Proc. of SPIE Vol. 0959, Optomechanical and Electro-Optical Design of Industrial Systems, ed. R J Bieringer, K G Harding (Jan 1988) Copyright SPIE

Flexure mounts for high-resolution optical elements

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Introduction

Flexures are passive mechanical-structural devices used to isolate optical elements from the mechanical and thermal effects of the structural support system in such a way that these effects on optical instrument quality are minimized. Mechanical effects include gravity, and inertial and vibratory loadings, as well as possible stresses resulting from assembly errors. Thermal effects include both steady-state and transient environments. For example, if an optical device, with a mirror or lens having a coefficient of thermal expansion orders of magnitude less than that of the material of the support structure (e.g., steel or aluminum), is assembled at a given temperature but operates in a thermally different environment, the optical quality of the instrument may degrade very significantly unless the mirror or lens is isolated from the thermal strain of the support structure. Normally, the mechanical precision of the mount is much less than the precision of the optical surface of the element. If the optical element is rigidly clamped to the mount, this lack of precision in the mount may distort the optical element, and degrade the optical quality of the element.

The following criteria must be met for successful mounting of optical elements:

1. The mount must exert low force on the optical element to minimize optical surface distortion.
2. The mount should have high stiffness to maintain the alignment of the optical elements.
3. The mount must be athermal; changes in temperature must not degrade the optical surface figure or change the position of the optical element.
4. Material stability and creep effects should be considered so that the position of the optical element remains stable with the passage of time.
5. The mount size and weight should be as small as practical.
6. Mounting fabrication and material cost should be as low as possible.

Kinematic mounts are widely used in optics to minimize the deformation of optical elements caused by mount-induced stresses. From kinematic theory, a rigid body in space has six degrees of freedom, three translations and three rotations. Normally, three orthogonal axes are used in kinematic theory, so there is one translation and one rotation per axis. Kinematic theory assumes that perfectly rigid (infinite elastic modulus) bodies contact only at infinitesimal points.

From these principles, it is apparent that any rigid body has 6 - n degrees of freedom, where n is the number of points in contact with the body. Any body with more than 6 contact points is overconstrained, and may be distorted by the additional contact points (assuming finite elasticity), and its position altered. An extremely important assumption used in kinematic theory is the transmission of the locating forces at the contact points to the rigid body. Since the contact points are assumed to be infinitely small, only forces locally normal to the surface of the rigid body are transmitted. It is assumed that shears and moments are not transmitted by point contacts.

In a practical kinematic mount, the finite elasticity of materials changes the point contact to a small-area contact. Hertz contact stress theory is used to determine the magnitude of the stresses in the point contacts. These point-contact stresses are usually very high, limiting the use of kinematic design to relatively small and lightweight components. Figure 1 is a representative example of the contact stress problem in kinematic design. In this figure, the contact stress for a classic two-point, radial-edge mount for a solid six-to-one (diameter to thickness) ratio glass mirror is given for mirrors with diameters of 1 in. to 12 in. Mirror diameters above approximately 5.5 in. have mount-induced contact stresses exceeding 50 ksi. This 50-ksi stress is considered the maximum safe compressive stress for many optical glasses.

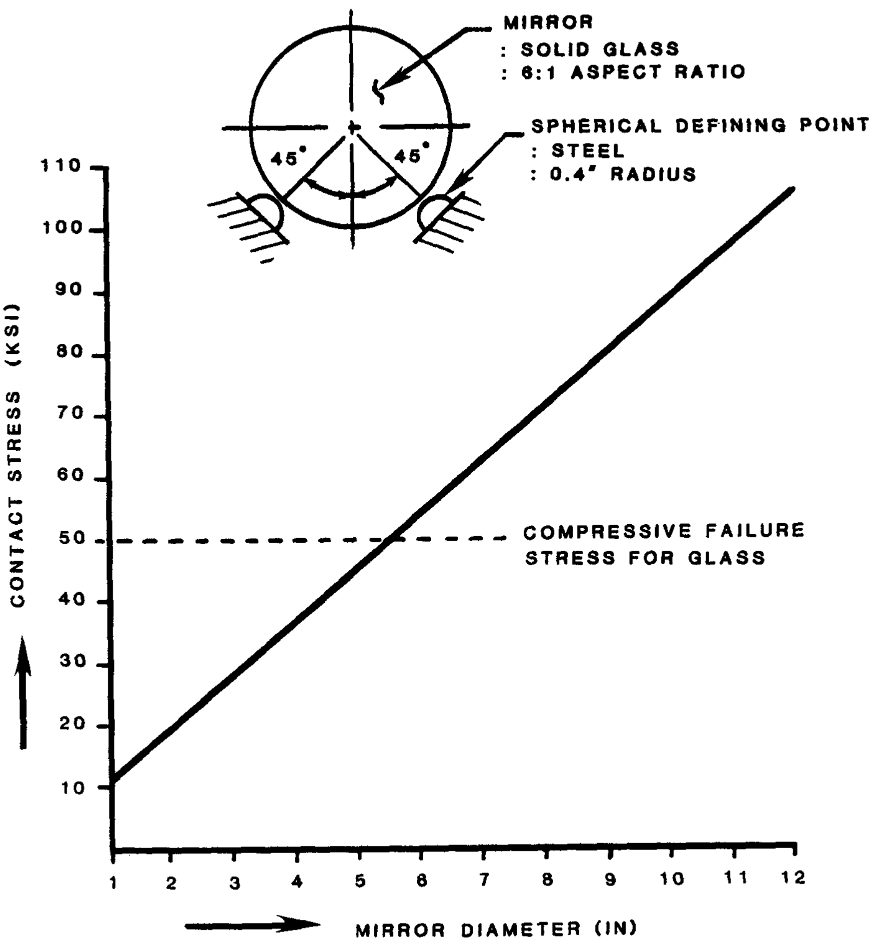

Figure 1. Contact stress vs. mirror diameter for radial defining points in a classical kinematic mirror mount.

Reduction in the mounting stress caused by point contacts is possible using the principles of semi-kinematic design. Semi-kinematic design uses kinematic principles to locate support points, but replaces each point contact with a small contact area or pad to reduce contact stress to acceptable levels.

A major concern in semi-kinematic design is accuracy of the mounting pads. If the mounting pads are not flat, or if the pads are not co-planar, the optical element may be distorted. For example, in diamond turning of optics, where semi-kinematic principles are often used, the rule of thumb for mount accuracy is that it must be as good as the optical surface accuracy. This is an expensive concept, which requires a mounting pad accuracy of a few microinches, a requirement that is not practical in many cases.

