Flexure mounts for high performance astronomical lenses

Robert Fata, Vladimir Kradinov, Daniel Fabricant Harvard-Smithsonian Center for Astrophysics, 60 Garden St., Cambridge, MA 02138

ABSTRACT

We have developed practical, high performance flexure mounts for large astronomical lenses in the Binospec spectrograph. Flexure mounts are an attractive alternative to the widely used elastomeric lens mounts when high axial stiffness is a priority and coupling fluids are incompatible with elastomers. We describe coupling fluid seals for the flexure mounts.

Keywords: Lens mounting, large optics, multi-object spectroscopy

1. INTRODUCTION

Binospec is a dual-beam, wide-field, multiple object spectrograph to be used at the Cassegrain f/5 focus of the converted 6.5 m Multiple Mirror Telescope^{1,2}. Each beam contains 19 lenses in seven groups, nine in the collimator (three groups) and ten in the camera (four groups). The groups are fluid coupled using Cargille Laser Liquid 5610, a siloxane-based fluid. The Binospec lenses have diameters between 8.4 and 14.6 inches, weights between 8.9 to 44.5 pounds, and are made from a variety of Ohara i-Line glasses, CaF2 and NaCl.

Early in Binospec's design phase, we developed mounts that used a continuous ring of RTV^2 . The advantages of mounting lenses with elastomers are well understood^{3,4,5,6}: ease of assembly, low cost, shock resistance, straightforward athermalization, and a convenient fluid seal. However, elastomeric mounts have two serious defects in our application: (1) the siloxane (silicone-based) LL5610 interacts with silicone-based RTV, causing a large increase in the couplant's refractive index and (2) it is difficult to achieve sufficient axial stiffness with the elastomeric mounts to reduce gravity and pressure-head induced displacements and tilts of the lenses to acceptable levels.

We tested three RTVs for elastomeric mounts: GE's red RTV560 and clear RTV655 and Dow Corning's clear Sylgard 184. The durometer hardness of the red RTV560 decreased after soaking in the LL5610 coupling fluid, but the two clear RTV specimens showed little or no change in their mechanical properties. However, samples of the LL5610 fluid in contact with both clear RTVs showed large changes in refractive index, which increased by 0.01. Our optical models show that we wish to control the refractive index to an accuracy of ~0.0001. The LL5610 lens couplant is a key element in the athermalization of the Binospec optical design and cannot be exchanged with a different material⁷.

An athermalized RTV mount using only the lens edges provided insufficient axial stiffness for the Binospec optics. Additional axial supports on the lens faces would have potentially allowed us to reach our goals for image motion, but would have been difficult to implement due to tight clearances, would have required good RTV bonds on coated lens surfaces, and would have significantly complicated the process of lens mounting.

Flexures can provide very high axial stiffness while decoupling radial loads due to differential thermal expansion of the lenses and lens bezels. In fact, choosing the appropriate axial stiffness of a flexured lens mount is a compromise between limiting both image motion and susceptibility to lens deformation from the overconstrained mount if more than three flexures are used. The primary disadvantages of flexured lens mounts are increased cost and the necessity of providing separate fluid seals if the lenses are fluid coupled.

Ground-based and Airborne Instrumentation for Astronomy, edited by Ian S. McLean, Masanori Iye, Proc. of SPIE Vol. 6269, 62695T, (2006) · 0277-786X/06/\$15 · doi: 10.1117/12.672634

2. FLEXURE AND LENS BEZEL DESIGN

2.1 Introduction

Each Binospec lens is mounted into its own bezel using tangent flexures that have a rectangular cross-section. The flexures are an integral part of the bezel and are formed using electrical discharge machining as shown in Fig. 1. This approach eliminates all hardware or entrapped air pockets inside the couplant volume. This integral design allows us to minimize the bezel diameter. The bezel (and flexure) material is heat treatable aluminum alloy 7050-T7451. This aluminum alloy is chosen for its high yield strength (57 ksi) and ultimate strength (75 ksi), good machinability, and availability in plates up to 6 inches thick. The athermalization of the optical design assumed that the inter-element spacings are metered by an aluminum structure⁷.



Fig. 1. Machined prototype bezel showing integral flexures.

A circular nub is pressed in the middle of each flexure. After the lens is aligned to the bezel, Hysol 9313 epoxy, filled 80% by weight with Siltex 44, will be injected to bond the nub to the lens. The nominal bond thickness is 0.010 inches. The Siltex filler reduces the thermal expansion of the 9313 epoxy to about 16.2 ppm/°F without reducing the epoxy's strength. The nub material used for each lens has a thermal expansion closely matching the lens material (to 1 ppm/°F or better) to minimize any thermally induced stresses at this bond interface. The nub materials are 304L stainless steel, Kovar, titanium, and 4150 steel.

