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Adjustment Mechanisms

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7.1 Introduction

This chapter discusses the design aspects of different types of adjustment mechanisms used in optical systems. Various optical elements in a sophisticated system must be precisely aligned to each other to obtain an aberration-free image. In optical systems with very tight alignment requirements, it is more cost effective to manufacture the optics and their mounts to rather loose tolerances, and then employ adjustment mechanisms to align the optics relative to each other at assembly. Another class of adjustment mechanisms is used to move one or more optical elements of a system in real time to correct the image degradation caused by environmental effects.

Certain optical systems, such as those for submicron lithography, can have 10 to 15 mirrors and lenses that must be axially positioned relative to each other and centered on a common optical axis within tolerances of a few microns. To achieve these kinds of positioning accuracies, it is impractical and extremely cost prohibitive to manufacture the optics and its mounts to micron-level machining accuracies. For such optical systems, it is more practical and economical to fabricate the optics and the mounting hardware to loose tolerances, and provide adjustment mechanisms to align the optical elements relative to each other at the time of assembly. This class of adjustment mechanisms is designed for infrequent use and generally has manual actuators. Once the optical system has been aligned, these adjustments are locked in place to retain the alignment.

Another class of adjustment mechanisms are employed to despace or tilt an optical element in real time to compensate for the degradation of the image quality due to environmental effects. These mechanisms usually have motorized actuators and position readout sensors operating in a closed loop control system. Such mechanisms are generally used to correct focus and/or magnification errors in the optical systems due to thermal effects or any other environmental degradation.

This chapter covers the three basic types of adjustment mechanisms, namely: linear, rotary, and tilt mechanisms. Each mechanism consists of a number of parts such as an interface between the moving and stationary part, an actuator, locking, and preloading components. The selection criteria for these components of the adjustment mechanisms are discussed in detail. A number of example mechanisms have also been included for the benefit of designers. Finally, some guidelines for proper design and application of adjustment mechanisms in complex optical systems are presented.

7.2 Types of Adjustment Mechanisms

The three basic types of adjustment mechanisms are linear, tilt, and rotary mechanisms. A rigid body in space has six degrees of freedom, namely, the three translations and the three rotations about x, y, and z axes. An optical element in a system may need one or more of these translation or tilt (rotation about an axis) adjustments for alignment purposes. To avoid cross-coupling effects between different adjustments, the preferable approach is to stack single axis adjustments on top of each other to achieve a multi-axis adjustment mechanism.

A typical adjustment mechanism consists of five basic components. These components are

1. An *interface* between the moving optical element and the fixed structure
2. An *actuator* to adjust the moving element relative to the fixed structure
3. A *coupling* device or method between the actuator and the moving element
4. A *preloading device* to eliminate backlash in the mechanism
5. A *locking mechanism* to retain the adjusted position

For each of these five components, a number of choices are available to a designer depending on the type of adjustment mechanism. The most commonly used components for linear, rotary, and tilt mechanisms are shown in Tables 7.1, 7.2, and 7.3, respectively. The size and shape of these components generally are dictated by the space constraints and the service requirements for that particular application. For laboratory prototypes of optical systems, a number of commercial linear rotary and tilt stages and optical mounts with built-in adjustments and actuators are available. Unfortunately, these commercial adjustment mechanisms are quite expensive and bulky to be incorporated into actual optical systems. Therefore, practical adjustment mechanisms for a particular application usually have to be custom designed to provide the desired adjustment capabilities within the given space and cost constraints. This requires synthesizing an adjustment mechanism by selecting its components from Tables 7.1, 7.2, or 7.3, and then sizing and assembling these parts to meet the desired performance specifications.

7.3 Linear Adjustment Mechanisms

General Description

Linear mechanisms are the most commonly used adjustment mechanisms in optical systems. These mechanisms are employed when an axial or centration adjustment of an optical element is required. It is clear from Table 7.1 that a designer has a wide choice in selection of the components for a particular application. The selection of a particular type of component is dictated by the perfor-

TABLE 7.1 Choice of Components for Linear Mechanisms

Interface	Actuator	Preload	Locking	Coupling
Flexure	Coarse screw	Compression spring	Set screw	Ball/cone
Kinematic	Fine screw	Extension spring	Jackscrew	Ball/flat
Ball bearing	Micrometer	Flat spring	Locknut	Ball/socket
Roller bearing	Differential micrometer	Belleville washer	V-clamp	Threads
Air bearing	DC motor/linear motor	Curved washer	Collar clamp	Flexible coupling
Dovetail slide	Stepper motor		Epoxy	Lead screw
Flat slide	Piezoelectric		Control system	

TABLE 7.2 Choice of Components for Tilt Mechanisms

Interface	Actuator	Preload	Locking	Coupling
Cross-flexure	Coarse screw	Compression spring	Set screw	Ball/cone
Kinematic	Fine screw	Extension spring	Jackscrew	Ball/flat
Spherical bearing	Micrometer	Flat spring	Locknut	Ball/socket
Journal bearing	Differential micrometer	Belleville washer	V-clamp	
	Stepper motor	Curved washer	Epoxy	
	Piezoelectric		Control system	
	Linear motor			
	DC motor			

TABLE 7.3 Choice of Components for Rotary Mechanisms

Interface	Actuator	Preload	Locking	Coupling
Cross-flexure	Coarse screw	Compression spring	Set screw	Ball/cone
Ball bearing	Fine screw	Extension spring	Locknut	Ball/flat
Spherical bearing	Micrometer	Flat spring	V-clamp	Ball/socket
Journal bearing	Differential micrometer	Belleville washer	Collar clamp	Flexible coupling
Roller bearing	DC motor/linear motor	Curved washer	Epoxy	Worm/gear
Air bearing	Stepper motor	Torsion spring	Control system	Rack/pinion
	Piezoelectric			Belt/pulley

mance requirements such as the frequency, range, and resolution of adjustment, and other design factors such as the size, cost, and the load capacity of the mechanism. In the following sections, a number of design options for different parts of a linear adjustment mechanism are discussed, and general guidelines are presented to help a designer in selecting the suitable type of components for a particular application.

