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Design and performance of an optical mount using cross-flexure pivots

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ABSTRACT

The design of an optical mount using commercially available cross-flexure steel pivots is described. The pivots and their incorporation into a 5- × 10-inch rectangular optical mount are illustrated. Test data on this mount show acceptable thermal stability and thermal hysteresis for use in a laboratory optical system. The design is easily scaled to other optic sizes.

1. INTRODUCTION

This design was generated by the need for mounting a medium-sized rectangular optic. The resulting mount design is relatively simple, performs well under tests, and is easily adaptable to other small to medium-sized optics.

Table 1 shows the original design specifications for this optical mount. These specifications are based upon the need to maintain a particular static optical alignment in a laboratory environment, where temperature is controlled to ± 2°C.

The general design philosophy to meet these specifications was to try a gimbal design using flexures, even though a kinematic design seemed more appropriate to meet the thermal specifications. A kinematic design for a similar size optic, however, had been reported to be inappropriate because of the moderate forces required to load the actuators.[1]

Table 1. Optical mount design specifications.

Optic size	5 × 10 × 1 in.^{3}
Alignment resolution	<10 μrad
Adjustment range	± 2°
Thermal stability*	<3 μrad/°C
Thermal hysteresis**	<1 μrad/°C
Natural frequency	>100 Hz
Adjustment mechanism	Manual or motorized

*Optic misalignment due to slow drifts in ambient temperature.

**Change in optic alignment after returning to original temperature.

2. DESCRIPTION OF THE OPTICAL MOUNT

2.1. General description

Figure 1 shows the optic mounted in the inner of three rectangular closed aluminum frames. Figure 2 shows the four flexures (each held firmly by a pair of locking pins), the two manual adjustment screws, and the three spring-loaded clamps that oppose fixed clamps to retain the optic.

