurface do not remain completely straight, because there is shear deflection in addition to bending deflection. Shear deflections are proportional to the weight per unit area, and inversely proportional to the thickness of the plate. This means that shear deflections are not reduced by increasing mirror thickness, since the weight loading increases in the same proportion.

For thin plates the shear deflection is normally small compared to bending deflection, but as the relative thickness of the plate increases the bending deflection decreases so that in relative terms shear deflection becomes more significant. For large mirrors on a multiple-point support, shear deflections are clearly significant; for a meniscus mirror 20 cm thick supported at points 70 cm apart approximately half the deflection comes from shear rather than bending.

This explains why large structured mirrors need approximately the same number of supports as a thinner meniscus mirror. The bending stiffness of a sandwich structure mirror depends on the ratio of its effective flexural rigidity to its weight, which increases rapidly with increased overall thickness. However, the resistance to shear deflection depends on the ratio of effective cross sectional area to weight, which is hardly affected by the overall thickness. As the bending deflections are reduced the shear deflections dominate.

Global stiffness vs. local stiffness. If a large mirror is supported at widely separated points, for example if it is only supported at its outer rim, and if the mirror is loaded by a force at its center, the deflections of the mirror could be well approximated by only considering the bending properties of the mirror as a whole. In this case, the bending stiffness of the mirror would determine the amount of deflection. This is an example of the concept of global stiffness.

On the other hand, when a mirror is supported by more than 100 points relatively close together, the deflections are strongly influenced by the local stiffness of the mirror immediately above the loading points. For example, when a lightweight eggcrate structure mirror is loaded by a force on the back plate midway between reinforcing ribs, the local deflections can be relatively large, but when the same load is applied to a point directly below a rib, the deflections will be smaller. In general, meniscus mirrors have relatively low global stiffness and relatively high local stiffness, and structured mirrors tend to be just the reverse. This is particularly an issue for the design of lateral supports.

Passive performance vs. active optics performance. One of the fundamental choices to be made in designing the support system is the level of performance that will be accomplished passively, as opposed to actively. For example, the axial supports of the 5-meter Hale Telescope use counterweights, which apply the correct force passively, without any control system. The 4-meter Mayall Telescope axial support is an active system, because the support forces are controlled by a regulated air pressure, but the control uses a passive mechanism to set the air pressure. By comparison, the 3.5-meter telescope at the Starfire Optical Range in New Mexico uses an entirely active system. The support forces are provided by pneumatic actuators supplied by several dozen air pressure regulators under computer control.

The support system in the Hale Telescope is not dependent on outside power or control signals. The support system in the Mayall Telescope depends on a continuous supply of compressed air. The support system in the Starfire telescope depends on an air supply and on receiving signals from the computer.

The 3.5-meter WIYN Telescope on Kitt Peak has a lightweight primary mirror that is very similar to the mirror in the Starfire Optical Range telescope, but it has a fully functioning passive support system in addition to active optics capability. With all the active forces set to zero, the support system will operate properly over the entire range of zenith angles, although the active optics system is still needed to compensate for thermal distortion. The passive capability of this system offers advantages but a compromise is required in the form of increased mechanical complexity. The support system for the Starfire telescope requires only a few different mechanism configurations, because the force settings are all done actively. The WIYN support mechanisms have 66 different configurations in order to establish the correct forces passively. This is necessary because, in order to place the supports at locations with good local stiffness, the supports must be located behind the junctions of ribs. Since the support locations are constrained, the support forces must be individually optimized to achieve best performance.

Another aspect of the decision between passive/active and fully active systems is the difference in update rate required. L. Noethe reports[10] that for the ESO VLT primary mirrors, a purely active support system would have to be updated once per second as the telescope tracks across the sky in order to control applied forces to the correct level. This is one reason ESO has chosen a passive/active support design for the VLT.

Gemini has also chosen a combined passive/active support design. On a meniscus mirror the support locations can be optimized instead of the forces, allowing the mechanisms to be identical (the Gemini design actually has two sizes of actuators in order to get better results around the central hole). This avoids the mechanical complexity of a passive design where all the forces must have individual magnitudes.

Force control vs. displacement control. The optical surface figure of the mirror can be controlled in two ways, either by force control or by displacement control. Traditionally, support systems for large mirrors have been based on force control. For this approach, the support mechanisms are designed to be astatic. They have very low spring constants, to ensure the applied force will not be influenced by small amounts of motion between the mirror and cell. The mirror shape is controlled by applying the correct forces, that is, controlling the mirror shape through the stiffness of the mirror substrate.