Interfaces for Linear Mechanisms

In a linear adjustment mechanism, the interface between the moving optical element and the fixed structure is generally determined by such design factors as the travel range, frequency of adjustment, shock, load capacity, cost, and size. If a long travel range is required and the mechanism is going to be adjusted frequently, a bearing interface must be used between the moving element and the fixed structure. Various types of slides suitable for linear mechanisms are illustrated in Figure 7.1. Ball and roller slides have a low friction and are suitable for long travel ranges. Ball slides (Figure 7.1[c]) are less expensive than roller slides (Figure 7.1[b]), but also have lower load capacity and accuracy (straightness of travel) as compared to roller slides. A dovetail slide, shown in Figure 7.1(c), has a high stiffness and load capacity and is less expensive. The main disadvantages of a dovetail slide are stiction and high friction. This slide is generally used in prototypes for laboratory type setups, where the adjustments are made infrequently and have to be simple and economical. Table 7.4 summarizes the linearity and running friction coefficients (RFC) for various types of slides.[1]

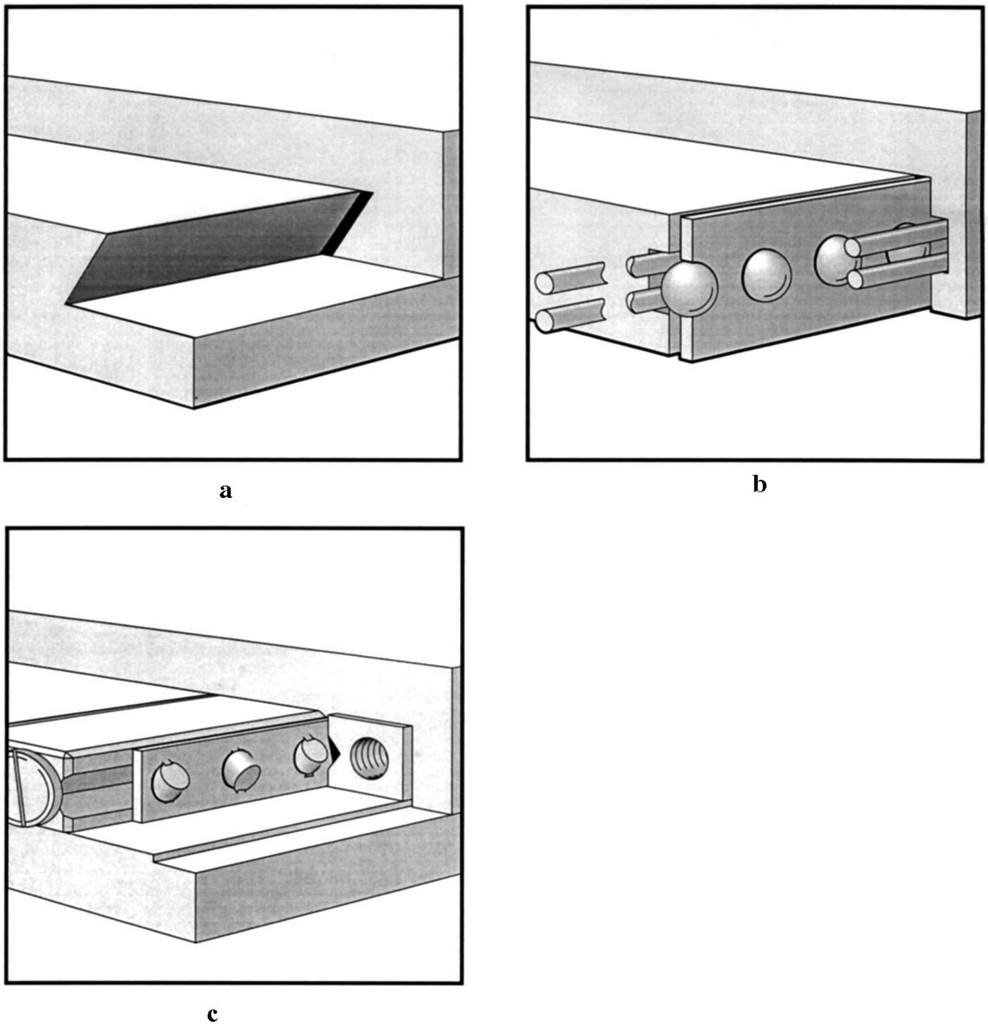


FIGURE 7.1 Types of linear slides for translation mechanisms. (a) Dove tail slide; (b) ball-bearing slide; (c) roller-bearing slide. (Courtesy of Newport Corporation, Irvine, CA.)

TABLE 7.4 Linearity, Running Friction Coefficients (RFC), and Load Capacity of Slides

Slide Type	Linearity (μm/10 mm)	RFC	Load Capacity and Stiffness
Dovetail	10	0.05–0.2	High
Ball	2	0.002	Low
Roller	1	0.003	Moderate

Hydrostatic Bearings for Linear Mechanisms

Hydrostatic bearings, which include both gas and oil bearings, are virtually free of friction and wear and have a negligible cross-axis runout. In optical systems, oil bearings are not commonly used because these are messy and present the risk of contaminating the optics. An optical system that requires an adjustment mechanism with a long travel range and high accuracy and load capacity can use a gas bearing. The pressurized gas is generally very clean and dehumidified air, but in some applications dry nitrogen or helium may be required. The main disadvantages of gas bearings are their cost and complexity. A remote and elaborate pumping and filtration system is required to supply clean and dry air. The design and fabrication of gas bearings is complex and expensive. The number and size of air jets, size of the air relief pockets, surface area of the bearing, and supply pressure of the air must all be taken into account when designing the air bearing for an application.

