UPDATE ON THE 18m APERTURE MULTI-MIRROR TELESCOPE CONCEPT

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1. Introduction

At the present time one of the major activities within the international astronomical community is the preliminary design of large optical/infrared telescopes. During the last few years workshops, conferences, and many discussion groups have been assembled to evaluate alternative telescope concepts. Segmented mirrors, multi-mirrors with dilute or filled apertures and arrays of smaller telescopes have been discussed and all give the increase in the collecting area required for the next generation of astronomical instrumentation.

A significant proportion of the concepts use one or more large primary mirrors. In the past British optical telescopes have used primaries of an aspect ratio of diameter to thickness of 6:1 and 8:1, while the United Kingdom infra-red telescope has an aspect ratio of 16:1 and has been figured to an optical specification. The use of a primary mirror of aspect ratio greater than 16:1 will tax the analyst's ability to predict the deformations of the mirror surface at all positions in the sky. The optical performance of the telescope relies upon the accuracy of this prediction and the designers ability to provide a suitable support system for the mirror when the mirror is contained in the telescope cell. Telescopes of 7.5m or 18m aperture may require large prime focus correctors and use sophisticated beam combining optics. The refracting elements required for a number of the proposed designs are of large diameter and have aspect ratios greater than 16:1. The optician is able with modern computer aided techniques to design very sophisticated optical solutions. However he must now also be able to determine the deflections and internal strains set up within the materials of the refracting and reflecting elements.

Early in the design of the support systems for the 2.6m Isaac Newton and 4.2m William Herschel telescopes large finite element generation programs were developed to provide accurate mathematical models of the primary mirrors. This enabled us to calculate the deflections of the reflecting surface and also to determine the stress and strain distribution within the mirror materials. It is of fundamental importance to develop the analytical tools which may be used to calculate the deflections of the optical elements. It must be proven analytically that it is economically possible to support a relatively thin, light weight

Proceedings of the IAU Colloquium No. 79: "Very Large Telescopes, their Instrumentation and Programs", Garching, April 9–12, 1984.

https://doi.org/10.1017/S0252921100108887 Published online by Cambridge University Press

primary mirror and the large optical elements now proposed for a 7.5m telescope.

We have therefore, in the last year, concentrated on the development of the finite element programs and models to enable us to determine the stresses and deflections of complex refractive elements, honeycomb and homogeneous primary mirrors.

This paper is divided into 4 sections. Section 1 deals with the axisymmetric finite element models set up to calculate the stresses and deflections within the material of a 7.5m primary mirror and determine initial radial positions and forces required for the axial supports.

Section 2 contains a brief description of the large three-dimensional finite element models used in, and presents the results of, an analysis carried out to determine the deflections of the front element of the 4.2m William Herschel prime focus corrector.

Section 3 provides a description of the analysis of a Borosilicate Honeycomb mirror, and contains the results of the local deflections within the honeycomb structure.

Section 4 describes the proposals currently under investigation for the design of the support system of a 7.5m primary mirror and describes the development to be carried out so that an efficient and accurate finite element model can be constructed to compute both the structural and thermal deflections.

Section 1.

The capacity of a finite element model to accurately produce a map of the internal stress distribution throughout the material of a primary mirror determines the accuracy of the strain distribution and therefore the deflections within the material and of the reflecting surface. The degree of stress diffusion from the axial and radial support points has a direct influence on the magnitude of the deflections. The aspect ratio, number of rings of axial supports and number of radial supports used for the primary mirror produce a very complex internal stress distribution. The paper, reference (1), contains a description of this stress diffusion problem. The level of accuracy of the stress distribution determined in a finite element model is influenced by the numbers and positioning of the elements set up to model the mirror and its supports.

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If rings of pneumatic cylinders are used for the axial support system then the reactions, number of cylinders, radial positions of the cylinders and the pressures in the cylinders can be determined using accurate axi-symmetric finite element models. Fig (1) shows the generated model, containing 3000 axi-symmetric elements, of a 7.5m diameter, 12:1 aspect ratio primary mirror. The model is used to optimise the radial positions of the axial support points. Fig (2) shows the "Magnified Graphically" deflected mesh of the mirror when supported on 4 rings of pneumatic cylinders. The deflections (under the influence of gravity only) are less than $\lambda/100$ (where $\lambda = 0.5\mu m$). From this analysis it is clear that 12:1 aspect ratio mirror only requires 3 rings of axial supports. The numbers of cylinders required at each radius will be determined and the forces at each axial support point, applied by the pneumatic cylinders, is then applied to the model to determine the degree of focus shift caused by the mirror deflections. The model is also used to determine the levels of internal stress for the design of the mirror handling equipment used for lifting the mirror into the telescope and the aluminising plant.

Section 2.