An alternative approach is to control the position of the actuators. This requires the actuators to be relatively stiffer than the mirror substrate; for this approach the spring constant of the actuators should be high. Adaptive optics mirrors are often controlled by displacement support systems.

Force control systems are normally designed to be kinematic, that is, they only constrain 6 degrees of freedom of the mirror. Displacement control systems are often overconstrained. They normally have as many constraints as the number of actuators, and if this number is more than six they are not kinematic. The Gemini support system has aspects of both. It can be used in a mode having nine constraints instead of six. In this mode it can be thought of as having both force control and displacement control of the mirror. These concepts are discussed further in the Gemini technical report, *The Distributed Defining System for the Primary Mirrors*.

Symmetric deflections vs. anti-symmetric deflections. Mirror surface deformations can be categorized as symmetric or anti-symmetric about the coordinate axes. We use the notation that an aberration that is symmetric about both the X and Y axes is designated "SS"; an aberration symmetric about X but anti-symmetric about Y is designated "SA", an aberration anti-symmetric about X but symmetric about Y is designated "AS"; and an aberration that is anti-symmetric about both the X and Y axes is designated "AA".

Each Zernike polynomial term can be categorized as SS, SA, AS or AA. Moreover, any possible combination of mirror surface figure errors can be modeled by superposition of four component parts, having SS, SA, AS and AA symmetries.

These concepts are important to an understanding of the interaction between the mirror cell deflections and the possible aberrations introduced in the mirror. They are discussed further in the reports, *The Distributed Defining System for the Primary Mirrors*, and *Primary Mirror Cell Deformation and its Effect on Mirror Figure Assuming a Six-zone Axial Defining System*.

Figure errors vs. print-through. Errors produced in the optical surface of the mirror by the support system can be divided into two categories: figure errors and print-through. Figure errors tend to be global in nature while print-through is local. As a working definition, figure errors can be defined as being caused by incorrect forces applied to the mirror. In this sense, if all unintended external loads have been eliminated, and if the support forces have been optimized to provide the minimum surface error, the remaining surface distortions can be considered print-through.

There are several possible ways to reduce print-through.

- The force per actuator can be reduced, either by increasing the number of support points so that the mirror weight is divided into more parts, or by off-loading some of the mirror weight onto a separate flotation system that does not cause print-through.
- The separation distance between support points can be reduced, by increasing the number of
- supports
- The contact pad at the mirror can spread the load more widely
- The contact with the mirror can be designed to produce bending moments that partially compensate for the force-induced print-through (as in the ESO VLT support design).
- The print-through can be polished out in the optics shop, using the telescope support system during optical testing; however, if the loading on the actuators changes with zenith angle, the print-through will exist to some extent at all other zenith angles.

"Three-point" defining vs. distributed defining. The function of a defining system is to define the position and orientation of the mirror. Depending on the support design, the defining system can carry all of the mirror weight, part of the weight, or none of it. However, the defining system does carry 100% of external loads. Examples of external loads include wind loading, loads applied by any mirror cell hardware inadvertently contacting the mirror, or inertial effects caused by accelerating the telescope during stewing. The defining system must constrain the mirror against movements in three orthogonal directions of translation, as well as against rotations about three orthogonal axes. In all, six degrees of freedom must be constrained.

Defining systems can be classified as simple or distributed. In a simple defining system each of the six degrees of freedom is constrained by one contact point between the mirror and defining system. A common example is a small mirror resting on three steel balls, with one ball on a flat surface, one in a V-groove, and one in a conical depression. There are six contact points, and six degrees of freedom constrained.

As mentioned in Section 5, defining systems can also be distributed. In a distributed defining system external loads are reacted on all the supports rather than on just six contact points. Distributed defining systems can be fully kinematic, constraining only six degrees of freedom. An example of a kinematic distributed defining system is a 9-point Hindle mount.

Distributed defining systems are better at resisting wind loading. If a uniform wind load is reacted at only three points, the mirror will tend to bend in a three-lobed shape. On the other hand, if the reaction is evenly spread over the back surface of the mirror, at a large number of points, the deformation of the mirror will be much smaller. The reaction of a distributed defining system is better matched to the wind loading input because a uniform wind load over the mirror surface is very similar to a zenith-pointing gravity load, but is approximately three orders of magnitude smaller. Distributed over all of the axial support locations, uniform wind loads produce negligible amounts of mirror distortion, even if these loads are time varying.