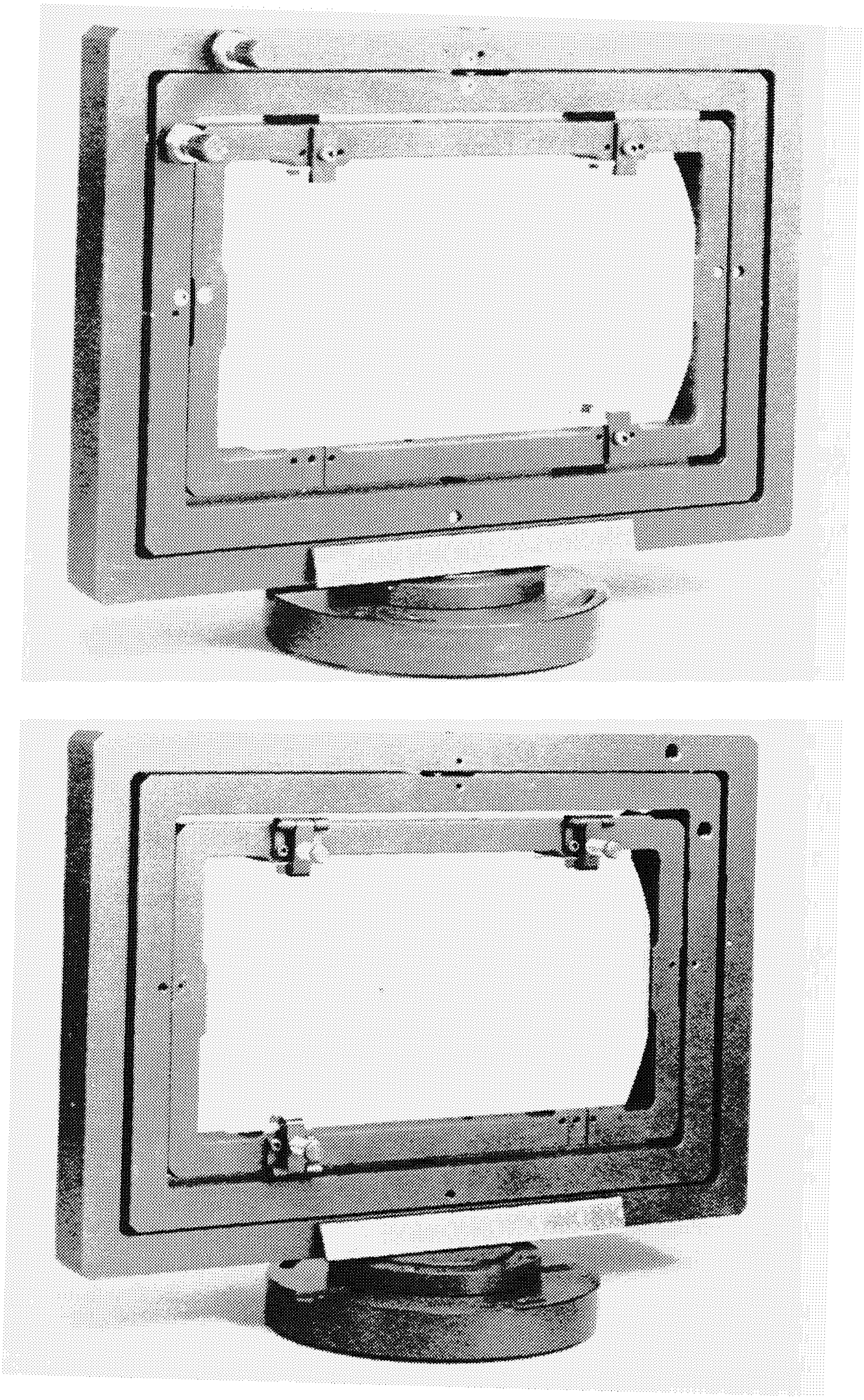


Fig. 1. Two views of the rectangular mount with the optic mounted in the inner of three closed frames.

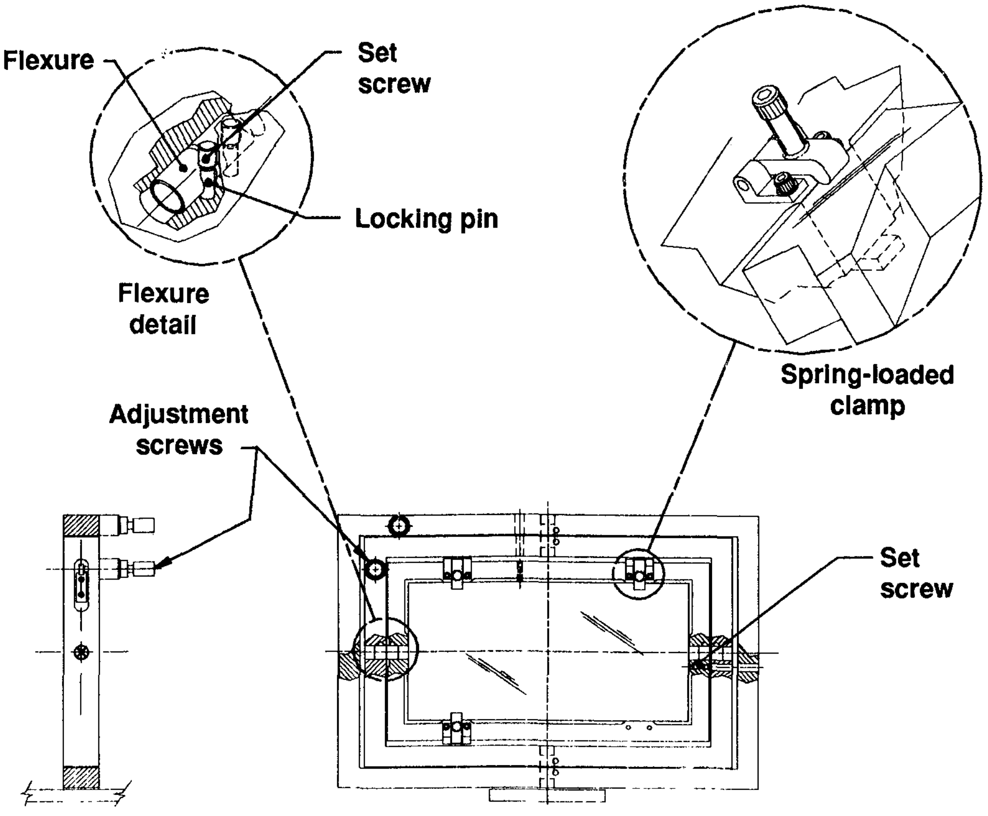


Fig. 2. Rectangular mount design features.