The preliminary design of the axial support system for a primary mirror can be carried out using the simple axi-symmetric models as described in the previous section. However the design of the radial supports for a thin meniscus type of mirror or a large, short focal length, flat backed, parabolic front surface mirror, is increasingly difficult.

The radial supports usually induce internal poisson ratio related deformations which are usually negligible for a thin mirror and can be easily estimated using various simple formulae and the results from other authors' papers. The deflections due to bending and thermal stability across the diameter are a much more difficult problem. Any temperature variation across the diameter of a meniscus mirror seriously degrades the quality of the reflecting surface and a zero coefficient material may have to be used.

The positions of the radial supports relative to the centre of gravity of the mirror determine the amount of transverse bending taking place in the mirror when the telescope tube is in the horizontal position.

Because it is impossible for the radial supports of a thin meniscus mirror to have their line of action through the centre of gravity then the mirror will be subject to bending moments. This also applies to mirrors of low focal ratio, below f/2, and many of the large front elements of corrector lenses.

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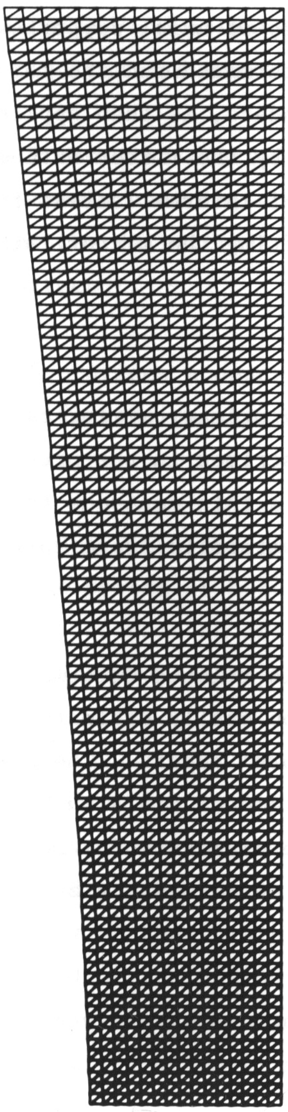

Fig. 1 3000 ELEMENT AXI-SYMMETRIC MODEL OF 7.5m DIA. MIRROR

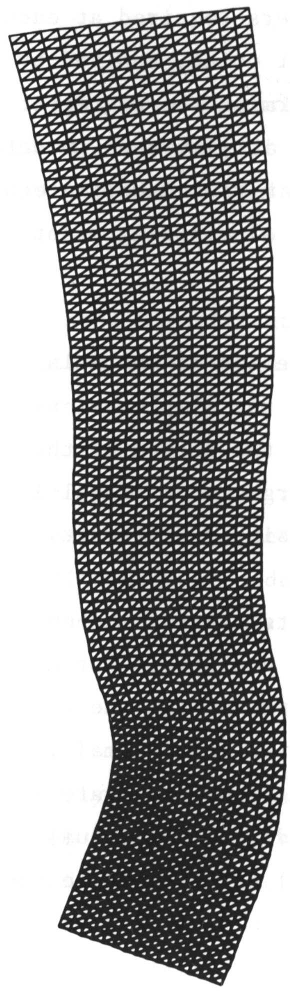

Fig. 2 'MAGNIFIED GRAPHICALLY' DEFLECTED MESH WHEN SUPPORTED ON 4 RINGS OF AXIAL SUPPORTS

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Fig (3) shows the finite element model now under development to determine the deflections induced by both the radial and axial supports of a 7.5m f/2, aspect ratio 12:1 parabolic primary mirror.

This model is to be used to carry out the majority of the analysis required for the final design of the support system, a larger finite element model will then be generated, using twice the number of elements, for the remaining optimisation. This reduces the very costly computational time required for these programmes. Figure (4) shows the finite element model used to calculate the deflections of the front element of the 4.2m William Herschel telescope prime focus corrector. This corrector is of a relatively low aspect ratio and approximately 50 cm diameter. The deflections, when the telescope tube is in the horizontal position, are given in fig (5). In this case they have been found to be negligible. Optical elements of aspect ratios greater than 20:1 and of 1m diameter proposed for the correctors for telescopes of 7.5m aperture will require more sophisticated support structures. A detailed finite element analysis will have to be carried out to determine their internal strains and surface deflections.

Section 3. Local deformation of a honeycomb structure

There are two methods of reducing the weight and cost of a primary mirror and both are influenced by the manufacturing constraints imposed by the type of material chosen for the blank.

The first method is to reduce the overall mass by reducing thickness which usually results in a meniscus type of mirror, and we have investigated the problems of supporting and lifting thin meniscus primary mirrors. These investigations have indicated some difficult areas in the optimisation of the support positions for mirrors of 25:1 aspect ratio.