The simplest form of an externally pressurized bearing is a circular thrust bearing with a central jet as shown in Figure 7.2.[2] The pressurized air is forced into a recessed pocket in the middle, and it escapes along the periphery, thereby creating a very thin lubricating film of very high stiffness between the two surfaces. For incompressible flow, the load capacity for such a bearing is given by the following equation:

$$W = 0.69\left(p_s - p_a\right)\frac{\pi\left(b^2 - a^2\right)}{2\ln\left(b/a\right)} \tag{1}$$

where p_s = supply pressure
p_a = exhaust or ambient pressure
a = the radius of central recess
b = the outer radius of bearing

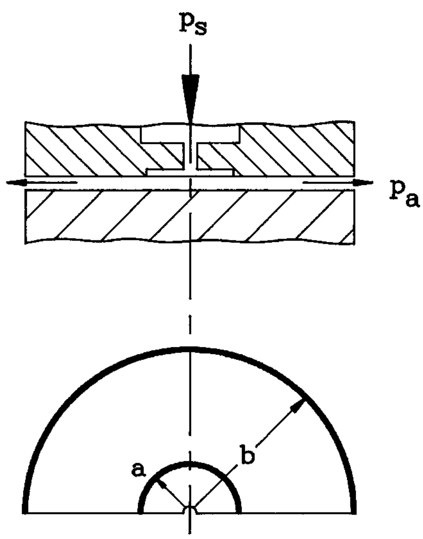


FIGURE 7.2 Schematic of a flat thrust gas bearing for linear mechanisms.

A bearing may also be designed with a ring of jets and radial grooves. It is very expensive to design and fabricate custom gas bearings; therefore, it is advisable to purchase standard bearings from commercial manufacturers, as far as feasible.

Flexures for Linear Mechanisms

Flexures are suitable for backlash-free adjustments over short travel ranges (1 to 2 mm). These have low friction and hysteresis, and do not require any type of lubrication. Flexures can be in several shapes such as a flat strip, circular, or universal. The design and fabrication of flexures are quite complex and are discussed in detail in Weinstein.[3] For each application, a flexure must be designed to have a specific stiffness which is determined by its length, width, thickness, shape and material. The materials having a high tensile modulus are more suitable for making flexures. The tensile modulus is defined as the ratio of allowable bending stress σ to the elastic modulus E. The

TABLE 7.5 Common Flexure Materials and Their Properties

Material	Yield Strength (ksi)	Elastic Modulus (Msi)	CTE (ppm/F)
Stainless steels			
302	35–40	28	9.6
440C	65	29	5.6
17-4 PH	125	28.5	6.0
Beryllium copper	85–110	19	9.9
Titanium	108	14.9	8.6
Invar 36	98	20.5	1.3
7075-T6 Al alloy	73	10.4	13.1

material with a higher σ/E ratio will have better compliance for a given length of the flexure. Some of the suitable materials for flexures along with their allowable bending stresses and elastic moduli are listed in Table 7.5.

Fabrication of flexures can be quite complex and expensive due to rigorous process control. To prevent failure due to stress concentration, the residual stresses due to machining must be minimized by selecting appropriate machining methods and proper heat treatment for stress relief. Similarly, a smooth-surface finish is desirable for a long life. When flexures are bent to move a component, a reaction force is induced in the component attached to the flexure. In applications where flexures are directly bonded to optical components, the reaction force from the flexure can produce a localized surface distortion. Therefore, the flexures must be designed for a proper stiffness to keep this distortion within acceptable limits.

The flexure design for linear (parallel) motion has been discussed in detail by Neugebauer.[4] For linear motion, the moving member is coupled to the fixed support through two parallel flat flexures as shown in Figure 7.3. These flat flexures have thin and rectangular cross sections, either solid or with cutouts as shown in Figure 7.4. These flat blade flexures are very stiff in tension and shear, but very compliant in bending. The deflection or travel due to actuator force F is given by:

$$Y = FL^3/2Ebt^3 \tag{2}$$

where L, b, and t are the effective length, width, and thickness of the flexure, respectively, and E is the elastic modulus of the flexure material.The vertical shear stress due to F is negligible, and the bending stress is given by:

$$\sigma = 3FL/2bt^2 \tag{3}$$

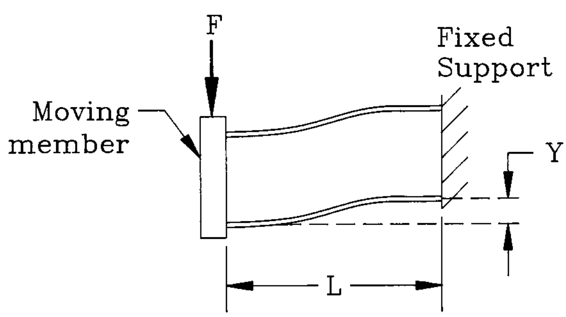


FIGURE 7.3 Parallel flat spring flexures for linear motion.

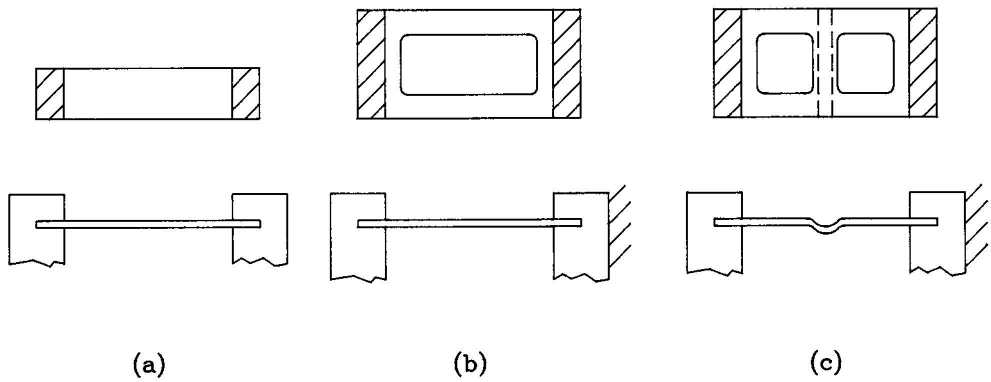


FIGURE 7.4 Types of flat spring flexure designs. (a) Solid; (b) single cut-out; (c) double cut-out.