Several large telescope projects, including WIYN, ESO VLT, and Gemini, plan to use distributed defining of the primary mirror.

Kinematic defining systems vs. overconstrained defining systems. As discussed in the section on force control vs. displacement control, a kinematic defining system constrains only six degrees of freedom. Force control systems are normally designed to be kinematic, but displacement control systems can be designed to be either kinematic or overconstrained. For example, the mirror mounted on three steel balls, mentioned above, has a displacement mount that is kinematic.

An overconstrained system has advantages in resisting wind buffeting. For the 8-meter Gemini mirrors, a *kinematic* distributed defining system will provide adequate stiffness to resist rigid body motion, and it will adequately resist mirror deformation from a *spatially uniform* wind pressure. However, some portion of the wind pressure will be spatially uneven. Because the kinematic defining system does not overconstrain the mirror it is free to bend in response to the uneven pressure input.

Gemini has developed an overconstrained distributed defining system that couples the stiffness of the Gemini mirror cell to the mirror. We refer to this as a six-zone axial defining system. This system is described in the Gemini technical report *The Distributed Defining System for the Primary Mirror*. The effect of this overconstrained defining system is to increase the stiffness and therefore to raise the lowest resonance frequency of the mirror.

When a large kinematically supported mirror is exposed to an uneven wind pressure the resulting deformation is dominated by the lowest energy bending mode; for a large thin mirror this mode is astigmatic. In the Gemini six-zone design the astigmatic bending mode resonant frequency is doubled, from 15 Hz to 30 Hz. This improves wind buffeting resistance in two ways.

First, the increased stiffness reduces the astigmatic response to spatially uneven steady state wind loads by a factor of four.

Second, the higher resonant frequency reduces the response of the mirror to high frequency wind variations. The power spectral density of wind pressure decreases rapidly with increasing frequency. The lowest resonant frequency of the Gemini primary mirror on a kinematic defining system is about 15 Hz, and there would be little resonant amplification of the wind input on a kinematic system. On the overconstrained defining system, the lowest resonant mode is a focus mode at a frequency of about 25 Hz, therefore there is virtually no amplification of the wind variations by resonance in the support system.

The dynamic response of the mirror on its support system is described in the *Gemini report on Response of the Primary Mirror to Wind Loads*.

8.2. The Design Process

The process of designing the support system for a large primary mirror involves a number of steps. First, the conceptual design must be developed, in response to the functional requirements. Section 8.3 of this report describes the evolution of the conceptual design of the support system. Second, the nominal design of the support system must be developed, in terms of locations for the support mechanisms and the nominal magnitude of forces they will exert in supporting the mirror; this work is described in Section 8.4. Third, the active optics system must be designed; the Gemini active optics system is described in Section 8.5. Fourth, the sensitivity of the system to errors in forces and positions must be evaluated. This study is described in Section 8.6. Fifth, the detailed requirements for the system hardware must be organized in a Design Requirements Document. Sixth, the support mechanisms must be designed to meet these requirements. The preliminary design of the Gemini support mechanisms is described in the reports *Mirror Supports Development* and *Preliminary Design of Lateral Support* .

In reality, all of these design activities must go on in parallel, with many iterations required in order to develop a consistent system that meets all the functional requirements. Also running in parallel is the design of the mirror cell structure, which is described in Section 10 of this report.

8.3. Conceptual Design

The conceptual design of the support system has evolved in response to the principal functional requirements, which are described in the Functional Requirements for the Primary Mirror Assembly. The requirements and the design features developed in response to them are described below.

To minimize print-through, most of the mirror weight is supported on a uniform air pressure. However, a complete air bag was undesirable for the following reasons:

- It would need to be very complex to fit around the large number of actuators that were planned.
- It could exert unwanted lateral forces under certain conditions.
- It would block heat transfer between the radiation plate and the back of the mirror.

It was decided instead to seal the outer and inner edges of the mirror with a flexible rubber seal. The air pressure required to float the desired portion of the mirror weight is approximately 3500 Pascals (0.5 psi), so the seal does not have to contain large pressure differences.