2.2. The flexures

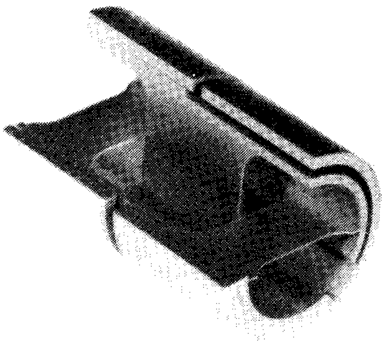
Figure 3 shows the flexural pivot used in this design. This cross-flexure, cylindrical pivot with a rotational capability of several degrees is ideally suited for high precision optical mount designs because it is vacuum compatible, frictionless, needs no lubricants, and has low hysteresis. These pivots, fabricated from 400-series stainless steel, are either brazed or welded into the assembly shown.

Cross-flexures were described 30 years ago by Billig.[2] The use of cylindrical cross-flexural pivots to replace ball bearing pivots in one optomechanical application was described by Yoder.[3] This same type of pivot in a mount design for a 6.75-inch square optic was described by Horn at this conference two years ago.[1] By coincidence, the engineer who proposed the use of the cross-flexure pivot in that mount worked on this design as well.[4] Such flexures, also found to be advantageous for cryogenic applications,[5] are now routinely discussed as viable alternatives to bearings in classes on optomechanical design.[6]

2.3. Pre-setting the flexures

The flexure is retained by a special pin that is forced against the circumference of the flexure by a set screw, as shown in Fig. 2. This feature eliminates the need for grinding a flat on the flexure for a set screw and does not damage the flexure. It allows one to easily pre-set the flexures by rotating the frame out-of-plane several degrees, locking the pin against each flexure, then rotating back to zero degrees. This preloads the frame against the actuators, precludes the use of springs or spring-loaded plungers as described previously,[1] and eliminates backlash from the actuators.

Fig. 3. Free-Flex® Cantilevered Pivot (Courtesy of Lucas Aerospace Power Transmission Corporation, Utica, NY).



The load P on the actuator can be calculated from that resulting from rotation of a pair of flexures through an angle A,

$$P = 2k\ A/L\ , \tag{1}$$

where k is the flexure torsional spring rate, L is the lever arm between the adjuster and rotational axis, and the factor of 2 results from the use of two flexures per axis. The torsional spring rate can be found by simply looking in the vendor's catalog; for the Model 5016-400 flexure[7] used here, k = 52 lb•in./rad = 0.91 lb•in./deg.

Designating L_x the lever arm for the actuator rotating the optic about its longer axis, and L_y that for the orthogonal (shorter) axis, the lever arms for this design are L_x = 3.0 in. and L_y = 5.3 in.

The loads on the actuators should be small in the event that motor-driven actuators are used (in order to minimize the motor size), but sufficient to maintain alignment during expected vibrations. The load levels chosen with this in mind were nominally 2 lb per actuator. In order to achieve this, the pre-set angles of rotation used prior to locking the flexures were A_x = 3.6 deg and A_y = 6.0 deg, resulting in calculated loads on the actuators from Eq.(1) of P_x = 2.2 lb and P_y = 2.0 lb.

2.4. The actuators

The actuators are commercially available 80 thread/inch adjustment screws (Newport Corporation, Model AJS-05) in a brass housing, as illustrated in Fig. 2. This pitch provides a translation of 35 μin./deg of rotation. Lever arms of 3.0 and 5.3 inches provide mount resolution capabilities of 12 and 7 μrad, respectively, for a one degree rotation of the screw, which is an approximate minimum adjustment.

2.5. Retaining the optic

The optic is held in the frame by three fixed metal "semi-kinematic"[5] pads with directly opposing spring-loaded metal clamps, as shown in Fig. 2. Each clamp is designed to provide a line contact on the optic with safe stresses; the nominal load per clamp is 8 lb. The clamps are located inboard from each end of the mirror to reduce mirror sag; they are located at the 18% points from the mirror ends. The optic thickness of one inch is also required to insure mirror sag is less than a small fraction of a wavelength.

The degrees of freedom in the orthogonal directions in the plane of these pads, as well as rotation about an axis perpendicular to the mirror face, are restricted by nylon-tipped set screws gently tightened against the ground edges of the optic at installation, as shown in Fig. 2.

2.6. Natural frequency of the mount

The natural frequency of the mount[8] can be expressed as,

$$f_n = (22/7)\sqrt{1/\delta}\,, \tag{2}$$

where δ is the static deflection of the optical mount (inches) due to its own weight. Calculations on this design gave a value of $\delta = 3.4 \times 10^{-4}$ in., which yields $f_n = 1.7 \times 10^2$ Hz; stiff, closed rectangular frames are used to achieve a high natural frequency.[4]

3. TEST ARRANGEMENT

This mount was tested for its thermal stability and thermal hysteresis. The objective was to measure the degree of angular misalignment of the optic when ambient temperature changed by 10°C and to measure its thermal hysterisis, i.e., the extent to which the mount failed to return to its original alignment upon return to its original temperature.