If the weight W of the moving part is taken into account, the equations for deflection and the resulting stress become more complex. If the load W is compressive, the deflection and stress are given by:

$$\delta_c = \frac{FL}{W}\left(\frac{1}{K}\tan K - 1\right) \tag{4}$$

$$\delta_c = \frac{3FL}{2Kbt^2}\tan K + \frac{W}{2bt} \tag{5}$$

If the load W is tensile, the equations for deflection and stress are:

$$\delta_t = \frac{FL}{W}\left(1 - \frac{1}{K}\tanh K\right) \tag{6}$$

$$\delta_t = \frac{3FL}{2Kbt^2}\tanh K + \frac{W}{2bt} \tag{7}$$

The coefficient K in all these equations is defined by the following expression:

$$K = \left(\frac{3WL^2}{2Ebt^3}\right)^{0.5}$$

Actuators for Linear Mechanisms

The choice of a suitable actuator for a linear mechanism depends on travel speed, range, resolution, and frequency of adjustment, and cost, size, and weight requirements for the adjustment mechanism. For example, the motorized actuators are generally used for making frequent adjustments in real time. These include DC, linear, and stepper motors and piezoelectric devices. The main advantages of such actuators are long travel range, high resolution and velocity, and position readout capability. These actuators usually come with built-in position encoders and can be used in a closed loop control system. Therefore, the position of an optical component can be monitored, and the drifts due to environmental effects can be corrected in real time. The principle disadvantages of motorized actuators are their high cost and weight and large size.

Motorized Actuators

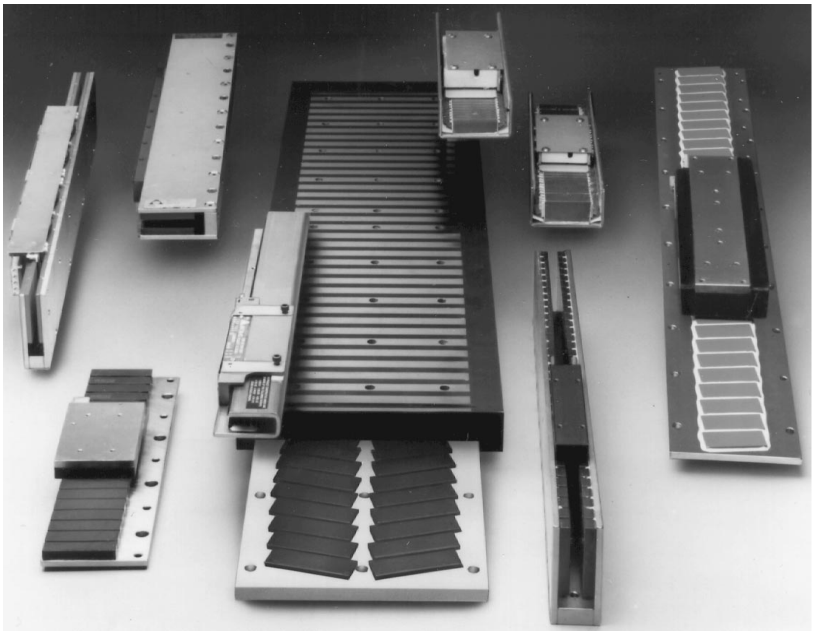
A number of motorized actuators are commercially available such as those shown in Figure 7.5 from Newport Corporation.[5] The actuator in Figure 7.5(a) is an ultraresolution electrostrictive type of actuator, which provides a piezo-class motion without any hysteresis and creep. It has an integrated fine pitch manual adjustment screw to center the electrostrictive fine travel where desired. The actuator shown in Figure 7.5(b) has a DC motor, which drives a precision leadscrew through low-backlash reduction gears. Both actuators provide a submicron-level positioning capability.

FIGURE 7.5 Types of motorized linear actuators. (a) Electrostrictive actuator. (b) DC motor. (Courtesy of Newport Corporation, Irvine, CA.)

Two other types of motors designed for high-end applications requiring micron-level positioning accuracy are the linear servomotor and piezoelectric inchworm motor. The brushless linear motors offered by Anorad Corp,[6] shown in Figure 7.6(a), consist of a fixed permanent magnet assembly and a moving coil assembly. A very small gap of the order of 0.5 mm is maintained between the coil assembly and magnets. When a current is applied to the coil, the electromagnetic force engages the moving mass with no mechanical link or friction. This friction-free force transmission eliminates jitter, stiction, backlash, and hysteresis. These types of linear motors are useful in applications requiring high acceleration (20 m/sec^2), high load capacity (up to 900 N), and high velocity (2 m/sec or more). A high accuracy planar stage combining air-bearing, linear motor, and servo-control technologies is shown in Figure 7.6(b). The model SG-2525 Servoglide x-y planar stage provides a travel range of 250×250 mm through direct drive motion. A closed loop servo-control system using an optical interferometer provides a long travel with dynamic yaw and rotation alignment capabilities. Table 7.6 compares the features of a brushless linear motor with those of a brush-type motor for precision linear motion applications. It is obvious that except for advantages in price and size, a brush type of motor has inferior performance in all other categories.