A number of different patterns for the axial support mechanisms were studied. A pattern was chosen that had 120 support points, arranged in five rings of 12, 18, 24, 30 and 36 actuators each. These numbers per ring were chosen because they approximated hexagonal close packing, that is, the arrangement would result in approximately equal radial and tangential support spacing for all rings. In addition, the pattern has 60^{o} symmetry, with no supports on the borders of the 60^{o} sectors, which makes it ideal for 3-zone and 6-zone distributed defining systems. Figure 5 illustrates the axial support pattern.

The report *Optimization of Support Point Locations and Force Levels of The Primary Mirror Support System* describes how the precise locations were chosen.

To obtain maximum stiffness the defining points have been located over ribs in the cell structure. Arranging the supports into five rings therefore only requires five circumferential ribs inside the mirror cell. This leaves wide enough spaces between the ribs to allow personnel access inside the cell for maintenance. The support mechanisms have been designed so that they can be removed from below, so that individual units can be replaced without removing the mirror from the cell.

To allow for easy disconnection of the mirror from the cell for recoating, we decided to make the support mechanisms push-only so the support pads would not have to be bonded to the mirror. In order for the active optics system to be effectively push-pull, a fraction of the mirror weight must be supported on the actuators. It was determined that active optics forces in the 350-400 Newton range would be required, therefore each support mechanism must carry at least this much weight, which represents about 20% of the total mirror weight. The active actuators can increase the applied force to about 800 Newtons maximum. or decrease it to a minimum of zero, at each support.

We decided that support mechanism print-through, caused by the 400 Newton nominal force, would be polished out at a zenith-pointing orientation. To avoid the print-through reappearing in inverse at other zenith angles, the air pressure support will be regulated to maintain the same load in the axial direction on the 120 defining points at all zenith angles up to 78^{o}, at which point the air pressure will be zero. Since the operating range of the telescopes covers zenith angles from 0 to 75^{o}, the print-through will be compensated over this entire range.

To resist mirror deformation caused by wind buffeting, a hydraulic distributed defining system has been developed. Each of the 120 supports incorporates a Bellofram rolling diaphragm cylinder. To compensate for static head pressure in the hydraulics when pointed away from the zenith, a dual chamber hydraulic design was used. one chamber in each support mechanism is connected to the appropriate zone of the distributed defining system, and the other is connected to a head pressure compensation circuit. To improve the support system stiffness, the two hydraulic systems will be preloaded against each other, so that the spring constants of both chambers contribute to the stiffness of each unit. To avoid changes in the differential pressure between zones of the support as a function of zenith angle, the preload hydraulic system will be one large zone connecting all 120 supports.

The defining system will be capable of being used in several operating modes to provide different responses depending on operating conditions. These modes include both kinematic and overconstrained operation, both of which are described in the report *The Distributed Defining System for the Primary Mirrors*.

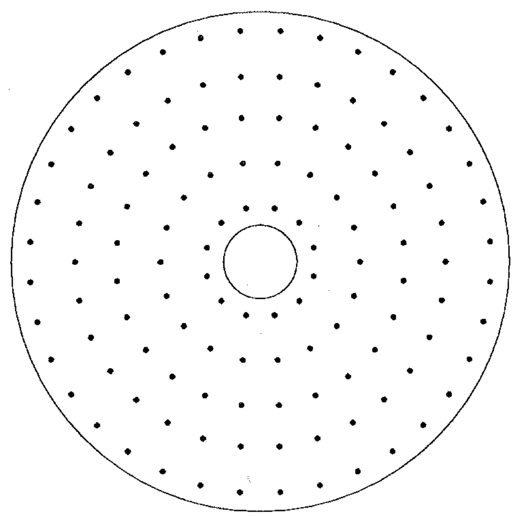

Figure 5. The chosen pattern of axial support locations.

The lateral support design is based on Schwesinger's concepts[11]. Support forces will be applied only at the outer edge of the mirror. Mechanisms on the lower side of the mirror (as pointed to the horizon) will push upwards, and mechanisms on the upper side will pull upwards. A distributed hydraulic system of astatic Bellofram cylinders will be used to ensure the correct forces are applied. Each unit will have two hydraulic chambers, with one carrying the mirror weight and the other providing preload and hydrostatic head pressure compensation. Because of the way the piping will be connected to the two chambers, preloading one hydraulic system against the other will not apply any pinching force to the mirror, but will prevent motion in both upwards and downwards directions. Two lateral support zones, left and right, will be used. Rotation of the mirror about the optical axis will be controlled by maintaining the proper fluid volume in each zone. The hydrostatic head pressure compensation system will be connected together in one zone around the entire edge of the mirror. Certain lateral supports at the left and right sides of the mirror have been divided into two dual-chamber mechanisms, so that the vector sum of the forces exerted by the two parts can take lines of action that are not physically easy to produce with a single mechanism.