We have prototyped and tested Teflon seals thermally formed from 0.010 inch thick sheet. The seals are sodium ammonia etched to allow good adhesive bonds, in our case using Hysol 9313 epoxy filled 80% by weight with Siltex 44. The bonds to the Teflon seals are very strong; in pull tests the Teflon fails before the bond. We have verified that all of the materials associated with the flexure mounts (seals, epoxy, nubs, etc.) are compatible with the LL5610: an entire lens with its mount can be submerged in LL5610 without affecting the fluid properties.

2.2 Design goals for the flexure mounts

The free parameters in the flexure design are the length, width and thickness of the flexure and the nub diameter. These parameters must be varied until we find a combination that simultaneously meets our goals for (1) stress levels in all of the components, (2) image motion and defocus due to lens deflections and tilts, and (3) deformation of the optical surfaces.

We have set the allowable stresses in each material as follows: 100 psi for the NaCl element and 500 psi for the other lens materials, 1,000 psi for Hysol 9313, 30,000 psi for 7050-T7451 aluminum and 16,000 psi for all of the nub materials. These stress values represent about half of the rated yield strength for the aluminum and a sixth of the ultimate tensile strength of the epoxy. Minimizing the radial forces (and lens deformation) resulting from the thermal expansion mismatch between the bezel and lens is a related constraint: we limit the radial load to one pound per flexure for a 75 °F temperature change.

To compensate for image motion due to lens displacements and tilts, as well as instrument flexure arising from the structure and mechanisms, Binospec's detector array will be mounted on a multi-axis piezoelectric actuator stage controlled by an active flexure control system. This stage has a travel range of 0.012 inches in each of three orthogonal Cartesian directions. We have set our total flexure limit (including the lens mounts, structure, and mechanisms) at 0.006 inches, half the available stage travel. A sensitivity table of the image motions at the detector due to tilts and decenters of the lenses, as well as the focus shifts due to axial displacements was generated with Zemax to guide the design. The amount of image motion caused by a unit lens tilt depends on the power and position of the lens. We initially set the stiffness of all of the mounts to 200 Hz, and were able to achieve this in most cases, but reduced the stiffness of most of the mounts to minimize stress. Our analysis procedures to determine the optical effect of lens deformations are discussed in section 2.6 below.

In operation, the lenses are subject to (1) gravity loads in any orientation, (2) the pressure head due to the lens couplant, (3) temperature changes of up to 75^{0} F, and (4) the assembly loads caused by bolting bezels together against slightly imperfect faces. The pressure head at the bottom of the largest lens is 0.55 psi and the corresponding load on the face of this lens is ~45 pounds, slightly more than the total weight of the lens. We require that the lens assemblies meet our stress limits when subjected to 3g shipping and handling loads.

2.3 Finite element models

We began the flexure design by generating spread sheets using classical beam equations but then created detailed finite element model of each lens/flexure assembly. We created four levels of finite element models: a model of the nub/epoxy/glass interface for each lens, a model of the lens/flexure assembly, a model of the collimator and camera assemblies, and a complete model of the entire instrument. The purpose of each model type is summarized in Table 1.

Finite Element Model	Used to Determine:	
Nub/epoxy/glass	Stresses in the nub, epoxy, and glass at the bond.	
Lens with flexures	Stresses in the lens and flexure, forces and moments at the nub, lens deformations. Deformation of the lens surfaces.	
Collimator and camera assemblies	Tilts and displacement of the lens in their mounts	
Entire instrument	Structural and mechanical deflections, motion of slit mask and grating.	

Table 1. Summary of Finite Element Models.

Fig. 2 shows the finite element model of the nub/epoxy/glass interface. This model was constructed with eight-node solid elements to represent the glass, the bond, and the nub. The epoxy bond, nub, and glass are modeled with four, six and 20 elements through their thickness, respectively. The mesh for the nub and glass was biased towards the epoxy layer to provide a reasonable aspect ratio for the elements. Two different sets of the boundary conditions were used to model mechanical and temperature loading. The results of these two load cases were combined to predict the total stress in the nub, epoxy and at the glass interface to determine the optimum nub diameter and flexure geometry.



Fig. 2. Finite element model of a nub/epoxy/glass bond.

Fig. 3 shows the finite element model of a typical lens and flexure assembly. The lenses and the nubs were modeled with solid eight-node brick elements. The glass elements are biased towards the nub location to produce a higher fidelity mesh in this critical region. The lens is modeled with at least six elements through its thickness, and the nub is meshed with four elements along each side. The flexure is modeled with four-node shell elements with four elements along its width and 52 elements along its length.



Fig. 3. Finite element model of a lens/flexure assembly.