A piezoelectric actuator is generally used for short travel range requiring high resolution. A high voltage is needed to produce a movement of the order of a few microns. The piezo actuators have high load capacity. Typical disadvantages are hysteresis, creep, and nonlinearity of travel vs. the applied voltage. Burleigh's[7] inchworm motors, illustrated in Figure 7.7, can achieve ultraprecise

(a)



(b)

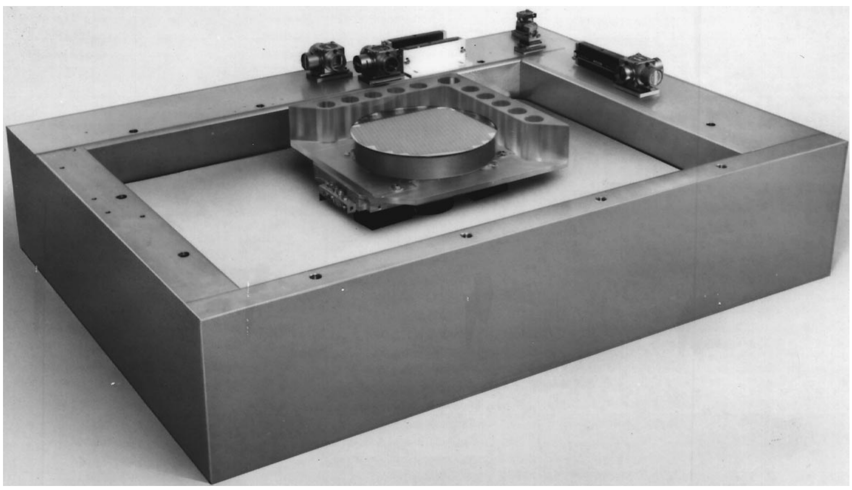


FIGURE 7.6 Brushless linear motors. (a) Single-axis motors; (b) two-axis air- bearing stage, Model SG-2525. (Courtesy of Anorad Corp., Hauppage, NY.)

positioning with a high degree of stability. These piezo-based motors provide angstrom- to nanometer-level resolution and travels of up to 200 mm. This feature of inchworm motors eliminates the need and complexity of a combination of coarse (fast speed)/fine (slow speed) positioning mechanism. The compact direct linear drive design of inchworm motors is backlash-free, nonmagnetic, and suitable for remote operation and ultrahigh vacuum applications.

A unique piezo-type actuator, depicted in Figure 7.8, has been recently introduced by New Focus, Inc.[8] under the trade name of *Picomotor.* With two jaws that grasp an 80-pitch screw, the Picomotor turns the screw much like as is done manually. A piezoelectric transducer slides the jaws in opposite directions. Slow action of the motor causes a screw rotation (right view), while a fast action due to inertia causes no rotation (left view). Traditional piezo-type actuators rely on contraction and expansion of the piezo to position or move an object. These actuators may exhibit backlash, hyteresis, and creep. The unique design feature of the Picomotor is that the piezo is only used to turn a screw, and not for holding the adjusted position. Therefore, this type of actuator is

TABLE 7.6 A Comparison of Motors Available for Precision Positioning Systems

Characteristic	Brushless Linear Motor	Brush-Type Linear Motor
Positioning accuracy	Good	Fair
Minimum step size	Good	Fair
Dynamic stiffness	Good	Good
Maximum velocity	Good	Fair
Constant velocity	Good	Fair
Settling time	Good	Fair
Friction hysteresis	Good	Fair
Lubrication/maintenance	Good	Fair
Varying load capacity	Fair	Fair
Cleanroom applicability	Good	Fair
Noise	Good	Good
Vertical applications	Fair	Fair
Durability	Good	Fair
Price <1-m travel	Fair	Fair
>1-m travel	Fair	Good

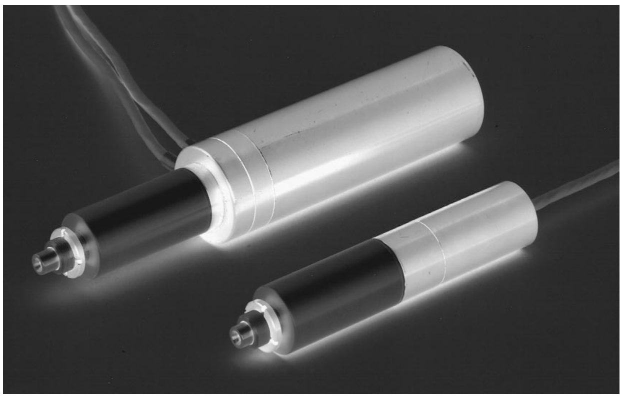


FIGURE 7.7 PZT-based *Inchworm* linear motors. (Courtesy of Burleigh Instruments, Inc., Fishers, NY.)

virtually free of backlash, creep, and hyteresis because a rigid screw is used to hold the position. Another advantage is that no applied voltage is required to hold the desired set position. Since this motor works in tandem with a fine threaded screw, its travel range is limited by the length (0.5- to 2-in. standard range) of the screw used. Some important performance specifications of Picomotor are listed in Table 7.7.

Manual Actuators

If an application does not require frequent adjustments, it is more cost effective to use screws or micrometers shown in Figure 7.9. If a position readout is not required, the screws are more economical and compact as compared to micrometers. The screws can be coarse, fine or of differential type depending on the resolution requirements. The principle of obtaining a fine linear travel from a differential screw is illustrated in Figure 7.10. By using two threaded screws in series, each with a slightly different pitch (number of threads/inch), a high resolution adjustment can be accomplished. When the screw is turned, the resulting differential translation T_d of the nut per revolution can be calculated by the following simple equation:

$$T_d = T_c - T_f \tag{8}$$

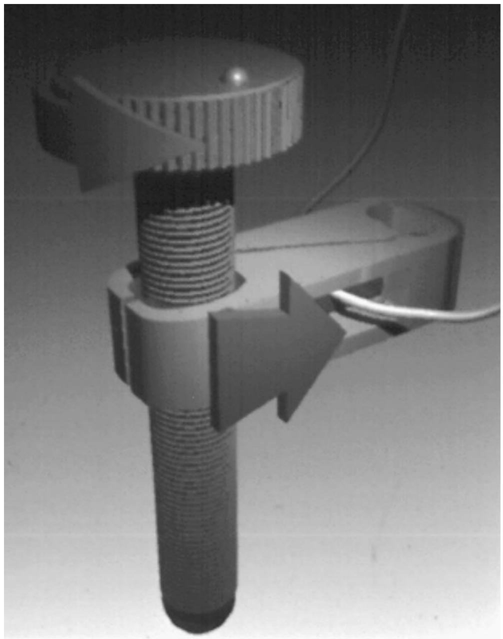


FIGURE 7.8 A piezo-type linear actuator, *Picomotor.* (Courtesy of New Focus, Inc., Sunnyvale, CA.)