A more realistic and less expensive tolerance for mount pad accuracy is to require pad co-planarity and flatness to be the same magnitude as the pad elastic deformation under the clamping stress. Although this tolerance is an improvement over the diamond turning rule of thumb, high pad accuracy, frequently to better than 0.0001 in., still is required. Calculation of pad elastic deformation requires assumptions about the surface condition of the pad which may not be accurate.[1]

Pad co-planarity requirements in semi-kinematic mounts are further relaxed by introducing rotational compliance into each mounting pad. If local rotation can occur between pad and mount during assembly and clamping, some "self-alignment" occurs, and any mount-induced stress in the optical element is reduced. For example, in one case involving a 40-in.-diameter, high-energy laser mirror mounted on a five-point semi-kinematic mount, use of simple spherical tooling washers inserted between pads and mount, reduced the mount-induced figure distortion from 225 μin. to less than 25 μin. Point and line contacts are often designed into mechanisms used to introduce rotational compliance, which limits the load capacity and stiffness of semi-kinematic mounts employing such mechanisms. Friction and hysteresis exist in any rotational coupling, and are additional limiting factors in the performance of semi-kinematic mounts.

An additional design concern in semi-kinematic mounting is athermalization of the mount. Athermalization requires radial compliance in the mounting; this is sometimes introduced as an additional degree of freedom in each semi-kinematic mounting pad. Shown in Figure 2 is an example of a sophisticated athermalized semi-kinematic mounting patented by Bernard Mesco.[2] In this design, three spherical sectors attached to the optical element run in three radial holes in the mount. Radial motion of the optical element relative to the mount is possible, and an error in the location of the radial holes, or point of attachment of the spherical sectors does not introduce a strain in the optical element. The accuracy of the mount is set by the clearance between spherical sector and radial hole; the load capacity by the size of the contact between spherical sector and radial hole; and the mount-induced strain by the friction between spherical sector and radial hole.

Flexure Mounts

Flexure mounting of optical elements avoids many of the problems inherent in the use of kinematic and semi-kinematic mounts. A flexure is defined as an elastic element providing controlled motion. A spring differs from a flexure in that a spring provides controlled force through elastic deformation. Flexures are free of stick-slip and friction effects which are problems for semi-kinematic mountings. Hysteresis is lower in flexures than in mechanisms employing rolling or sliding contacts. Adverse environments, such as high or low temperatures, vacuum, nuclear radiation, and abrasive dust, are less likely to affect the operation of properly designed flexure mounts. Normally, a flexure mount will require little if any maintenance. Flexure performance is ideal for space applications, and many space optics are mounted using flexure designs.

Flexure mounting may be regarded as an extension of semi-kinematic design, with the important exception of providing all motions by the use of flexures. As in semi-kinematic mounting, the flexure location is determined by the use of kinematic principles. To minimize optical surface

ABSTRACT

An optical element (10) is mounted and aligned with minimum strain and distortion within a housing (12) to which at least three mounts (14) are attached and spaced about 120° apart on the periphery of the optical element. Each mount consists of a spherical disc (16) housed within an annular ring or sleeve (18), and each disc abuts and supports the optical element. Any loading exerted on the housing causes the discs to slide and/or rotate with respect to the annular disc-retaining sleeves to prevent such loadings from being transmitted to the optical element, thereby enabling the optical element to maintain a chosen, strain-free orientation

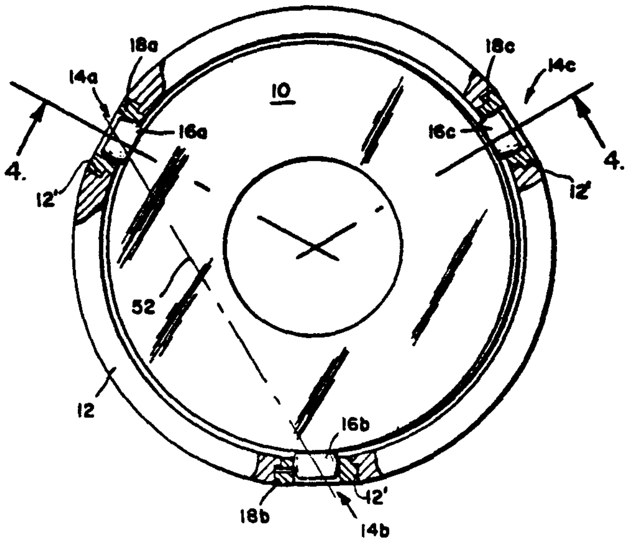

Figure 2. Semi-kinematic mirror mount (from U. S. Patent description).

deformations, and to maintain optical alignment of the mounted optical element, the flexure support forces exerted on the element must act through the center of gravity of the optical element. Athermalization of the mounting is achieved by providing flexural compliance acting in the radial direction with respect to an axi-symmetric optical element. Additional degrees of freedom or compliance may be required to allow for fabrication tolerances.

Illustrated in Figure 3 are flexure forces acting through the optical-element center of gravity. In the simple case of a circular mirror mounting, with the flexure forces acting only in the plane of the optical surface of the mirror, the flexure effects must provide high stiffness to prevent de-centration of the optical element, while providing radial compliance to allow for athermalization of the mount. This principle is shown in Figure 4. If the mounting flexures are equally spaced about the perimeter of an axi-symmetric optical element, and the flexures are identical in compliance, the spring constant or stiffness of the flexure-mounted optical element is given by

$$K_x = K_y = 3/2\ (K_R + K_T)\ , \qquad (1)$$

where K_x is the stiffness in the x direction; K_y is the stiffness in the y direction; K_R is the stiffness of each flexure in the radial direction; and K_T is the stiffness of each flexure in the tangential direction.

Using Eq. (1), the fundamental frequencies of the flexure-mounted optical element are estimated by

$$f_{nx} = f_{ny} = \frac{1}{2\pi}\left[\frac{3}{2}\,\frac{(K_R + K_T)}{m}\right]^{1/2}\ , \qquad (2)$$

$$f_{nzz} = \frac{1}{2\pi}\left\{\frac{3r^2 K_T}{I_{zz}}\right\}^{1/2}\ , \qquad (3)$$

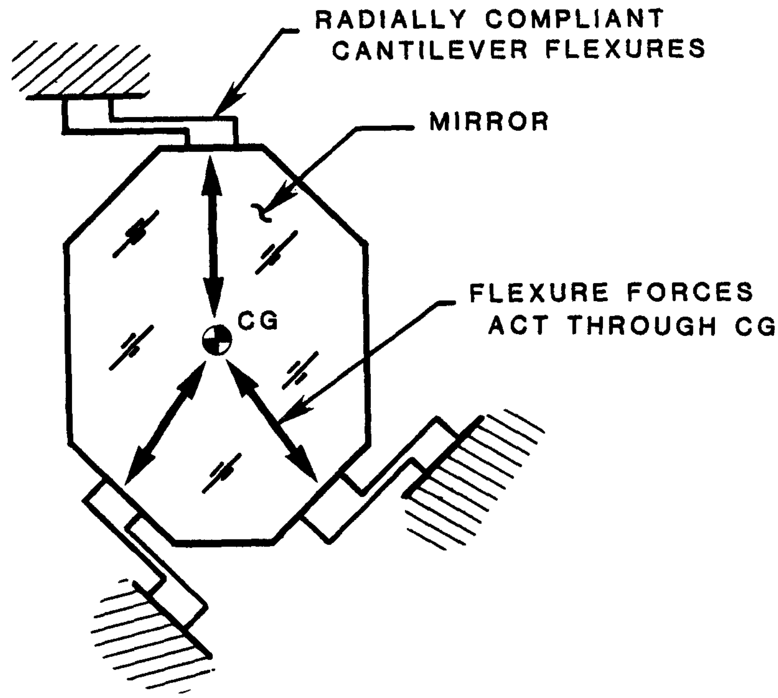

Figure 3. Center-of-gravity flexural mounting.