To model the assembly loads arising from mounting flange surface irregularities, $\cos(2\theta)$ and $\cos(4\theta)$ axial displacement profiles were enforced at the flexure ends with a normalization of ± 0.001 inch. The $\cos(2\theta)$ case produced the largest surface errors, and the $\cos(4\theta)$ case produced the maximum stress levels in the flexures, lens and epoxy. Combined load cases considering gravity with the pressure head, a 75^oF temperature change, and assembly loads were used to evaluate optical performance. Similar load were used to calculate the maximum principal stress levels except that 3 g shipping and handling loads were used.

2.4 Iterative flexure design

Early in the design process we set the flexure thickness at 0.040 inches as a good compromise between high radial compliance to reduce stress from the lens/bezel thermal expansion mismatch while retaining sufficient thickness to avoid manufacturing difficulties. For the gravity and pressure head loads, longer flexures lead to higher stress in the flexures. For the bolt-up deformation loads, shorter flexures produced higher stress in the flexures, lens and epoxy. For reasonable tolerances on the flatness of the lens bezel faces, the bolt-up stresses dominate, so the length of the flexures was maximized based on the lens diameter and number of flexures.

The nub diameters are limited in many cases by the edge thickness of the lenses. Larger nubs reduce the stress levels in the lenses and epoxy for all of the load cases except temperature changes, and the former load cases dominate. The lens

weight, diameter and available lens edge width for bonding the nubs determine the quantity and size of the nubs required to spread the force resulting from the shipping and bolt-up assembly loads. The lens circumference limits the number of flexures given the length of the flexures and the diameter of the nubs. As the nub diameter increases, the flexure length must decrease for a fixed number of flexures.

These design tradeoffs are summarized in Table 2 and Fig. 4 is a flow chart of the flexure design process.

Table 2. Key flexure and nub design tradeoffs.

Load case	Longer flexure results in:	Larger nub results in:
Gravity and pressure head	Higher stresses in flexure	Lower stresses in lenses, epoxy bond, and nub, but forces shorter flexure if number of flexures is fixed
Bolt-up loads	Lower stresses in flexure, lenses and epoxy	Lower stresses in lenses, epoxy bond, and nub, but forces shorter flexure if flexure number is constant
Temperature change	Lower stresses in flexure, lenses and epoxy	Increases stress in lenses, epoxy bond, and nub

2.5 Flexure mount performance

The optimized flexures all have thicknesses of 0.040 inches; their lengths vary between 1.00 and 1.64 inches, and their widths vary between 0.14 inches and 0.35 inches. The nub diameters vary between 0.40 and 0.80 inches; one is elliptical with major axes 0.36 by 0.54 inches. To maximize the length of the flexures, we use twelve flexures for the largest diameter lenses, dropping to nine flexures for intermediate diameter lenses, and six flexures for the smallest diameter lenses. The mount frequencies range between 100 and 200 Hz. The maximum stresses are summarized in Table 3.

Table 3. Maximum principal stresses resulting from the worst-case combined 3g shipping loads, the $cos(4\theta)$ bolt-up displacement, and a 75°F temperature change. The flexures stresses were multiplied by a stress concentration of 1.4 based on the fillet radii at the ends of the flexure blades.

	Max Flexure Stress	Max Lens Stress	Max Epoxy Stress	Max Nub Stress
Modeled	29870	425	750	3500
Allowable	30000	500	1000	15000



Fig. 4. Flow chart of flexure design process.

2.6 Optical analysis of the deformed lens surfaces

Typically, it has been difficult to directly measure the degradation of the system image quality and the image motions caused by the deformed and displaced optical surfaces predicted by finite element models. To overcome this problem, we have developed a software tool to read the output of the finite element models and fit 37 Zernike polynomials to each deformed surface, and determine the decenter of the surface directly. We use the tilts, and displacements and the optical sensitivities determined with Zemax to predict the image motion and defocus at the detector. The Zernike polynomials can be fed back into Zemax to determine the effect of surface deformations. A sample output from the program is shown in Fig. 5.



Fig. 5. Analysis of finite elements results for the deformed surface of collimator lens 8. All units are inches.

2.7 NaCl Lens Mount

The single NaCl lens (lens 8 in the camera) could not practically be mounted with bonded flexures while meeting our stress limit of 100 psi. The thermally induced stresses at an epoxy bond interface would likely fracture the lens. We have designed an alternate mount that applies preload forces with flexures against three axial and two radial hard points. The radial hard points are athermalized with Teflon spacers. A single radial flexure acts as a spring to provide a 3g preload on the two radial hard points. A ring with three flexures is mounted to the top of the bezel to provide a 3g preload to the axial hard points. The maximum stresses are summarized in Table 4.

Table 4. Maximum principal stresses for the NaCl lens resulting from the 3g preload and 3g shipping loads. Since this lens is supported at three points and encapsulated in oil there are no additional loads from bolt-up or a pressure head. Temperature changes have minimal effect. The flexures stresses were multiplied by a stress concentration of 1.4 based on the fillet radii at the ends of the flexure blades.