8.4. Optimization of Support Locations and Forces

Preliminary finite-element analysis indicated that 120 axial supports and 72 lateral supports would be adequate. With a solid meniscus mirror both the forces and the locations of the supports can be adjusted as required to optimize the design. We made a number of design decisions to simplify fabrication, assembly and maintenance. These included:

- The axial supports would be evenly spaced on concentric circular rings.
- The lateral supports would be evenly spaced around the outer diameter of the mirror.
- No more than two sizes of hydraulic cylinders would be allowed in each support system.
- The angular line of action of the lateral supports were adjusted in some cases, in order to fit mechanisms without interference around the edge of the mirror.

The positions and sizes of the support units were optimized, within these practical constraints, by least squares fits based on finite-element analysis. This is discussed further in the Gemini Technical report *Optimization of Support Point Locations and Force Levels of the Primary Mirror Support* .

8.5. Active Optics System

The goals for the active optics system included:

- Ability to correct any errors introduced by the mirror support system
- Ability to change the conic constant of the mirror from the Ritchey-Chrétien hyperboloidal design to a paraboloid, for future use of other secondary mirrors
- Ability to correct for thermal distortion of the mirror from nonuniform temperatures and CTE distribution
- Ability to correct for other errors in the telescope, for example, gravity sag of the secondary mirror on its 3-point tip-tilt mount.

It was determined that approximately 150 active actuators would be adequate to accomplish these goals. Because of concern for the susceptibility of the support system to small angular errors in the lateral supports which could produce relatively large force errors at the edge of the mirror in the normal direction, it was decided to include an active optics actuator at each lateral support location. By including an active optics actuator at each axial and lateral support, a total of 192 actuators are available. This also provides an efficient means of correcting any figure errors caused by changes in the force exerted by the air pressure seals.

Finite-element analysis has confirmed that this system will be capable of accomplishing the goals stated above. This analysis is described in the report *Active Optics Capability of the Primary Mirror System*. Further studies are planned to see if it would be possible to reduce the number of actuators, to save cost and reduce system maintenance.

8.6. System Sensitivity to Tolerances

An extensive study has been conducted to evaluate the sensitivity of the support system to errors in:

- mechanism locations
- applied forces
- applied moments
- unintended "cross talk" forces and moments in orthogonal directions
- forces exerted at the air pressure seals
- pressure in the air support system

The study showed that the active optics system would be able to correct these errors very efficiently, therefore the allowable component tolerances are all quite achievable. This work is described in the report *Response of the Primary Mirror to Support System Errors*.

It is interesting to note that the advantages of the slightly overconstrained defining system to resist wind loading also carry over to the resistance to support force errors. The six-zone axial defining system is approximately 4 times as resistant to normal force errors as a fully kinematic three-zone system.

9. Thermal Control System

The mirror thermal control system must satisfy two types of functional requirements. The temperature of the front surface of the mirror must be kept in thermal equilibrium with the ambient air in order to avoid mirror seeing and to avoid dewing, and thermal distortion of the mirror from temperatures must be minimized.

As mentioned in Section 4, the Gemini primary mirror thermal control system will use a radiation plate behind the mirror. Its temperature will be controlled by a water-glycol mixture pumped through coils built into the plate. The concept is not new; in fact, it was used for a while at the 100-inch Hooker Telescope on Mount Wilson. As David O. Woodbury wrote in his book, <u>The Glass Giant of Palomar</u>,

> “A system of coils carrying cold water was installed at the back of the disk, connected through a rubber hose to a circulating pump. The astronomers were to make a wise guess as to the temperature of the coming night and all day long the mirror would be held at that point. Then when darkness fell the telescope would be ready to go to work at once."