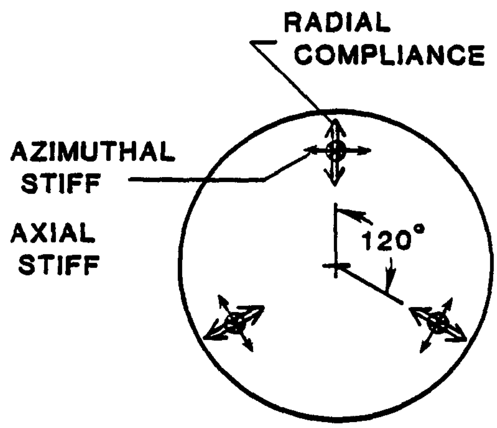

Figure 4. Flexure principle; the vector sum of flexural stiffness uniquely defines mirror position, while radial compliance reduces stress caused by difference in thermal coefficient of expansion between mirror and mount as temperature changes.

where f_{nx} is the fundamental frequency in the x direction; f_{ny} is the fundamental frequency in the y direction; f_{nzz} is the rotational fundamental frequency about the z axis; m is the mass of the optical element; I_{zz} is the moment of inertia of the optical element about the z axis; and r is the optical element radius.

Assembly-error-induced moments in the optical element are reduced by adding additional compliance to each flexure mounting point. Illustrated in Figure 5 is the use of an additional flexure (a cross-strip rotational flexure) to provide a rotational compliance that reduces the effect of co-planarity error on a mounting flexure. The complexity of many flexure mounting systems generally arises from considerations of assembly error. The alternative to introducing additional degrees of freedom, or compliance, is higher precision in fabrication. In many cases, such precision is simply not practical. In cryogenic applications, distortion of the mirror-mount structure in a unpredictable fashion is virtually guaranteed, which requires the use of additional compliance in the flexure mounts, regardless of the fabrication precision.

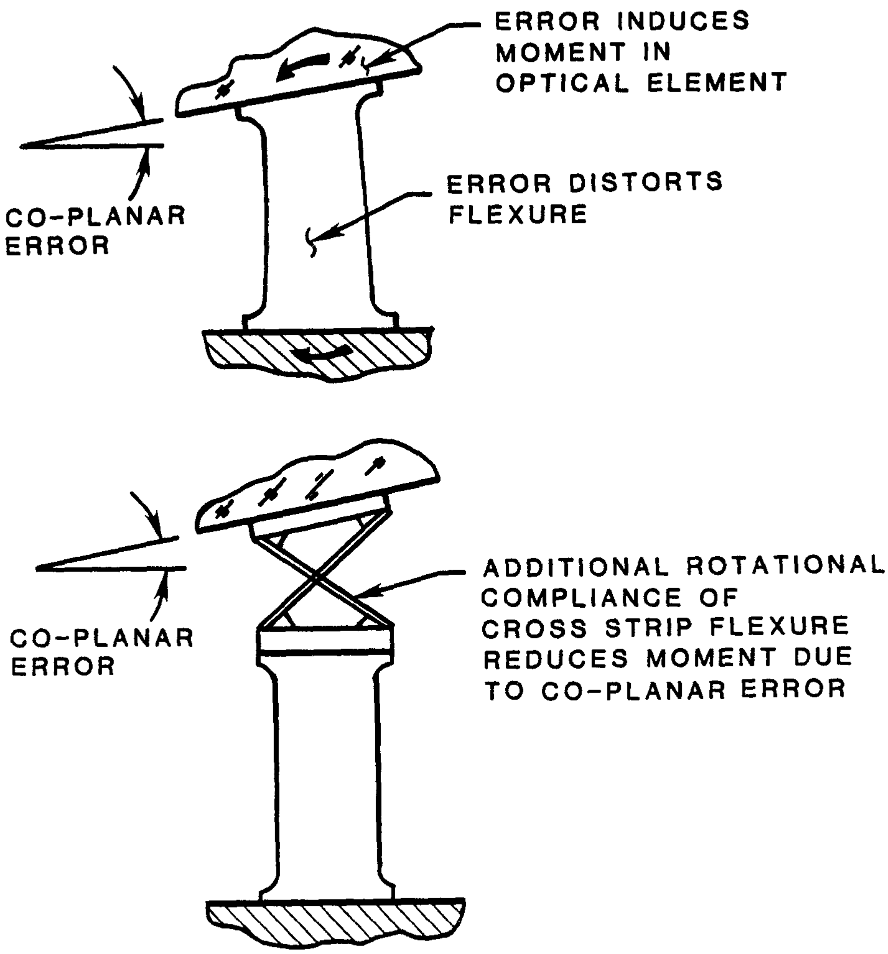

Figure 5. Flexure assembly error; co-planar errors in flexure or flexure mount may require additional flexures to reduce mounting forces on the optical element.

Flexure Materials

Design of a flexure mount for an optical element requires first a selection of mounting flexure configuration, then a detailed design of the individual flexures, followed by choice of a flexure material. This process is inverted in this paper, since choice of flexure material heavily influences the design of the individual flexure, and hence plays an important role in overall mount configuration.

Compliance is defined as the inverse of spring rate or spring stiffness

$$C = \frac{1}{K} , \quad (4)$$

where C is the compliance and K is the spring rate or stiffness.

Compliance of an individual flexure depends of the flexure shape and flexure material. For a constant flexure length, the greatest compliance is obtained with the use of a material with the greatest reduced tensile modulus. The reduced tensile modulus is a material parameter defined as the ratio of the allowable yield strength of the material to the elastic modulus of the material. High values of the reduced tensile modulus imply high yield strengths and low elastic moduli. For a given type of material, such as steel, the largest flexure compliance for a given length is obtained through the use of high-strength steels.

Stability of the flexure material with respect to time is a very important design consideration for maintaining alignment of the optical element. For an individual flexure, stability is dependent on the residual stresses in the flexure and the dimensional stability of the material. Residual stress in the flexure is controlled by fabrication procedures and rigorous post-fabrication stress relief. Dimensional stability of the basic flexure material is controlled by choice of flexure material, and by processing the material for stability. As a very rough rule of thumb, the higher the transition temperature or melting temperature of a material, the greater the stability of the material.