Figure 4 shows the test arrangement. The optical mount was attached to a vertical *optical* wall by a sturdy aluminum bracket. It was then enclosed within an aluminum housing containing a 200-W disk heater and a 60-cfm circulation fan. A small hole in the enclosure allowed an electronic autocollimator beam to probe the mirror. A 10°C step rise to the ambient air was applied (monitored by a nearby thermistor), until the mirror mount alignment stabilized. The heater was then turned off and the enclosure opened to allow the mount to return to room temperature. A dual-channel chart recorder plotted the mirror tilt in one axis and the ambient temperature versus time.

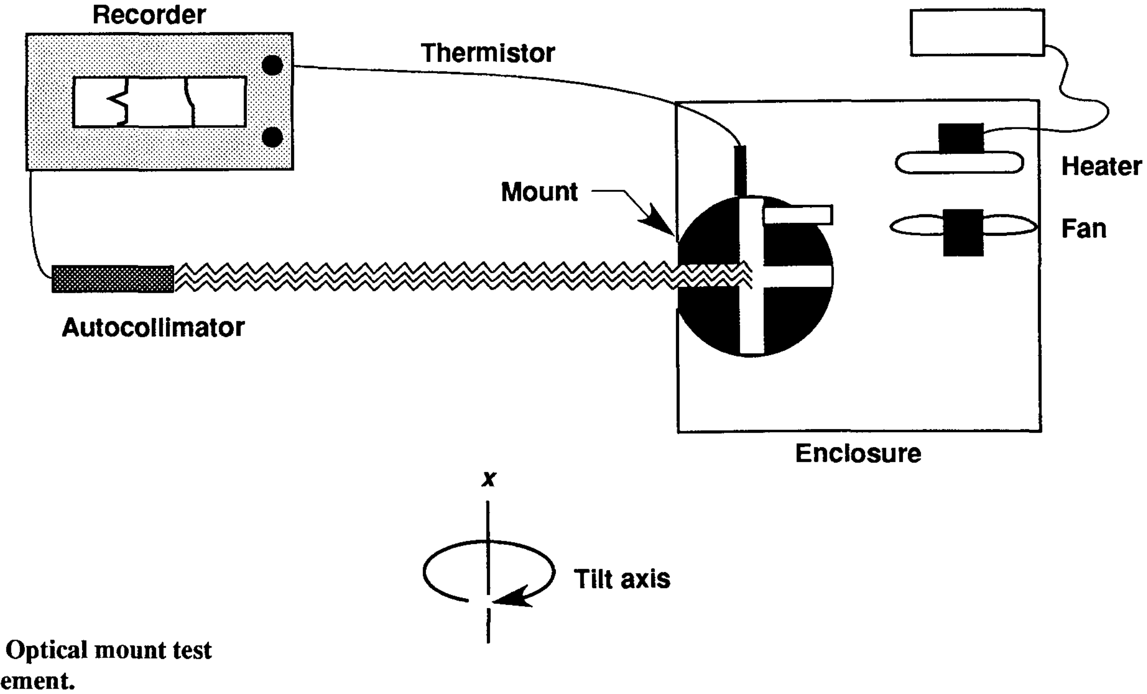


Fig. 4. Optical mount test arrangement.