The system did not work very well, because during the day the plate glass mirror had a cold coil behind it and warm daytime air in front, and its high CTE caused it to warp. David Woodbury reports,

> "The cooling coils were soon abandoned. It was found better to keep the observatory tightly closed all day and the mirror heavily insulated in a cork-lined

chamber. Thus the glass mass was held at the average night temperature constantly, and the time lost in waiting for it to settle became negligible."

The reason this approach is feasible for Gemini is that the Corning ULE™ glass in the Gemini mirrors has a CTE more than two orders of magnitude lower than the plate glass in the 100-inch mirror. Gemini will also air condition the telescope enclosure during the day to maintain night-time temperature.

Other options were considered, including controlling the mirror temperature by blowing air on its back surface. We decided not to adopt that approach because it complicated the air pressure support, and would have been difficult to implement without introducing vibration source such as blowers into the mirror cell assembly.

The radiation plate thermal control system has the following advantages:

- It is very uniform -- this complements the mirror design, which has a uniform cross section thickness
- It can transfer sufficient heat by radiation with a AT of only a few degrees
- The radiation plate will be inside the air pressure support seal (described in Section 8), therefore condensation can be avoided by simply using dry air in the support system
- It can be used during observations without need for fans, blowers or other vibration sources
- It shields the mirror from heat sources in the cell

Figure 6 shows the relationship between the mirror, radiation plate, thermal insulation, support mechanisms, air pressure seals, and cell structure.

In addition to the radiation plate system, a second system is planned as an enhancement to improve the speed of response. This second system is based on the realization that it is the *surface* temperature that must be in equilibrium with the air. It is not necessary to change the temperature of the entire mirror at a rapid rate, all that is needed is to implement a system that can change the temperature of the front layer of glass.

The approach we have adopted involves running an electrical current through the optical coating to provide controlled heating of the front surface. The substrate temperature will be maintained by the radiation plate at a temperature slightly below ambient, and controlled heating will keep the front surface in equilibrium with the air. Heat transfer studies have shown that as little as 40 watts per square meter of heat input will change the surface temperature one degree C in 18 minutes.

This system is not really new. There have been many applications where windows have been heated by electrically conductive coatings, for aircraft wind screens, architectural glass, Apollo astronaut sun visors. etc. This is simply a new application.

Experiments have been conducted at Rutherford Appleton Laboratory in the United Kingdom to assess the feasibility of this approach. The results are described in the report *Design Study and Tests of a Thermal Management System for the Primary Mirrors.* So far the results have been good. No coating degradation effects have been caused by the current, and theoretical calculations about the heating have been confirmed. More work is needed on electrode design to ensure even heating immediately adjacent to the electrodes at the edge of the mirror, but in theory there is no reason the heating cannot be quite uniform over the entire mirror surface.

The performance of the proposed Gemini thermal control system has been evaluated by extensive heat transfer analysis using several years of night time temperature data from Mauna Kea, to evaluate the ability of the system to minimize mirror seeing effects. Analysis has been done for the radiation plate baseline design, as well as for the enhanced design including the coating heating capability. Both systems show promise of working very well, with the enhanced system having clear performance advantages. This analysis is described in Appendix A of the report *Design Study and Tests of a Thermal Management System for the Primary Mirrors*.

Studies have also been conducted to predict the level of mirror distortion that could be produced by nonuniform temperatures in the glass. In general, the thermal control system should minimize nonuniform temperatures across the mirror, but we expect gradients through the thickness of the mirror to be produced.