A commonly employed parameter characterizing dimensional stability of materials is the microyield strength (MYS) or precision elastic limit (PEL) of a material. The microyield strength of a material is defined as that stress which is required to produce a permanent plastic strain of one part per million. Although frequently used as a safe working stress to avoid dimensional instability or room-temperature creep, material instability or room-temperature creep under applied load is possible at stresses below the microyield stress. Permanent strain is estimated for some materials through the use of a power law relating stress and strain. Unfortunately, parameters for use in this power law are usually not readily available for most flexure materials.

Instability with time, or room-temperature creep, may be predicted by Andrade's Beta law,[3] which states that creep is proportional to the cube root of time or

$$\epsilon = \beta t^{m} , \quad (5)$$

where ϵ is the creep strain at time t; β is a constant dependent on material, stress, and temperature; t is the time in hours; and m is a constant, usually about 0.33.

The Andrade Beta relationship for creep has been found to be valid for a very wide range of materials: steel, cast iron, stainless steel, aluminum, titanium, magnesium, copper, beryllium, and molybdenum. Unfortunately, the published values for the Beta constant are limited and difficult to obtain. For titanium, which is often used as a flexure material, Andrade's relationship is[3]

$$\epsilon = \left[\frac{\sigma}{93.4}\right]^{14.88} t^{0.28} \; ; \quad \text{For } T_i \text{ 6Al-4V, } t \leq 1000 \text{ hrs} \quad (6)$$

where ϵ is the creep strain (1×10^{-6}); σ is the stress (ksi); and t is the time (hrs).

Fracture toughness is an important material parameter characterizing the resistance to fracture or cracking of a material. Thin flexure sections are prone to cracking under applied mounting stresses, and a fracture mechanics analysis should always be performed on critical mounting components. Unfortunately, most high-strength, and thus high reduced-tensile-modulus materials have poor fracture toughness. Careful selection of materials can partially compensate for the poor fracture toughness of these high-strength materials. Titanium has relatively poor fracture toughness, particularly at low temperatures. Use of the extra-low interstitial (ELI) alloys, such as 66Al-4V ELI titanium, improves fracture toughness. Similarly, the 7075 aluminum alloy has higher fracture toughness in comparison with a conventional 6061 or 2024 aluminum alloy.

Another material consideration is the thermal properties of the flexure materials; in particular, the thermal coefficient of expansion and thermal conductivity. For most optical applications, a good match of the thermal conductivities between flexure material and optical substrate or mirror material is important. For example, the thermal coefficient of expansion of titanium is a good match with beryllium, and has been used as a flexure material for mounting beryllium optics such as the IRAS (Infrared Astronomical Satellite) primary mirror. Thermal conductivity is usually not as important, except for systems exposed to large or rapid temperature changes.

Flexure Designs

Mounting flexures are assembled from combinations of simple or primary flexure geometries. For most applications, the primary flexure form is that of the single-strip flexure, which is in turn assembled into the forms of parallel spring guides, cross-strip rotational hinges, and cruciform flexures. These more complex forms are, in turn, assembled to form complex flexure mounting systems.

Starting with the primary flexure, the single-strip flexure, several different support and loading conditions can be identified wherein a single-strip flexure can be used to guide rotational or translational motion. Flexure performance may be influenced by the presence of axial loads. Different equations for compliance are required as the direction of the axial loads changes. For the single-strip flexure shown in Figure 6 used to guide rotation, and assuming no axial loading, the relationship between moment and rotation of the flexure end or flexure end slope is given by:[4]

$$M = \frac{EI\theta}{L} . \qquad (7)$$

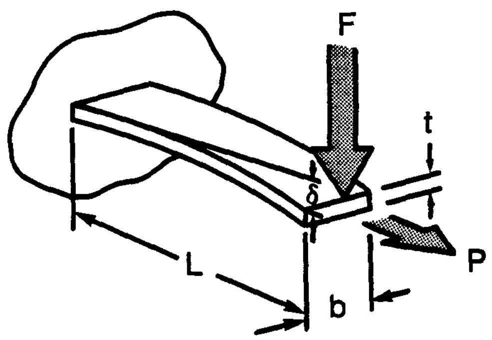

Figure 6. Single-strip cantilever.

If the single-strip flexure is used to guide rotation, and has a compressive actual load, the rotation of the flexure end or flexure end slope is given by

$$M = \frac{EI\lambda\theta}{\tan(L\lambda)} ; \tag{8}$$

$$\lambda = \left[\frac{P}{EI}\right]^{1/2} . \tag{9}$$

If the single-strip flexure is used to guide rotation, and has a tensile axial load, the rotation of the flexure end or flexure end slope is given by

$$M = \frac{EI\lambda\theta}{\tanh(L\lambda)} . \tag{10}$$

In the above equations θ is the end slope of the flexure, in radians; L is the flexure length; M is the moment required to rotate the flexure end; E is the elastic modulus of the flexure material; I is the moment of inertia of the flexure cross section; and P is the applied axial load.

By constraining the flexure against rotation, the single-strip flexure is used to guide translation. The force required to translate this flexure, providing that no axial load exists in the flexure, is given by[4]

$$F = \frac{12EI\delta}{L^3} . \tag{11}$$

If the flexure is used to guide translation, and a compressive axial load exists in the flexure, the force required to translate the flexure is given by

$$F = \frac{P\lambda\delta \cot\left[\frac{L\lambda}{2}\right]}{2 - L\lambda\cot\left[\frac{L\lambda}{2}\right]} . \tag{12}$$

If the flexure is used to guide translation, and a tensile axial load exists in the flexure, the force required to translate the flexure is given by

$$F = \frac{P\lambda\delta\coth\left[\frac{L\lambda}{2}\right]}{L\lambda\coth\left[\frac{L\lambda}{2}\right] - 2} . \tag{13}$$

In the above equations F is the force required to translate the flexure; δ is the translation of the flexure; L is the flexure length; E is the flexure material elastic modulus; I is the cross-sectional moment of inertia of the flexure; and P is the axial load in the flexure.

Two simple single-strip flexures may be combined to make the cross-strip rotational hinge. In this flexure, two single-strip flexures placed are right angles to provide a guide for rotational motion. Figure 7 shows a cross-strip rotational hinge. A commercial version of this flexure is the Bendix flex pivot. Rotation of a cross-strip flexure is not true rotation; a shift of the instantaneous center occurs as the flexure rotates. This shift of instantaneous center limits the precision of rotation of the flexure and is given by[5]

$$\Delta r = \frac{\sqrt{2}}{12} L\theta^2 , \tag{14}$$

where Δr is the shift in instant center; L is the length of the individual flexure blades; and θ is the rotation angle in radians.