TABLE 7.7 Picomotor Characteristics

Travel range	0.5, 1, and 2 in. standard
Resolution	<0.1 μm
Load capacity	2 lb
Speed	2–3 rpm
Repeatability	<0.1 μm with feedback
Lifetime	>5 × 10^8 pulses

where T_c = travel per revolution for coarse thread
T_f = travel per revolution for fine thread

For example, for a differential screw with a pair of 40 and 32 threads per inch, the net travel per revolution will only be 0.00625 in., i.e., the equivalent of 160 threads per inch pitch screw. The advantages of a differential screw are obvious from this example. The threads of 40 and 32 pitch can be machined inexpensively with standard tools in any shop, while machining the threads of 160 pitch is not trivial. The differential screws have their own disadvantages, also. These screws have higher friction due to the two nuts and, therefore, require a higher turning torque. The friction and torque can be reduced significantly by using ball screws in place of conventional screw threads, but this will add substantial cost. In differential screws, it is critical that the two threads be very concentric to minimize a premature failure due to rapid wear.

Regular, fine thread and differential screws and micrometers are commercially available in a large variety of sizes, shapes, materials, travels, and resolutions. The regular and differential micrometers are used for small travels, and are quite bulky and more expensive as compared to regular screws. Therefore, in general, micrometers are used in linear translation stages for laboratory prototypes only.

FIGURE 7.9 Fine-threaded screws and micrometers. (Courtesy of Newport Corporation, Irvine, CA.)

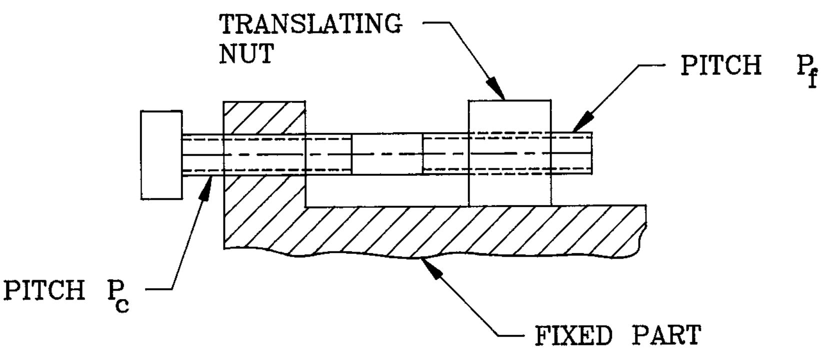


FIGURE 7.10 A differential screw with two threads of slightly different pitch can be used to produce a fine linear motion.

Coupling Methods for Linear Mechanisms

A number of options are available to couple an actuator to the moving component of an adjustment mechanism. The choice of a suitable method depends on such design factors as frequency of adjustment, shock requirements, weight, cost, and size of the mechanism. The tip of an actuator is generally rounded and polished to a high finish to minimize wear due to friction. The tip of an actuator can bear against a flat or a conical surface. The flat surface, shown in Figure 7.11(a), results in a point contact with the tip of an actuator and therefore produces high contact stresses. The ball/cone interface, shown in Figure 7.11(b), results in a line contact and produces much lower contact stresses. For infrequent adjustments of light weight components, a ball tip acting directly against a flat surface is the most simple and economical choice. The ball cone interface is more expensive to machine and is generally used for heavy components which must be adjusted frequently.

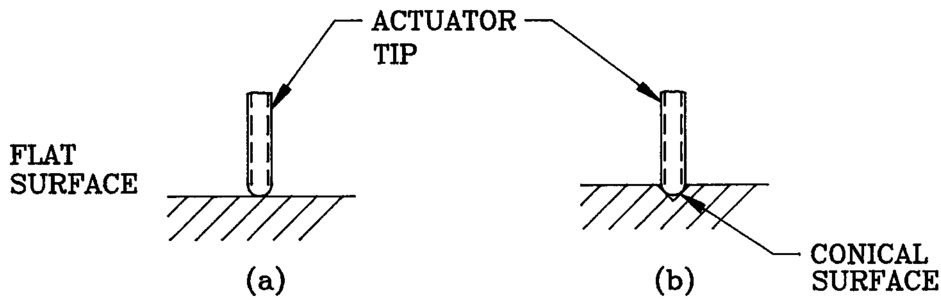


FIGURE 7.11 Two types of common interfaces for the round tip of an actuator. (a) Point contact with a flat surface; (b) line contact with a cone.

The moving component can also be directly attached to an actuator using a threaded attachment or a flexible type of coupling. The threaded coupling is not commonly used because a slight misalignment of the line of travel relative to the actuator axis induces lateral loads in the actuator and can cause rapid wear and damage to the actuator. A flexible coupling does not have this problem because slight misalignments in the mechanism are compensated by the flexibility of the coupling. The flexible couplings come in many types such as bellows, spring, Oldham, jaw, and Schmidt type. All these flexible couplings are relatively bulky and heavy, and are expensive and, therefore, are used in large mechanisms where heavy components need to be adjusted frequently.

For longer travels and heavy loads, an actuator may be coupled to the moving part through a lead screw shown in Figure 7.12. In this translation stage, the motor is coupled to the main screw while the moving platform is attached to the nut. The backlash and wobble of the nut can be minimized by preloading the nut. The friction can be reduced by using ball or roller bearings between the screw and nut. The lead screws are large in size and are expensive but they provide very linear travel over a long range. The torque T required to translate a load W is given by:

$$T = Wp/2\pi E \tag{9}$$

where p = the screw pitch or lead per revolution
E = the lead screw efficiency

Commercial lead screws have a minimal lead error (10 μm/300 mm) and come at a reasonable cost.

Preloading Methods for Linear Mechanisms

Most mechanisms employ some form of preloading arrangement to ensure a positive movement, free of backlash, when the adjustment is made. The selection of a suitable type of preloading method depends on such design factors as the range and frequency of adjustment, load capacity, cost, and size of the adjustment mechanism. In adjustments with a long travel range, a helical compression or an extension spring may be used for preloading the moving part against the stationary part. Normally the springs are placed around the adjustment screw, or can be centered between two adjacent adjustment screws to preload both screws with one spring.