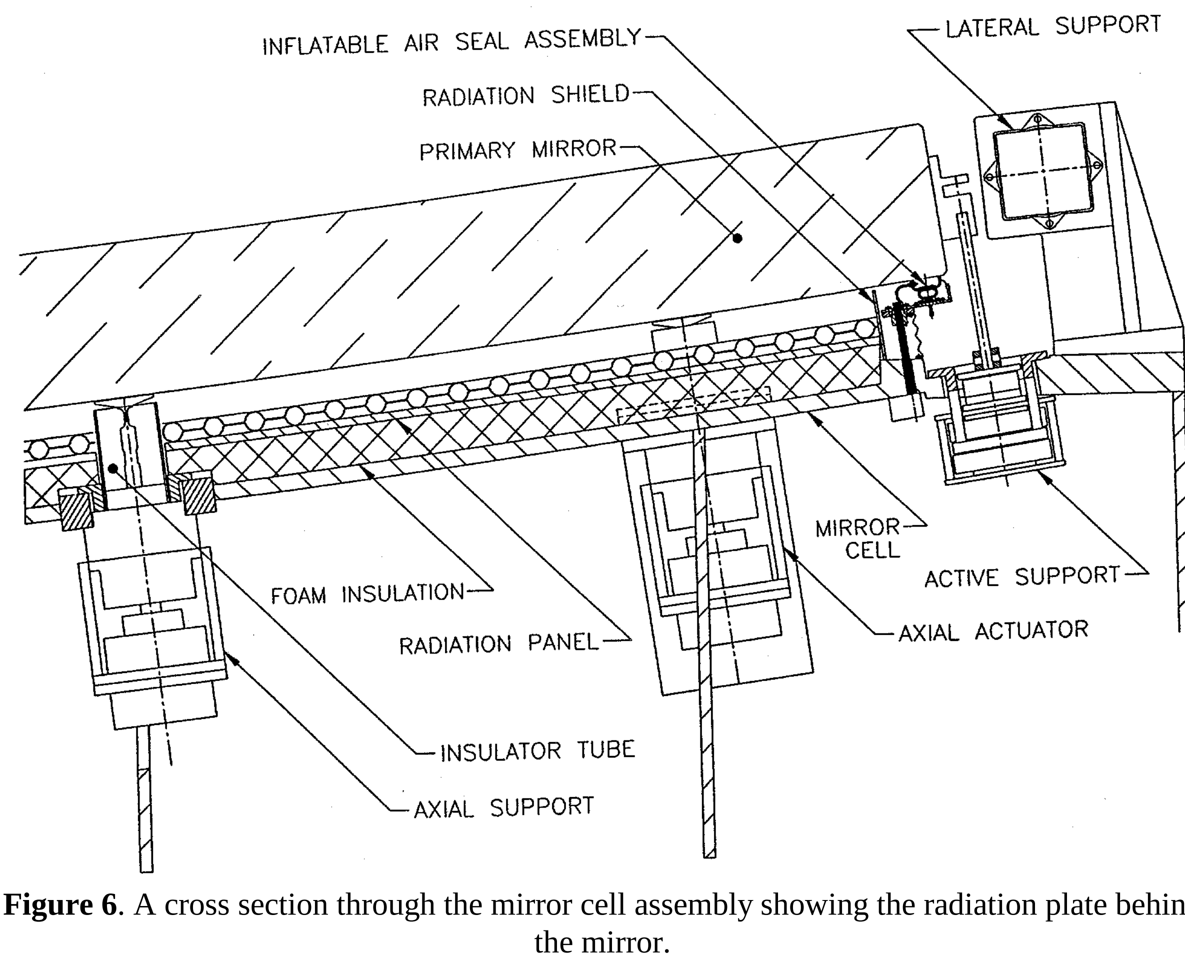

Figure 6. A cross section through the mirror cell assembly showing the radiation plate behind the mirror.

The CTE of the ULE™ glass is very low. At the telescope operating temperature it is about -40 parts per billion per degree C. Analysis has shown that this will allow gradients of several degrees C without exceeding the small error budget allocation given to thermal distortion. This analysis is described in the report *Distortion of the Primary Mirror Due to Temperature and CTE Nonuniformity*.

10. Mirror Cell Structure

The mirror cell has been designed to be relatively stiff, to carry the weight of the mirror and a large complement of Cassegrain focus instruments. Even with these weights included in the analysis, the current design of the mirror cell has a lowest resonance at about 40 Hz. The zenith pointing gravity flexure of the top surface of the mirror cell on the 4-bipod support will be 42 microns.

To minimize gravitational flexure of the mirror cell, it is supported on the telescope center section at a 60% radius, almost as though it were a mirror. The mirror cell is attached to the telescope by four bipods. An alternative design was investigated having three bipeds to support the cell, but its gravitational flexure was found to have a larger effect on the mirror figure relative to the error budget, when used with the six-zone axial defining system .This study is documented in the report *Primary Mirror Cell Deformation and its Effect on Mirror Figure Assuming a Six-zone Axial Defining System*.

As described in Section 11 below, the mirror cell will carry the Cassegrain instrument rotator, deployable optical correctors above the Cassegrain focus, covers that can be closed to protect the acquisition and guiding system during the daytime, the cable wrap for the Cassegrain instruments, and the primary baffle tube. It will contain a VME crate to hold control electronics. It will support the mirror axial and lateral supports and defining systems, as well as earthquake safety restraints. It will house the mirror thermal control system. The main components of the mirror cell assembly are shown in Figure 1. The structure must be designed to be compatible with all these subsystems, and to allow personnel access inside the structure for maintenance.