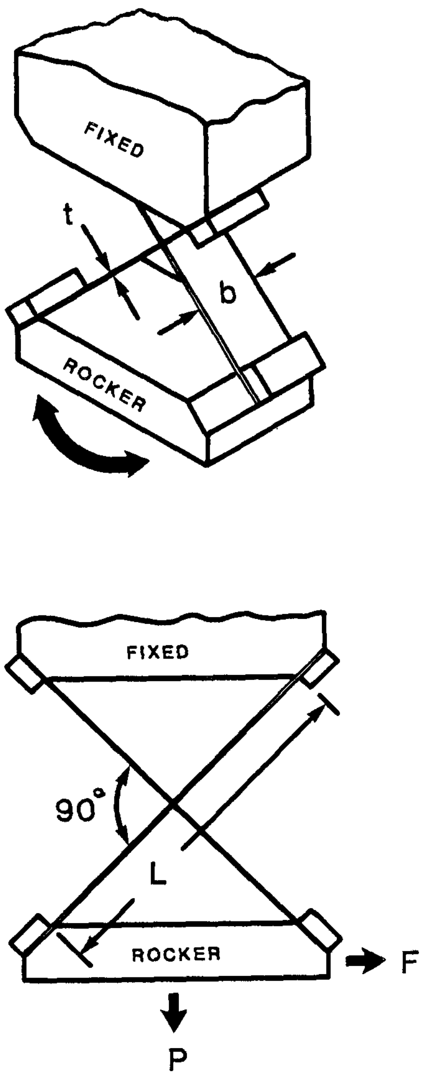

Figure 7. Cross-strip rotational flexure.

For small angles of rotation, less than about 0.1 radian, and where there are no axial loads carried in the flexure blades, the rotation angle of a cross-strip flexure that results when a moment is applied to the flexure is given by[5]

$$M = \frac{2EI}{L}\,\theta\ . \tag{15}$$

If a tensile load is present in the flexure blades, the rotation angle for a cross-strip flexure that results when a moment is applied is given by

$$M = \frac{EI\lambda\theta}{2}\left[\coth\left[\frac{L\lambda}{2}\right] - \frac{L\lambda}{2}\right]\ . \tag{16}$$

If a compressive load is present in the flexure blades, the rotation angle for a cross-strip flexure that results when a moment is applied is given by

$$M = \frac{EI\lambda\theta}{2}\left[\frac{L\lambda}{2} + \cot\left[\frac{L\lambda}{2}\right]\right]\ . \tag{17}$$

In the above equations θ is the rotation angle; M is the applied moment; L is the length of the individual flexure blades; E is the elastic modulus of the flexure material; I is the cross-sectional moment of inertia of one of the two flexure blades; and P is the applied axial load.

A pair of single-strip flexures mounted parallel to each other are used to form a parallel spring guide and provide linear translation. This type of flexure assembly is useful when a small range of linear translation, about 1 to 2 mm, is required. The motion so provided is not truly linear; as the flexure moves, the flexure is displaced in an axis perpendicular to the motion axis and parallel to the flexure blade length. Shown in Figure 8 is the geometry of a parallel spring guide. Error in linear translation motion for a parallel spring guide is given by[6]

$$\Delta L \cong \frac{\delta^2}{2L}\ , \tag{18}$$

where ΔL is the error in motion, or motion along an axis perpendicular to the axis of motion; δ is the translation motion; and ω is the flexure length.

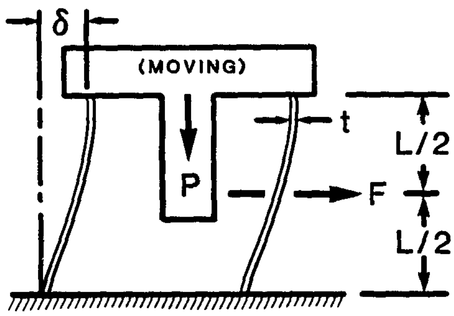

Figure 8. Parallel spring guide.

If the parallel spring guide is not subject to any axial forces in the flexure blades, the force required to translate the flexure is given by[7]

$$\delta = \frac{FL^3}{2Ebh^3} \ . \tag{19}$$

If the parallel spring guide is subject to an compressive axial load, the force required to translate the flexure is given by

$$\delta = \frac{FL}{P} \frac{1}{\psi L/2} [\tan (\psi L/2) - 1] \ . \tag{20}$$

If the parallel spring guide is subject to a tensile axial load, the force required to translate the flexure is given by

$$\delta = \frac{FL}{P} \left[1 - \frac{1}{\psi L/2} \tanh(\psi L/2)\right] \ . \tag{22}$$

In the above equations F is the force required the translate the flexure; P is the axial force; L is the flexure blade length; b is the flexure blade width; h is the flexure blade thickness; E is the elastic modulus of the flexure material; and $(\psi L/2)$ is given by $(3PL^2/2Ebh^3)^{1/2}$.

If the force used to translate the flexure is not applied precisely at the midpoint of the flexure blade length, the flexure will tilt as it translates, and this tilt is given by[8]

$$\theta = \left[\frac{\delta}{L}\right] \left[\frac{6(L - 2a)h^2}{3b^2L - 2h^2L - 6ah^2}\right] , \tag{24}$$

where θ is the flexure tilt; L is the flexure blade length; b is the flexure blade width; h is the flexure blade thickness; δ is the flexure translation; and a is the location of the force used to translate the flexure.

Fabrication errors introduce a tilt into a parallel spring guide when the flexure is translated. If the flexure blades are not the same length, the tilt error is given by[8]

$$\phi \cong \frac{\Delta L \delta^2}{2L^2 b} \ . \tag{25}$$

If the flexure blades are not parallel to each other, the tilt error when the flexure translates is given by[8]

$$\phi \cong \frac{\Delta y \delta}{Lb} \ . \tag{26}$$

In the above two error equations ϕ is the tilt error; ΔL is the flexure length error; Δy is the flexure out-of-parallelism error; L is the flexure length; b is the flexure width; and δ is the flexure translation.

For most mounting applications, the cross-strip flexure and parallel spring guide provide the necessary rotational and translational compliance. Certain applications require the shortest possible flexure length, or the stiffest flexure for a given available space. Two special configurations for this purpose are the cruciform flexure and the constant stress flexure.

Cruciform flexures are used to provide limited rotation in very confined spaces. Rotational compliance for the cruciform flexure shown in Figure 9 is given by[9]

$$\theta = \frac{ML}{8Gb^4} \frac{57(b/t)^4}{[19(b/t) - 12]} , \qquad (27)$$

where θ is the flexure angle of rotation; M is the moment applied to produce the desired angle of rotation; G is the flexure material shear modulus; L is the flexure length; b is the flexure width; and t is the thickness of the individual flexure blades of the cruciform.