A spring is characterized by its *spring rate k*, which is the ratio of an applied force F to the resulting deflection δ, i.e.,

$$k = F/\delta \tag{10}$$

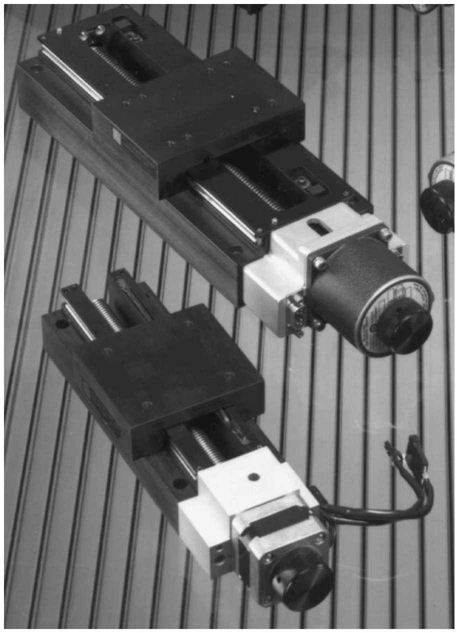


FIGURE 7.12 Commercial linear stages with lead screw coupling for a higher load capacity. (Courtesy of New England Affiliated Technologies, Lawrence, MA.)

Helical compression and extension springs are commercially available in a variety of spring rates, diameters, and free lengths. For an application, the required preload to hold the adjustment under shock loading is normally known. Also, the mechanism is designed to produce a known deflection in the spring. With this information, the required spring rate can be calculated by using the above equation, and then a suitable spring with the right free length and spring rate can be selected from a catalog. The number of coils in a spring must be sufficient to ensure that the spring wire remains within its elastic limit when the spring is at its maximum deflection. The number of coils in a compression spring also determines the minimum length, which is realized when adjacent coils come into contact. A long compression spring may buckle under stress. Therefore, long compression springs must be avoided unless they are guided by a rod or a screw through their center.[9]

If a mechanism is not locked by any other means, the natural frequency of a spring-loaded system can be calculated by:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_{st}}} \tag{11}$$

where g = the gravitational constant
δ_{st} = static deflection produced by the load

Helical springs are generally made from music wire or stainless steel wire. Music wire is very strong and hard because of the drawing process used in its production, and does not need to be further hardened after forming. Type 302 stainless steel wire springs are resistant to corrosion and are more commonly used in optical systems, although these have a lower (0.833) spring rate as compared to music wire springs.

Belleville and curved washers are used in compression mode to obtain high preloads over very small travels. A Belleville washer is a cone-shaped disk with a hole in the center as shown in Figure 7.13(a). When a load is applied, the cone flattens slightly, thus acting as a very stiff spring. These washers have very high stiffness and therefore, produce high loads as a result of relatively small

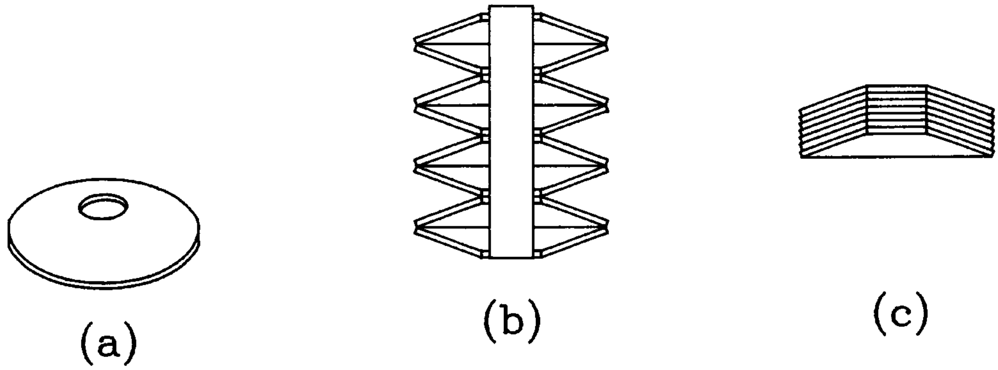


FIGURE 7.13 (a) A Belleville washer; (b) stacked in series for increased travel range; (c) stacked in parallel for higher load capacity.

compression. These washers are compact in size and are very economical and suitable for high shock applications. The travel range of mechanisms or its preload can be increased by stacking the Belleville washers in series or parallel as shown in Figure 7.13(b) and (c). When stacking these washers, a rod or a screw passing through their center is needed to retain them in the configuration shown.

Locking Methods for Linear Mechanisms

After an element has been adjusted, it must be locked in place to retain its adjusted position. The design factors affecting the choice of a locking method are the frequency of adjustment, shock, vibration, weight, size, and cost requirements. The locking can be accomplished by locking the actuator itself to prevent its accidental movement, tampering, or its drift under environmental vibrations. For micrometers or screw-type actuators, simple caps or covers can be used to prevent accidental movements. The movement due to vibrations can be prevented by using set screws to lock the rotation of actuator screws. The second option is to positively lock the moving element relative to the fixed structure, and several methods are available for doing this. If an optical element is going to be adjusted and aligned only at assembly, it can be locked in place by using epoxy bonding shown in Figure 7.14. The epoxy locking has the advantage of being a very low cost method. The main disadvantage is that the mechanism cannot be readjusted without breaking the epoxy bond, which is generally very difficult and also poses a risk of damaging the parts.