The axial support optimisation work showed that relatively small forces could deform the mirror and the mirror deflections were very dependent upon the correct support forces and wind loading.

The second method used to reduce the mirror weight is the production of a lightweight blank of honeycomb construction and there are many reasons why mirrors of this type, with high stiffness to weight ratios, should be used. These are as follows:-

1) Easier optimisation of the axial support system because of the high stiffness to weight ratio.

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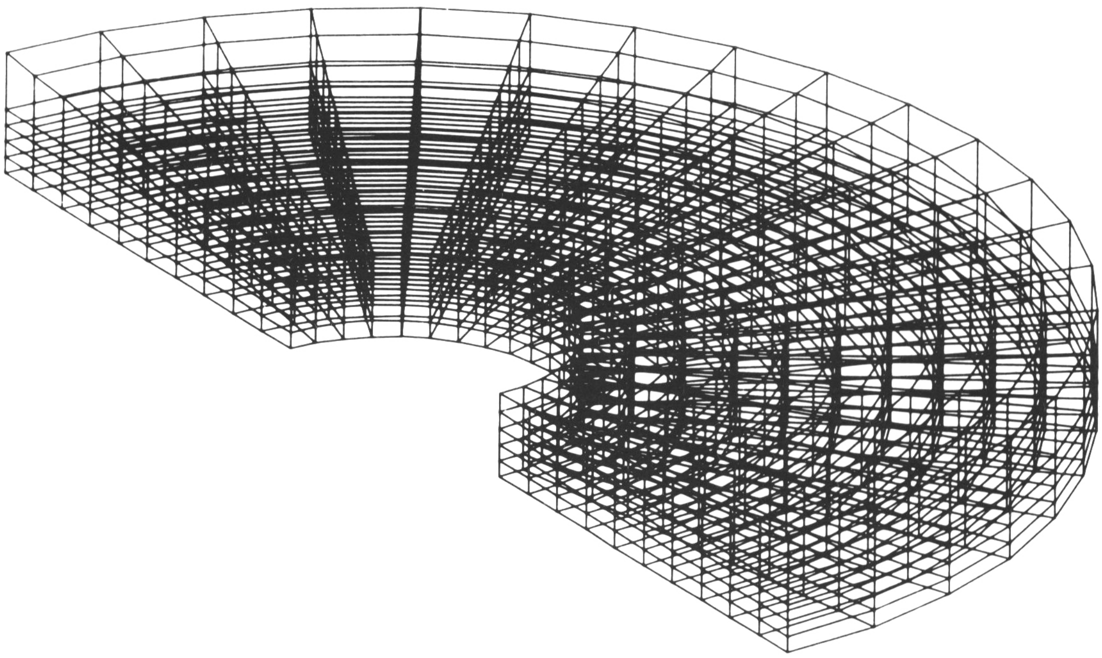