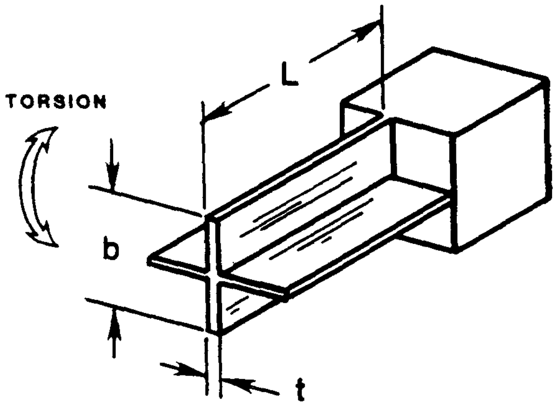

Figure 9. Cruciform flexure.

The tapered uniform-stress cantilever flexure blade is often used to provide a small range of translational motion in a very confined space. The end motion of the uniform-stress cantilever flexure shown in Figure 10 is given by

$$\delta = \frac{6FL^3}{Et^3(b_0 - b_1)^2} \left[b_0 - 3b_1 + \frac{2b_1}{(b_0 - b_1)} \log \left[\frac{b_0}{b_1} \right] \right] , \qquad (28)$$

where b_0 is the width of the flexure at the base; b_1 is the width of the flexure at the tip; t is the flexure thickness; L is the flexure length; E is the elastic modulus of the flexure material; δ is the motion of the flexure tip; and F is the force required to move the flexure tip.

Types of Flexure Mount

In 1964, Chin described a three-point tangential mount for use in a space telescope.[10] This type of tangential flexure mount, and derivatives of this design, have been widely used since for space telescope mirror mounts. Three flexures, equally spaced around the circumference of the mirror, are attached to the mirror edge. These flexures are either tangent bars, or cantilever flexures. In the tangent bar, the mirror is attached to the midpoint of the flexure, as shown in Figure 11. In the cantilever flexure, the mirror is attached to the end of the flexure, as shown in Figure 12. In both cases, the direction of flexure compliance is in the plane of the center of gravity of the mirror, or the mid-plane of the mirror.

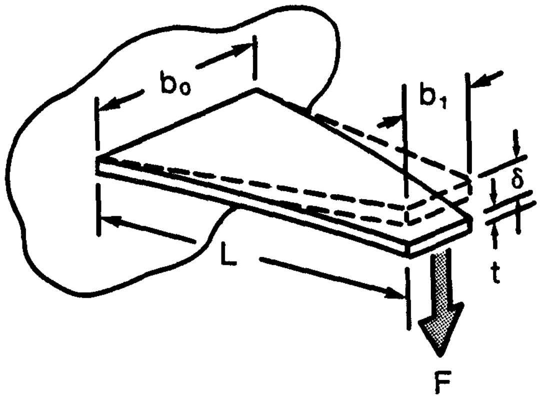

Figure 10. Uniform-stress cantilever flexure.

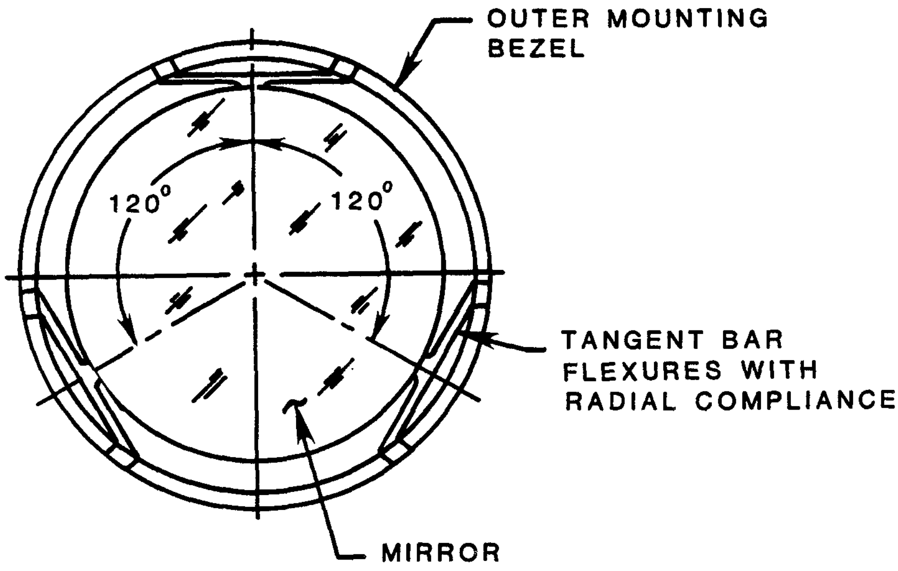

Figure 11. Tangent bar flexure mount.

In 1972, a 0.89-m, fused-silica, slotted egg-crate, lightweight mirror, mounted with cantilever-edge flexures was placed into orbit as part of the OAO-C.[11] The TEAL RUBY 0.5-m, lightweight, fused-silica primary mirror is mounted with similar cantilever-edge flexures.[12] Bosses extending from the edge of the TEAL RUBY primary mirror are used to attach a secondary system of flexures to the main cantilever flexure. The secondary flexures are used to provide moment isolation for the mirror to minimize surface figure distortion resulting from mount assembly error or distortion of the mount during temperature changes.

Another type of edge flexure mount was described by Hog in 1975.[13] Flexures are attached to the mirror edge, but the long axis of the flexures is parallel to the optical axis of the mirror. A similar system of flexures is described in 1981, in French Patent 8106724 (U.S. Patent No. 4,533,100; issued August 6, 1985). In addition to the flexures parallel to the optical axis, the French patent