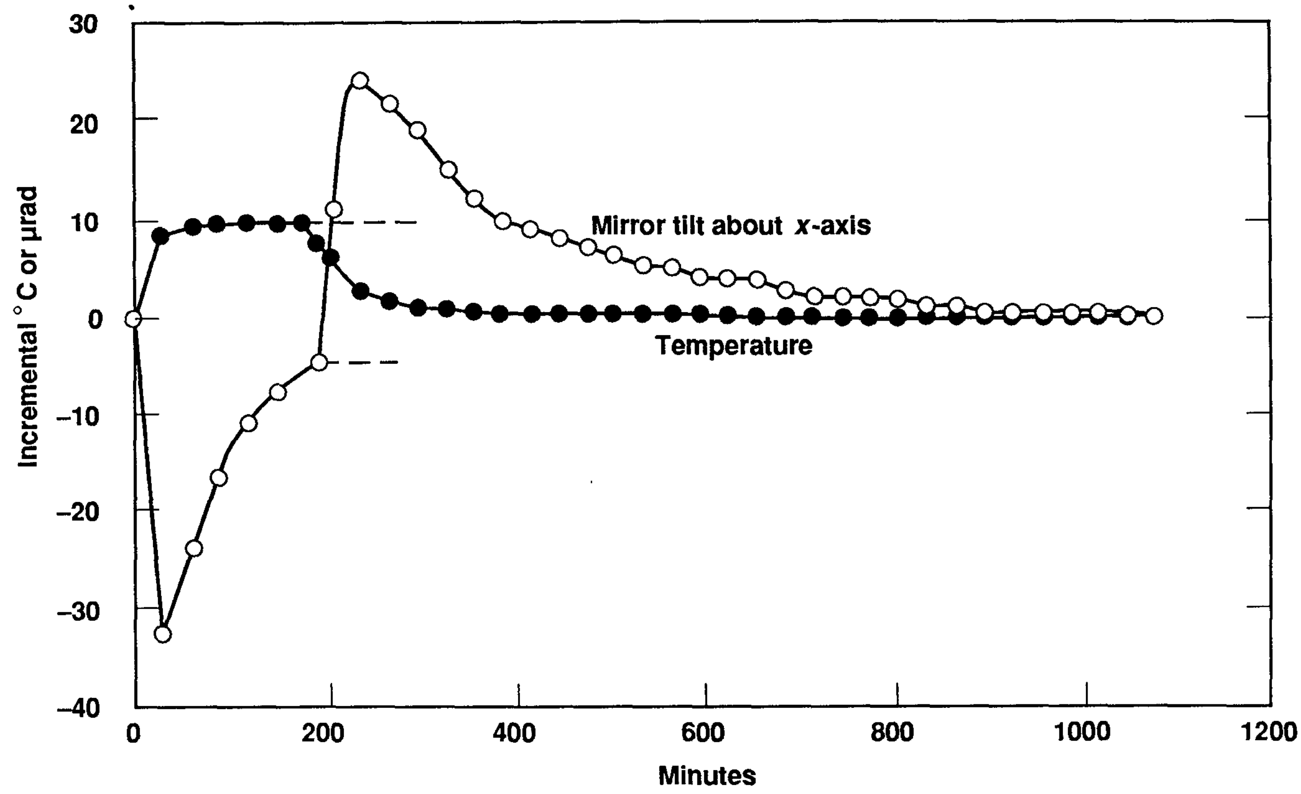


Fig. 5. Rectangular mount thermal test data.

4. TEST RESULTS

Figure 5 shows the test results on this mount After applying the 10°C step in ambient temperature, the optic alignment moved quickly to a –32 µrad misalignment, then stabilized in about 3 hours at a –5 µrad tilt. The heater was then turned off and the enclosure removed, causing the mirror to tilt in the opposite direction. It then slowly returned to its original alignment. The thermal hysteresis was negligible—less than 1 µrad, which is about the experimental error of this test.

These test data were obtained without any locking features on the actuators. Additional tests used two locking schemes: In the first, the set screw provided with the commercial actuator was used to lock the 80-pitch screw; in the second, an opposing 80-pitch screw was used. In each case, the alignment was disturbed by the locking operation (~10 µrad), and the thermal hysteresis was higher.

5. CONCLUSIONS

The test results on this mount show a significant improvement in thermal stability and thermal hysterisis as compared to values seen in similar tests of either custom or commercially available optical mounts, whether gimbal or kinematic.[9] For increases in ambient temperature that occur over time periods on the order of hours, Fig. 5 implies that the mirror tilt would change from its nominal position to –5 µrad for a 10°C change. We would designate that as a long-term thermal stability of 0.5 µrad/°C. These results are unexpectedly good for an *aluminum* mount.

Most thermal motion is due to the dissimilar materials of the mount and the adjustment mechanisms, a situation that could be improved by use of different materials in the adjuster and/or frames, but at additional cost. Figure 5 implies that the aluminum/brass part of the adjuster expands quickly to tilt the optic to –32 µrad. The slower temperature rise of the steel adjusting screw causes the mount to tilt in the opposite direction, resulting in a net offset of only –5 µrad at equilibrium.. The opposite effects occur during the cooling of the mount. The low hysteresis is partially the result of the good flexure properties.

Although mirror tilt about the *y*-axis was not tested, its longer lever arm would provide even smaller angular tilts for comparable linear displacements.

This mount design is adaptable to optical elements of almost any size or geometry, limited only by the characteristics of the flexure pivots.

6. ACKNOWLEDGMENTS

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