Fig. 3 MODEL BEING DEVELOPED TO DETERMINE EFFECT OF AXIAL AND RADIAL SUPPORTS

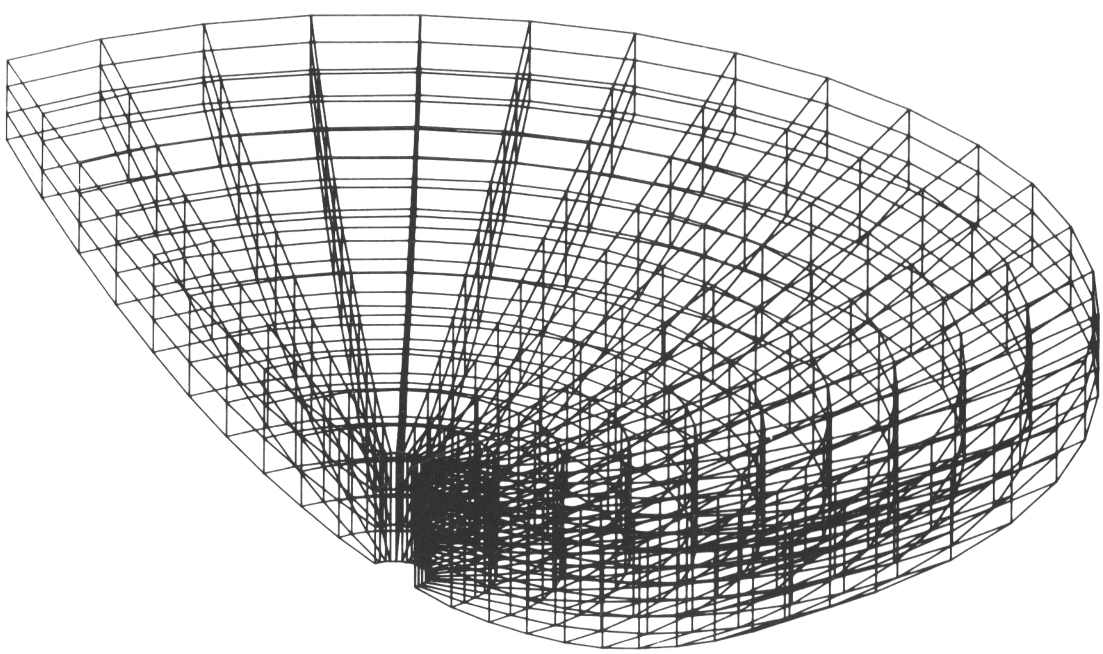

Fig. 4 MODEL OF PRIME FOCUS CORRECTOR FRONT ELEMENT

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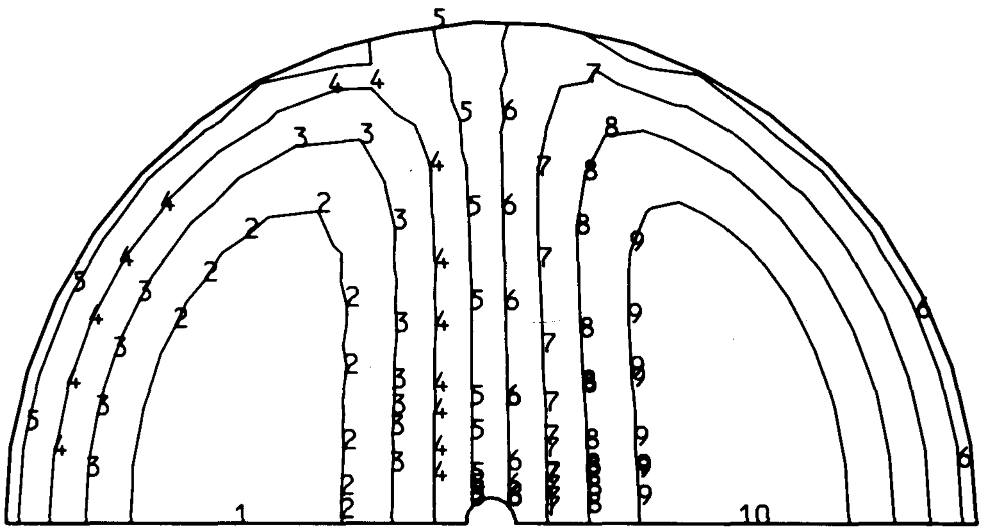

Fig. 5 CONTOUR MAP SHOWING DEFLECTIONS OF CORRECTOR LENS ELEMENT WITH TUBE VERTICAL

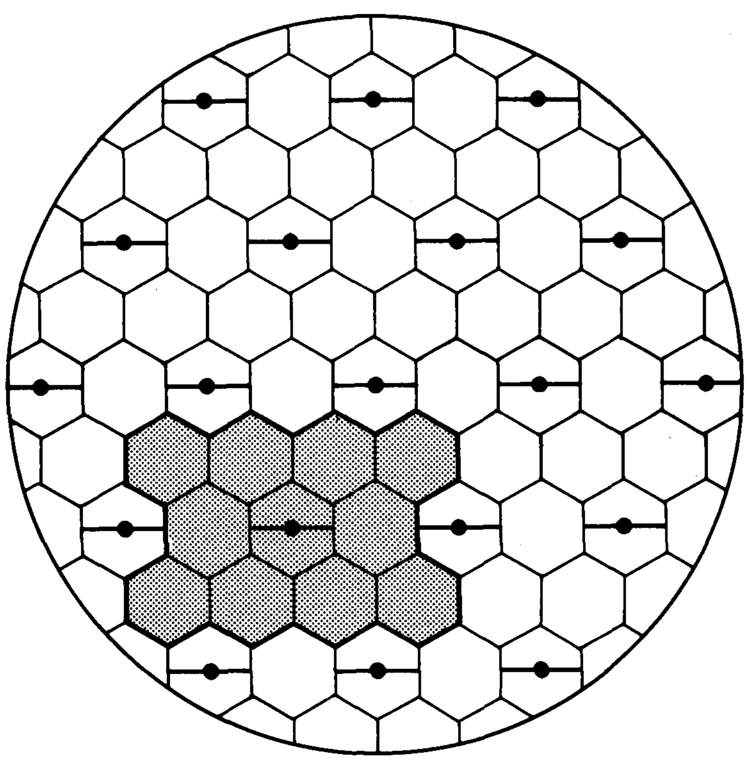

Fig. 6 CELLS AND SUPPORT LOCATIONS OF 1.8m HONEYCOMB MIRROR BLANK

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2) The number of radial supports required is minimised and the line of action of equal transverse forces passes closer to the plane of the centre of gravity of the mirror. This reduces the bending moments in the mirror.