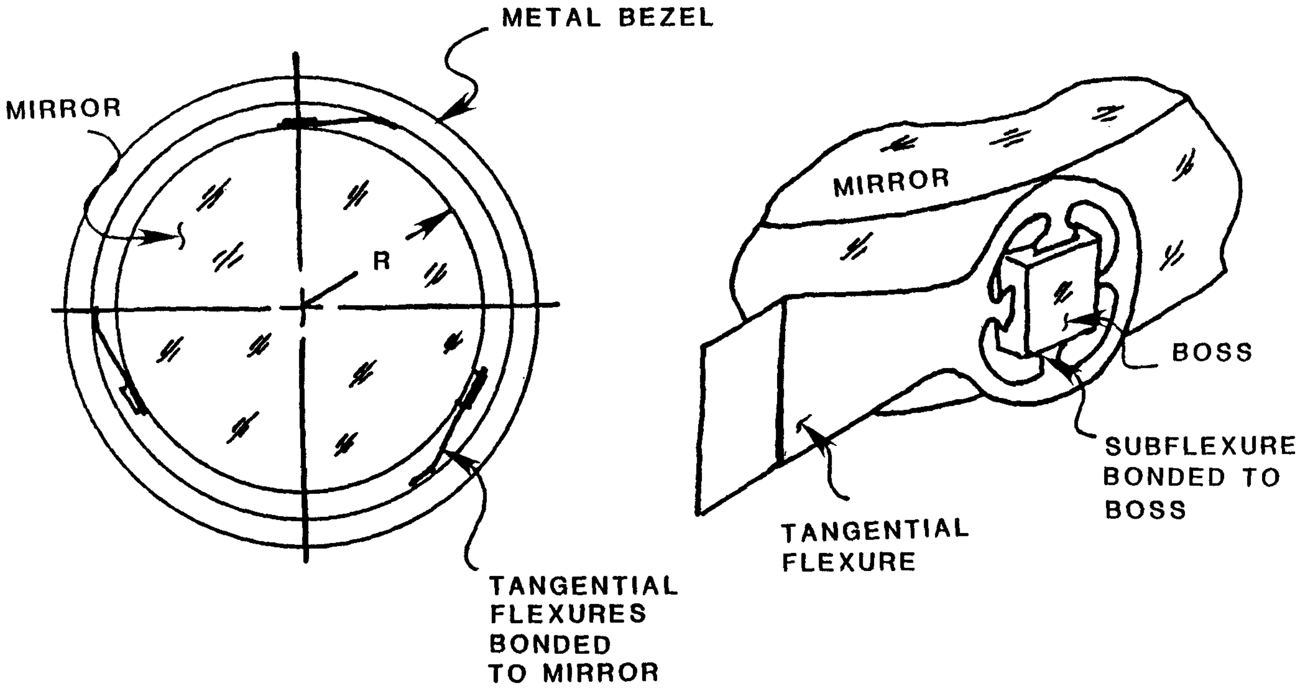

Figure 12. Cantilever tangent bar mounting.

uses an intermediate two-degree-of-rotational-freedom flexure assembly to isolate the mirror from mount distortion or assembly error. Cross-strip flexure hinges are used to provide the required rotational compliance, as shown in Figure 13. A further modification of the French patent design was described by Espiard in 1985.[14] In this modification, parallel spring guides replace the cantilever flexure. Use of parallel spring guides increases stiffness in comparison with a similar length cantilever flexure, and reduces the moment transmitted to the cross-strip flexures.

Space at the edge of a mirror for mounting is not available in some optical systems, leading to the use of flexure mounts that attach to the back of the mirror. In 1980, Schriebman and Young described the flexure mount for the IRAS primary mirror.[15] Titanium cantilever flexures were attached to pockets in the back of the mirror at the three equally spaced points on a common diameter. Cruciform flexures were placed between the end of the cantilever flexures and the mirror mounting surfaces to provide moment isolation for the mirror. The long axes of the flexures were parallel to the optical axis of the mirror. IRAS operated at 4 K; the flexures provided good isolation of the lightweight, 0.6-m-diameter, beryllium primary mirror from distortion caused by temperature changes.

Iraninejad, et al., in 1983 described a 0.5-m-diameter, fused-silica, double-arch mirror developed as part of the SIRTF program.[16] As shown in Figure 14, this mirror was mounted using T-clamps and sockets in the back of the mirror by means of three equally spaced parallel spring guides. The parallel spring guides were parallel to the optical axis and were fabricated from 6Al-4V ELI titanium. The mirror and mount system were tested at 7 K. Use of parallel spring guides in place of cantilever flexures increases stiffness as compared with cantilever flexures. A similar approach was used in the 0.5-m test mirror for the German Infrared Laboratory satellite described by Schlegelmilch and Altman in 1985.[17] The mirror was intended to operate at between 5 and 10 K. Mounting was by means of three mushroom-shaped sockets in the back of the mirror. Clamps

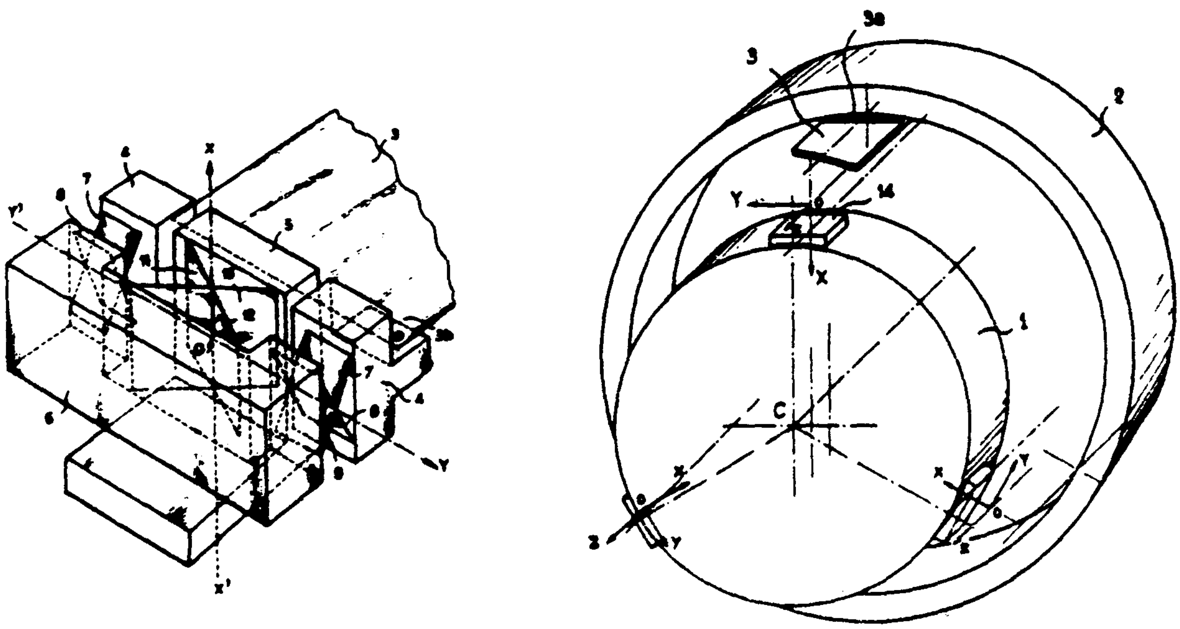

Figure 13. Edge flexure mounting (from U. S. Patent description).

located in the socket were attached to a two-degree-of-freedom, single-blade flexure system, which in turn was bolted to an Invar subframe. Use of the two-degree-of-freedom flexure system isolated the mirror from moments caused by mount error by providing rotational compliance. Parallel spring guides attached the Invar subframe to the aluminum mount and provided isolation from temperature changes.

Bipod flexures are used in many high-precision mirror mount applications. In a bipod flexure mount, three bipod flexures are attached to the mirror back or mirror edge. Each bipod flexure is equivalent to a two-strip rotational flexure, or cross-strip flexure, and provides rotational compliance. An advantage of the bipod flexure is virtual pivot motion; by changing the angles of the two flexures of the bipod, the instantaneous center location of the flexure assembly may be placed where desired, as shown in Figure 15. Use of bipod flexures places the center of rotation of the mounting flexures into coincidence with the midplane of the mirror, even when the flexures are attached to the back of the mirror. Bipod flexure geometry normally requires the use of a two-degree-of-freedom flexure at both the top and bottom of the flexure assembly. Very high stiffness is readily attained with bipod flexure mounts.