	Max Flexure Stress	Max Lens Stress
Modeled	23100	90
Allowable	30000	100

3. LENS ASSEMBLIES

The collimator and camera lens assemblies are shown in Fig. 6 and Fig. 7. These assemblies are bolted together from the individual lens/bezel assemblies. Precision pilots are used to align the bezels to each other and o-ring seals retain the fluid lens couplant. In most cases, the seals will be located as close to the oil coupled surfaces as possible to minimize the required volume of fluid. Bladders to accommodate the thermal expansion of the coupling fluid are not shown. Two of the lenses (and their flexure mounts) in the collimator quintuplet are submerged in couplant because there is insufficient room to bond fluid seals onto these lenses. The NaCl lens in the camera triplet is also completely submerged in couplant. Each of the collimator and camera cell assemblies is mounted to an optical bench that supports the other critical optical assemblies, including the gratings and science cameras.



Fig. 6. Collimator assembly. The Teflon seals will be bonded around the perimeter of the lens to a seal interface ring to retain the couplant fluid. The seal interface ring is a separate aluminum piece bonded to the bezel.



Fig. 7. Camera Assembly. Lens 8 is made from NaCl and is mounted differently from the other lenses: flexures apply preloads against three axial and two lateral hard stops. Lens 10 is the dewar window (not shown).



Fig. 8. Finite element model of the complete Binospec instrument. The model includes 755,337 elements and 877,807 nodes. The mounting flange is pinned at 24 nodes.

4. PREDICTED IMAGE MOTION FOR THE COMPLETE BINOSPEC INSTRUMENT

As described in section 2.3 and Table 1, we have created finite element models of the collimator and camera lens assemblies, and of the complete instrument containing these assemblies. As described in section 2.6, we have calculated the image motion and defocus for various load cases. The finite element model of the complete instrument is shown in Fig. 8. Fig. 9 shows the image motion at the detector due to gravity acting on the collimator and camera lens assemblies (including the pressure head effects) at telescope elevation angles of 0, 30, 60, and 90°. Fig. 10 is the corresponding image motion for gravity acting upon the entire instrument.



Fig. 9. Image motion at the detector due to gravity acting on the lens assemblies. Each point represents a different rotator angle.

Fig. 10 shows the corresponding image motion for the complete instrument, including the lens assemblies, structure, mechanisms, and grating.



Fig. 10. Image motion at the detector due to gravity acting on the complete instrument.

Fig. 11 and Fig. 12 show the focus shifts due to gravity acting on the lens assemblies alone and the complete instrument, respectively.



Fig. 11. Focus change at the detector due to gravity acting on the lens assemblies.



Fig. 12. Focus change at the detector due to gravity acting on the complete instrument.

Binospec will be equipped with an active flexure control system similar in concept to that used in the DEIMOS instrument⁸. Following flexure correction, the images will be stabilized to better than 0.0002 inches.

ACKNOWLEDGMENTS

We thank Joe Zajac, Henry Bergner, Jack Barberis and Mario Pieri for their dedicated work throughout the design and development of the lens mounts. We thank Harland Epps for his optical design. We thank Bechdon Manufacturing and D&S Manufacturing for their fabrication advice and supplying us with the prototype flexured bezels.

REFERENCES

- 1. D. G. Fabricant, R. G. Fata and H. W. Epps, "Binospec: a dual-beam, wide-field optical spectrograph for the converted MMT," *SPIE Proc.*, 3355, pp. 232-241, 1998.
- 2. Daniel G. Fabricant, Harland W. Epps, Warren L. Brown, Robert G. Fata, Mark Mueller, "Development of Binospec and its optics," *SPIE Proc.*, 4841, pp. 1134-1144, 2003.
- 3. R. Fata and D. Fabricant, "Mounting large lenses in wide-field instruments for the converted MMT", *SPIE Proc.*, 3355, pp. 275-284, 1998.
- 4. Paul R. Yoder, Jr., "Optomechanical systems design," Marcel Dekker, 1993.
- 5. Terry Mast et al., "Elastomeric lens mounts," SPIE Proc., 3355, pp. 144-154, 1998.
- 6. B. C. Bigelow and A. M. Dressler, "IMACS: the multiobject spectrograph and imager for Magellan: a status report," *SPIE Proc.*, 4841, pp. 1727-1738, 2003.
- 7. H. W. Epps and D. G. Fabricant, "Athermalizing refractive optics with fluid lenses," *PASP*, 114, pp. 1252-1259, 2002.
- 8. Robert I. Kibrick et al., "The DEIMOS flexure compensation system: overview and operational results," *SPIE Proc.*, 5492, pp. 799-810, 2004.