3) Cheaper figuring costs, if the thermal problem is solved.

4) Easier handling, this has a major affect on figuring plus handling costs, aluminising is also made easier as complex support systems are not required in the tank.

5) The axial supports can be pneumatic and therefore simple, radial supports are also simplified.

6) Deflection due to wind loading although a major problem is minimised, the mirrors are stiff enough to withstand reasonable wind loads with minimum deflections. This reduces the need for adaptive optics control of the mirror surface.

If a large optical/infrared telescope is to be built then one of the major cost factors is the design of the primary mirror. A honeycomb type of construction has certain advantages, but the material Borosilicate, which is suitable for its manufacture, has a positive coefficient. If it were possible to construct a honeycomb mirror economically, using a zero coefficient material, then that would be the optimum choice. However, it has been shown possible to manufacture small Borosilicate mirrors at the University of Arizona.

Fig (6) shows the hexagonal cells and support locations of a 1.8m mirror blank, Steward Observatory is responsible for the overall design and mirror fabrication, KPNO for the making of the cell and supports, and Optical Sciences Centre for the mirror figuring.

A very large finite element model would be required to simulate a full mirror and therefore in order to reduce the computational effort the local deformations, induced by the supports, were calculated by analysing a section of the mirror. The outline of the section under analysis is shown on Fig. (6).

Fig. (7) shows the finite element model of this section of the honeycomb. The top surface consists of three layers of isotropic 20 node brick elements to simulate the face plate. Each of the honeycomb cell walls was modelled using 12 plate elements, twelve elements were chosen so that realistic loading conditions could be applied to simulate the radial support forces when acting at the mid-plane of the mirror. A single set of plate elements were used to model the rear surface plate of the mirror. With this model deflections due to figuring loads and surface deformation due to the axial and radial supports were calculated.

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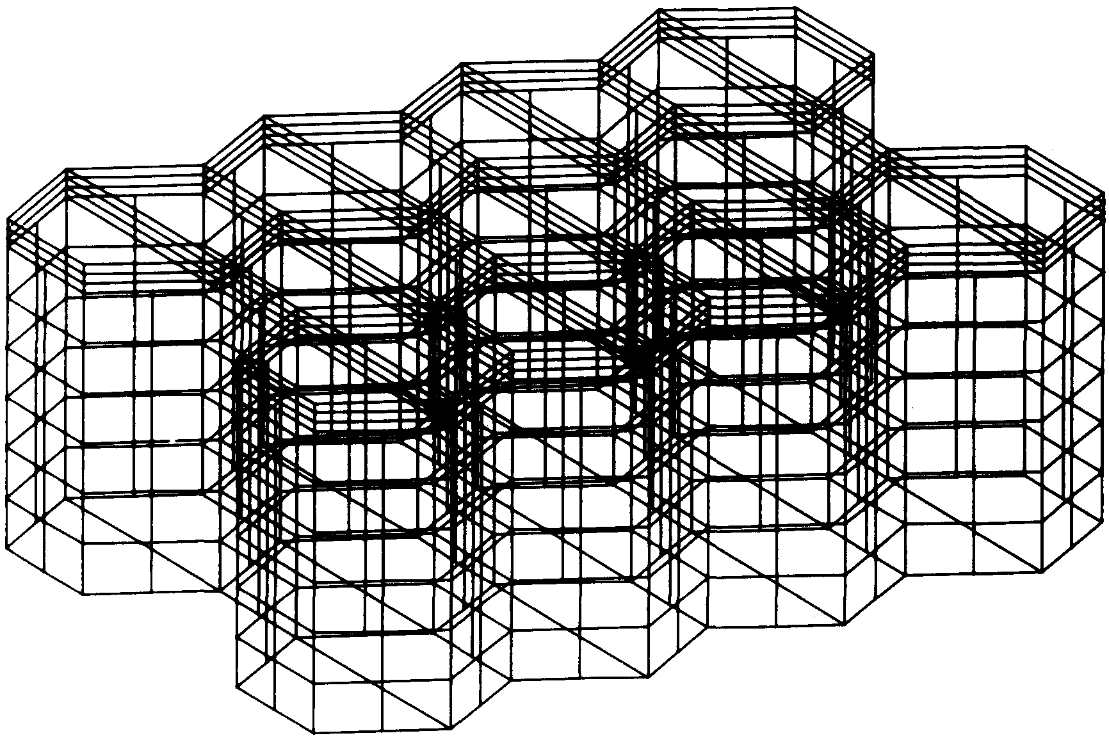