Conclusion

A wide variety of flexure mount designs are possible for high-resolution optical elements. These flexure mounts are intended to provide isolation of the mirror from distortion of the mount caused by temperature changes or assembly errors. Flexure mounts permit the relaxation of tolerances that are often required in other types of mirror mounts. In certain cases, such as cryogenic optics, flexure mounts may be the only possible mounting technique.

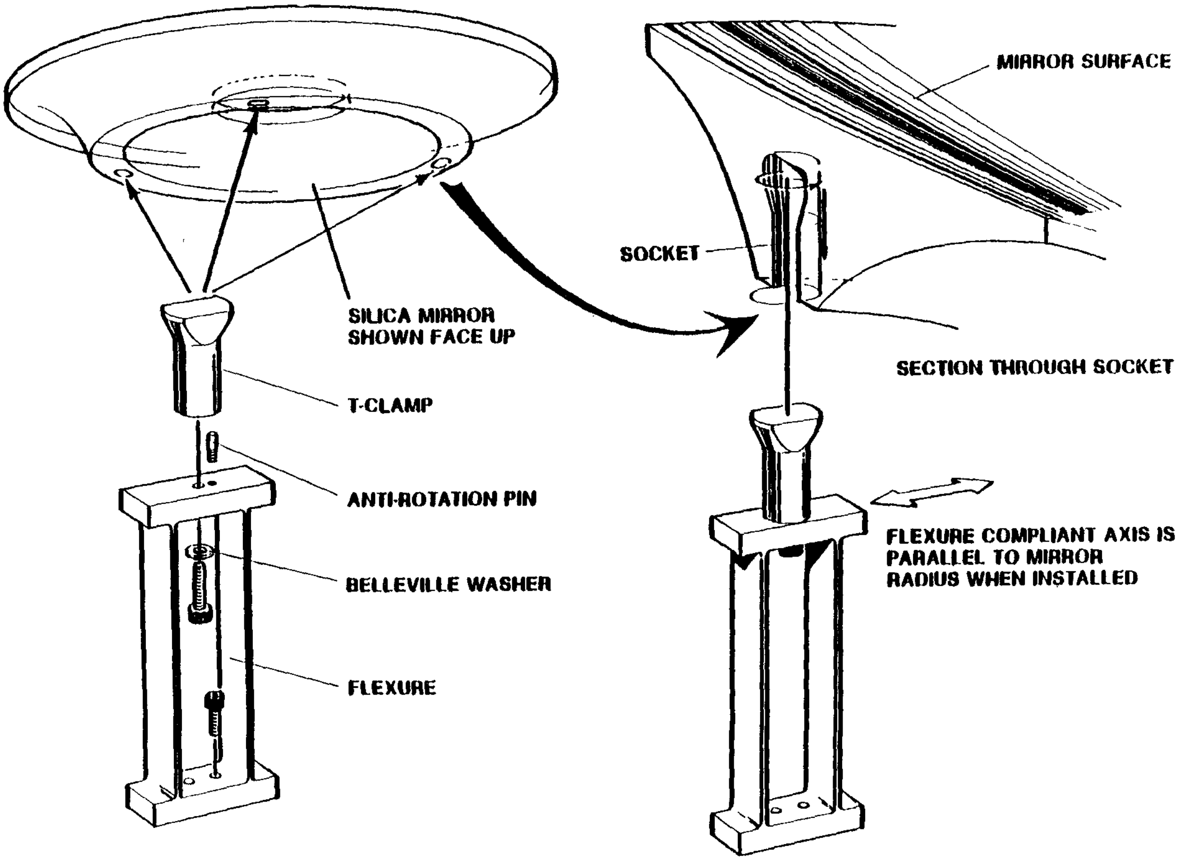

Figure 14. Parallel spring guide mounting (source: R. K. Melugin, NASA Ames Research Center).

Critical issues in flexure mount design include selection of the flexure material and fabrication of the flexure. Techniques ensuring dimensional stability and long life are required during fabrication. Flexure mount geometry must be planned using kinematic principles to provide stiffness and isolation from mount distortion.

Careful design of flexure mounts permits mounting lightweight optics of 0.6 m in diameter or greater, for use in space systems cooled to below 10 K, with mount-induced optical surface figure distortions of less than 65-nm RMS. Although expensive, flexure mounts are an important opto-mechanical technique.

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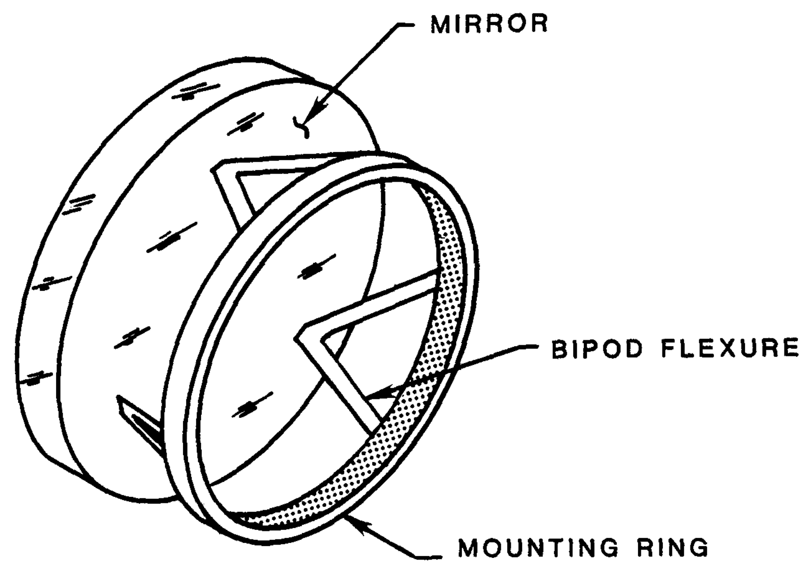

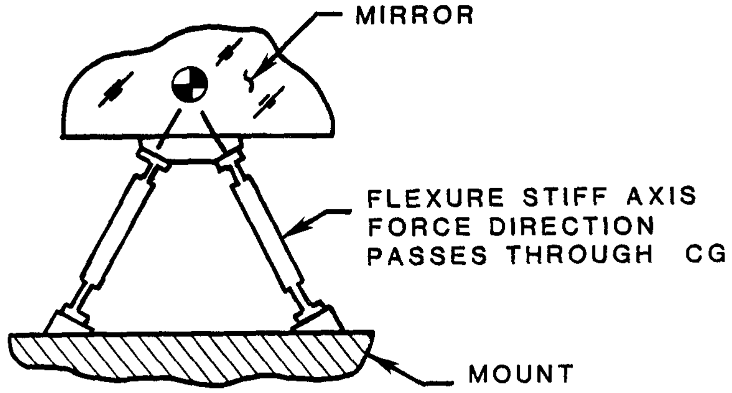

Figure 15. Bipod flexure mount.

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