11. Interfaces

The primary mirror cell assembly interfaces with many of the systems in the telescope, including:

- The telescope structure center section
- The primary mirror covers
- The Cassegrain rotator and instrument support structure, with its cable wrap
- The optical f/6 and f/16 correctors
- The central f/6 and f/16 baffles
- The telescope control systems
- The mirror cell removal cart
- The primary mirror cleaning system

Interface control documents are being developed to define all the interfaces between the mirror cell and the other telescope subsystems.

12. Summary of Design Features

Appendix A contains layout drawings of the primary mirror cell assembly. The cell structure is a steel weldment designed to have high stiffness and resonant frequencies. The mirror supports are attached to the cell at locally stiff points in the structure. The ribs of the structure are far enough apart (about 70 cm) to allow personnel access for maintenance. Mounted to the inside diameter of the cell are instrument safety covers and the attachment mechanism for the primary baffles. Deployable optical correctors will sometimes be installed inside the central hole of the cell, at a level just above the Cassegrain rotator bearing.

The mirror support system consists of an air pressure support plus axial and lateral distributed hydraulic systems. An active pneumatic actuator is also associated with each axial and lateral support. Axial support mechanisms can be replaced from below the mirror, and lateral support mechanisms can be replaced by personnel at the outer edge of the cell, without removing the mirror from the cell. The axial support mechanisms are not bonded to the mirror, to facilitate mirror removal for recoating. An ambient temperature cooling water loop will remove heat generated by the active optics actuators.

The thermal control system consists of a radiation plate located behind the mirror, and a coating heating system on the mirror front surface. Insulation behind the radiation plate and around the cooling water pipes will minimize the effect of the thermal control system on the temperature of the cell structure.

The entire cell assembly along with the mirror cell cradle can be lowered onto a handling cart to remove the cell from the telescope.

13. Acknowledgments

Many individuals have contributed to the conceptual design of the Gemini primary mirror assembly, particularly Dale Circle, Joe DeVries, Justin Greenhalgh, Eric Hansen, Eugene Huang, Jim Lidbury, Brian Mack, Earl Pearson, and a number of other engineers on the Gemini Project staff, at Royal Greenwich Observatory and at Rutherford Appleton Laboratory. As stated in the introduction, the designs also draw heavily from work done for other telescope projects, particularly the ESO VLT, Subaru, and WIYN Projects.

The author would like to acknowledge the help of Dale Circle and John Roberts in preparing the figures for this report.

14. References

1. Rene Racine, "Mirror, Dome, and Natural Seeing at CFHT," pp. 1020-1032.
2. M. Iye, R. Noguchi, Y. Torii, Y. Mikami, Y. Jamashita, W. Tanaka, M. Tabata, and N. Itoh; "Active Optics Experiments with a 62cm Thin Mirror," pp. 929-939.
3. L. D. Barr, J. Fox, G. A. Poczulp and C. A. Roddier; "Seing Studies on a 1.8m Mirror," pp. 492-506
4. C. M. Lowne, "An Investigation of the Effects of Mirror Temperature Upon Telescope Seeing," pp. 249-259.

5. Lorenzo Zago, "The Design of Telescope Enclosures for the VLT."
6. G. A. Harding, B. Mack, F. G. Smith and J. R. Stokoe, "On the Avoidance of Bad Seeing Conditions Within Telescope Domes, pp. 241-247
7. J. Bouvier, "2nd Dome Air Cooling Experiment (May 29-June 19, 1991) - Impact on Image Quality.
8. M. Iye, T. Noguchi, Y. Torh, Y. Mikami and H. Ando, "Evaluation of Seeing on a 62cm Mirror," pp. 712-722.
9. Paul Schecter, private comunication, 1993.
10. L. Noethe, "Axial Support System -- Push/pull versus Push and Passive/Active Versus Active, ESO report VLT-TRE-ESO-11120-0004, 11112/92.
11. G. Schwesinger, J. Mod Optics, 38(8), pp. 1507-1516, 1991.

APPENDIX A

Primarv Mirror Assemblv Layout Drawings