Fig. 7 MODEL OF SECTION OF 1.8m HONEYCOMB MIRROR BLANK

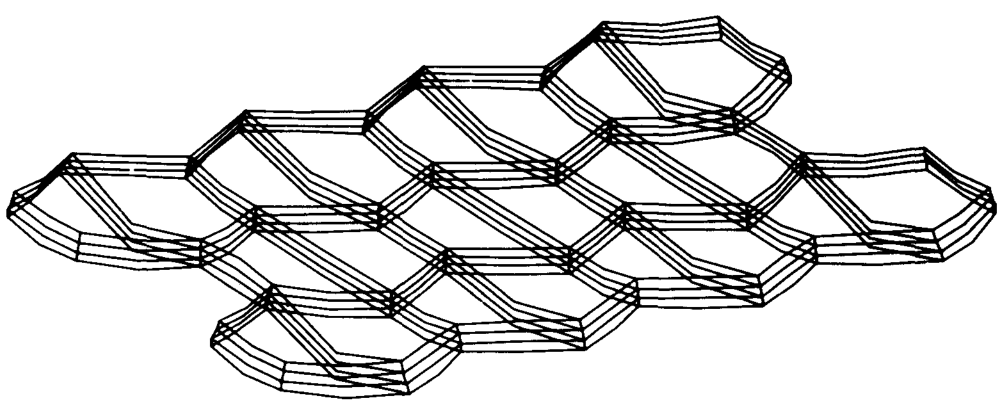

Fig. 8 DEFORMED SHAPE OF TOP SURFACE DUE TO FIGURING PRESSURE

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Fig. (8) shows the deformation of the top surface of the honeycomb when under the influence of Gravity and an estimated figuring pressure of 0.2 psi. The rear surface of the mirror was constrained in the vertical direction. The central cell of this section is representative of the majority of the cells in the mirror whilst the outer cells represent those near the edge of the mirror.

At the centre of a cell near the mirror edge the resulting maximum depression is .115 μm, while the top of the cell wall moves .086 μm, this gives a .029μm variation which is equivalent to $\lambda/17$. The corresponding central cell depressions are .09 μm and .083 μm giving a .007 μm variation ie $\lambda/71$.

This result is influenced by the outer cell wall having no constraints in the transverse direction. The face plate is therefore allowed to deflect and tends to pull in the outer edges. This should be eliminated by stiffening the mirror wall on the external diameter.

Axial Flotation Local Deformation

In the finite element model the outer edge of the section is assumed to be fixed in the vertical direction, gravity is applied throughout the material and the point loading due to the supports is presented as two forces, one in each cell wall of the honeycomb.

Fig (9) shows the deflections of the Section, they are given as deflections relative to a fixed support point. Deflections induced by both rear surface and mid plane supports were calculated. In the case of the mid plane axial support the variation across the mirror surface, "quilting" is .034 μm, $\pm\lambda/30$. The support induced shear deformation is .132μm. The deflection of the top surface when the supports are on the rear surface is larger in magnitude because of the increased shear. The variation across the surface is .019μm $\pm\lambda/52$. This indicates that there is a significant advantage if the axial flotation supports are placed on the rear surface rather than at the mid plane. Although for this mirror the deflections may still be within specification, this would also be the case for any solid 6:1 aspect ratio primary mirror.

Surface deformation due to Radial Supports.

Fig (10) shows the Local deformation of the honeycomb when under the influence of a radial support force. When the telescope points to the horizon Fig (10)(a) shows the transverse deflection of the top surface. The cell wall shear deformation is .202μm which would be the same for all cells. Fig (10)(b) shows the vertical deflection of the reflecting surface. Some twisting occurs and is .025μm; equivalent to peak to peak deviation of $\pm\lambda/40$.

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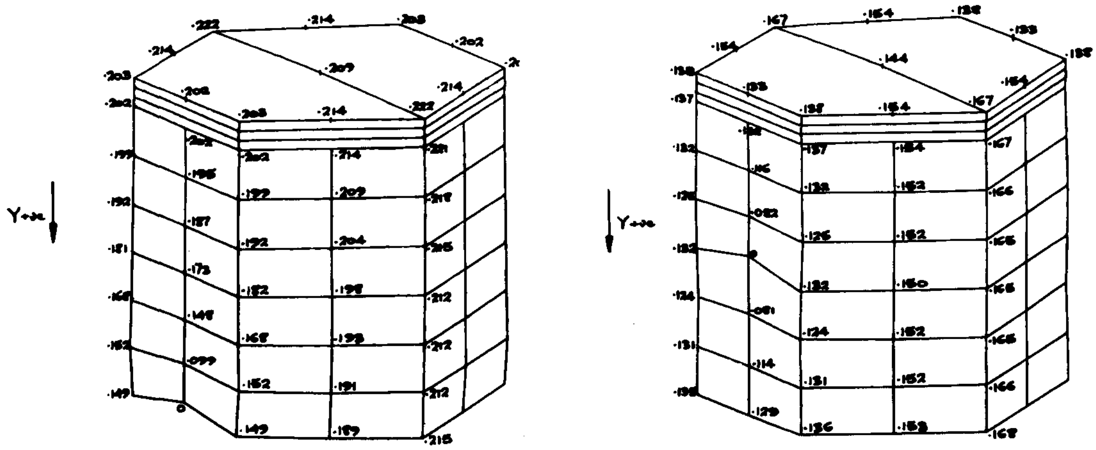
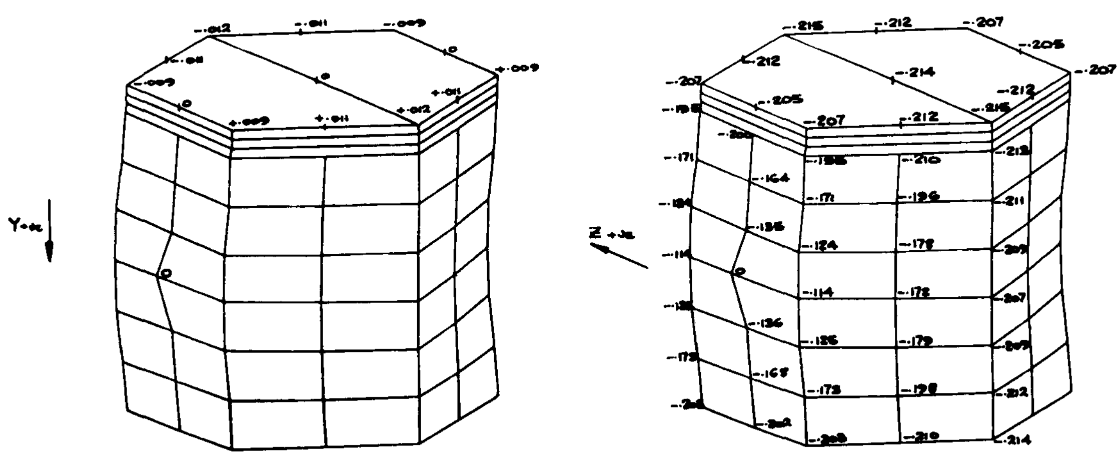


(a) (b)

Fig. 9 DEFORMATION OF CELL WALLS DUE TO
(a) AXIAL SUPPORT ON REAR SURFACE
(b) AXIAL SUPPORT AT MID PLANE



(a) (b)

Fig. 10 DEFORMATION OF THE REFLECTING SURFACE DUE TO RADIAL SUPPORT
(a) OUT OF PLANE
(b) TRANSVERSE

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Evaluation of the Equivalent Youngs Modulus and Density

To obtain an approximate value of the Youngs Modulus for the mirror the section model, as shown in Fig (7), was fixed in all directions, on all nodes, on a short edge. This was equivalent to an encastre beam. Forces were applied at the extreme edge and the Equivalent Youngs Modulus calculated using simple Beam formula. Similar tests of deflection for uniformly distributed load were also applied and the results indicated that the beam behaved as if it were manufactured out of a solid of Youngs Modulus 1.5 x 10^5Kgf/cm., approximately one quarter of the Modulus of solid Borosilicate.
This section has discussed the results of the deflection analyses of a honeycomb mirror under gravity conditions. No attempt has been made to investigate the difficult problem of the deflections due to thermal variation throughout the honeycomb structure. To be able to investigate the thermal problems of the mirror a large accurate finite element model will have to be developed. This model will need to be able to detect small deflections due to temperature variations throughout the cell structure.

It is difficult to model the full mirror using the same number of elements for each honeycomb cell as used in the analysis of the representative section. Both time and computational units required are excessive and therefore prohibitively expensive. However the section model may be modified to determine the minimum number of elements essential to give an accurate solution.

It is intended to continue to optimise the model by reducing the number of elements in the top surface and cell walls.

Thermal and Static loads will then be applied to the honeycomb model for a 7.5m telescope and the results compared to those of the accurate section model. The minimum number of elements for a realistic local solution can then be determined. A full analysis of a honeycomb mirror will then be carried out under thermal, figuring and static loading conditions.

Section 4

A 7.5m diameter primary mirror of 12:1 aspect ratio may require only 3 rings of pneumatic cylinders for the axial supports. The honeycomb structure as described in section 3 has Youngs modulus in bending equivalent ¼ of of the modulus of solid Borosilicate. The weight of the honeycomb is also of ¼ solid Borosilicate.
Therefore 3 dimensional homogeneous models can be used to initially investigate the proposals for the axial and radial supports. The internal loads induced by

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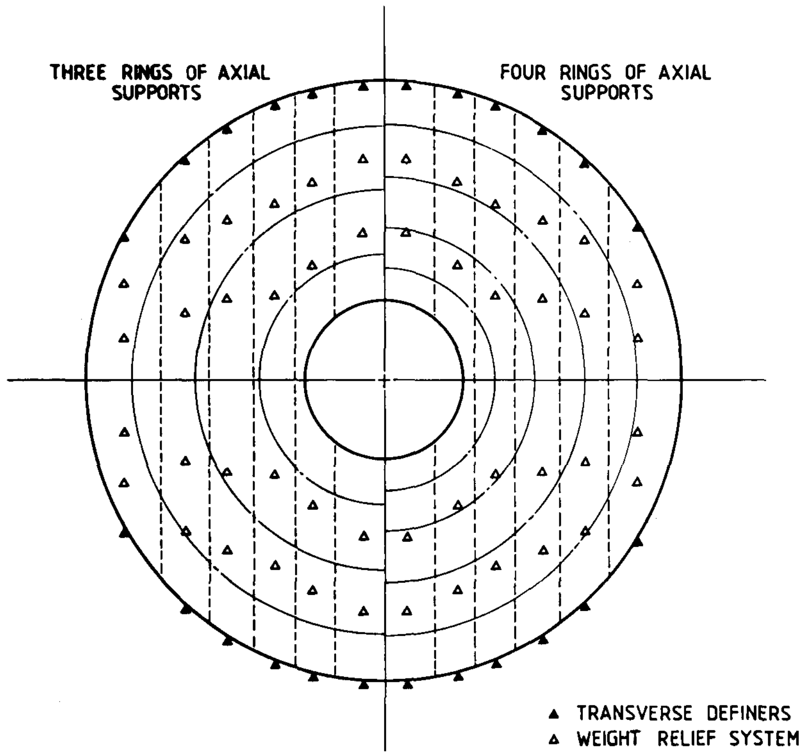

Fig. 11 PROPOSED SUPPORT SYSTEMS FOR 7.5m HONEYCOMB MIRROR

the radial supports of a honeycomb mirror produce bending loads on the internal structure and therefore have to be minimised. The result is that a honeycomb mirror will probably require a larger number of radial supports than if the mirror was homogeneous.

We have provided a transverse support system for the 4.2m primary mirror, the arguments for such a system are provided in paper (1). A system of transverse supports for a 7.5m primary as shown on fig (11) is now under investigation.

The transverse/radial supports will consist of a set of master supports attached to the outer diameter of the mirror. These will probably take approximately 1/3 of the mirror total weight. The other sets of transverse supports positioned with in the honeycomb structure will provide slave load reducing supports. It has not been decided whether these will be totally load reducing or have any defining influence on the mirror. A large homogeneous finite element model is now

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set up to investigate the deflections due to the radial support system. The local deformations calculated in the section model are to be added to the results of the homogeneous model to give an estimate of the deflection before a full honeycomb structured model is set up to calculate the deflection of the front surface of the primary due to gravity, wind and thermally induced loads.

References

1 B Mack, Appl. Opt. 19, 1000 (1980)

DISCUSSION

R. Wilson to B. Mack: I would like to congratulate you on taking up the matter of flexure in future correctors which could, indeed, be a major problem with some of the designs we have seen. In a recent ESO preprint (No. 281) to be published soon in A&A, Cao and I have referred to a flexure problem we had with our quite conventional Wynne-type PF corrector for the ESO 3.6m telescope. As a consequence of the Wynne scaling law mentioned by Harvey Richardson, our two Triplet correctors for our quasi RC primary of relatively high excentricity are fairly large (360mm front lens diameter), which is favourable for field correction. However, this led to space constraints in the corrector mounts and these were not as robust as we might, in retrospect, have wished. Overclamping at the edges of the front and middle lenses combined with mount flexure led to bending distortions causing asymmetrical field coma sometimes three or more times larger than the theoretical aberrations of the system. After much work to analyse the source of the problem, it was possible largely to eliminate these constraints and our Triplets are now consistently giving excellent results. But this practical example in an existing system of modest size and lens steepness shows the sort of problems we might get with future correctors for VLT's with very steep primaries.

J. Nelson: I commend you on your analysis of the honeycomb mirror and support system. What are your results of the analysis of the deflections of the large refracting elements that you hinted at?

B. Mack: The William Hershel Telescope prime focus corrector is approx. 45cm diameter with a central thickness of 4cm. The deflections are very small, 10^{-4}-10^{-5} microns. However, one of the correctors, as proposed by H. Epps, has an aspect ratio of 16:1 and a sagitta of approx. the mirror thickness. We are going to model this within the next 4 months.

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