Jack screws or locknuts, shown in Figure 7.15, are economical ways of locking the mechanisms that do not require a great precision. The reason for lower accuracy is that a large force can be exerted on the adjusted element when a locknut or jack screw is tightened. This high force can cause a slight drift of the components. The advantages are that these locking components are

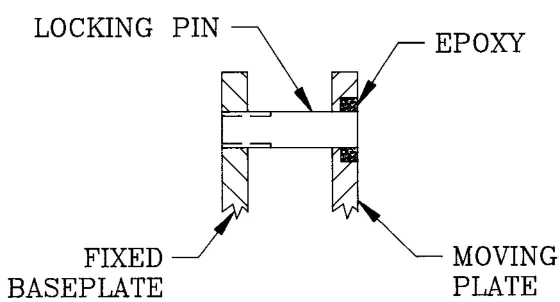


FIGURE 7.14 Typical locking scheme using epoxy bonding.

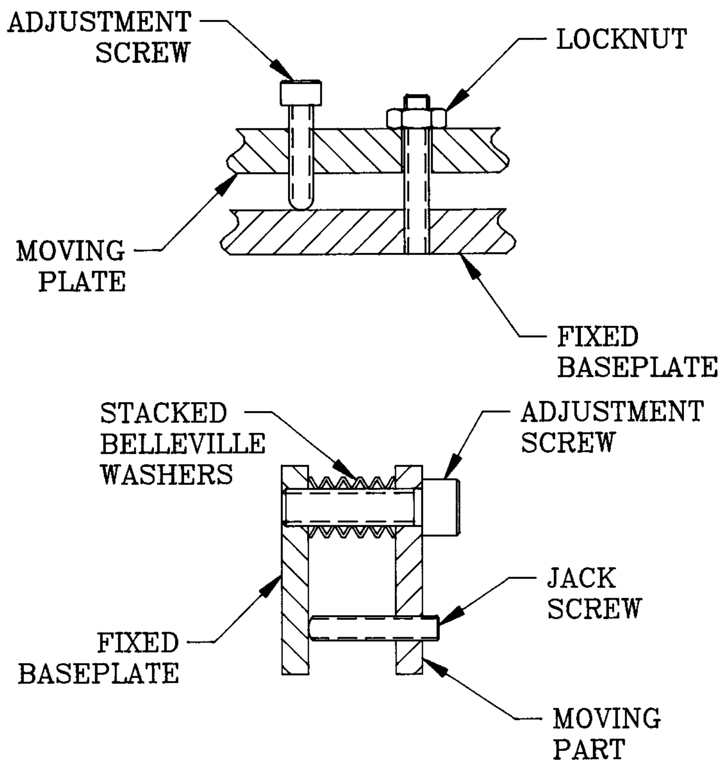


FIGURE 7.15 Two commonly used locking methods. (a) Locknut; (b) jack screw.

compact in size, and can be disassembled very easily without posing any risk of damage to the mechanism parts.

For motorized actuators, a closed loop feedback control system can be employed to retain the adjusted position of the moving element. As mentioned earlier, this method is expensive and is used only in those mechanisms which require real time adjustment.

Examples of Linear Translation Mechanisms

This section illustrates how various components of a mechanism can be assembled together to create a practical linear adjustment mechanism. These examples of a few simple linear mechanisms are meant to illustrate the principles only. As mentioned earlier, the size and shape of individual components are mostly application specific. Therefore, a designer can modify the parts of a particular mechanism, or combine various parts from different mechanisms shown in this section to design a mechanism that best meets the performance requirements. Most of the adjustment mechanisms described here are suitable for making height and axial (despace) adjustments.

In some examples here, the design of a mechanism at a single adjustment point is described. In actual practice, if a mirror mount is to be adjusted relative to a fixed structure, three similar and equally spaced adjustment points are needed around the periphery of the mirror mount. For round mirrors, windows, filters, and beamsplitters, the adjustment points must be equally spaced 120° apart, if feasible. For rectangular or square optics, the adjustment points can be located at any three corners. In such cases, if all three points are adjusted equally, a pure height or axial adjustment will result. If the adjustment is made at only one point, then a tilt motion occurs about an axis defined by a line joining the other two adjustment points. Therefore, most of the mechanisms discussed in this section are suitable for making axial as well as tilt adjustments.

Two simple linear adjustment mechanisms proposed by Tuttle[10] are shown in Figures 7.16 and 7.17. These mechanisms, though not very practical for optical instruments, illustrate the application of simple screws and sliding interfaces for linear motion. The mechanism shown in Figure 7.16 consists of two opposing bowed springs for preloading the moving part, which slides on the fixed structure. The bowed springs can be in the form of thin, rectangular sheet metal strips. The linear motion is obtained when one screw is threaded in, while the other screw is moved out simultaneously. Once the desired adjustment is achieved through an iterative process, the locknuts are tightened to retain the adjusted position. The principle disadvantages of this mechanism are high friction and poor resolution.

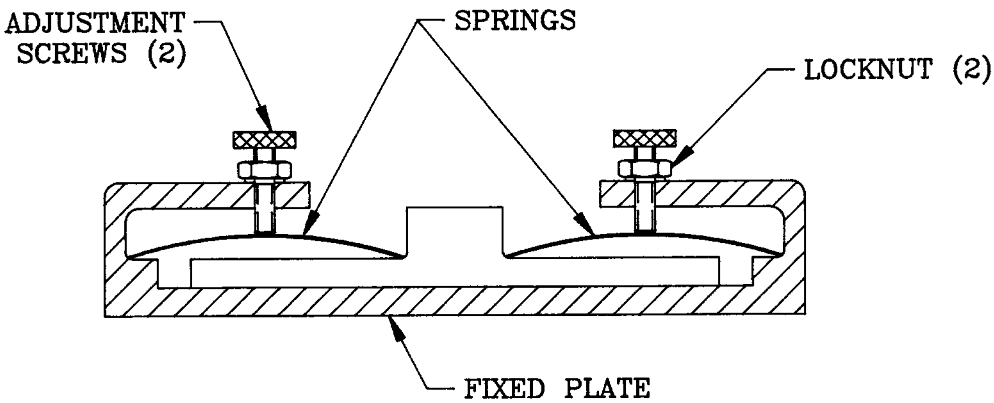


FIGURE 7.16 A simple linear mechanism with screw actuators and bowed springs for preloading.

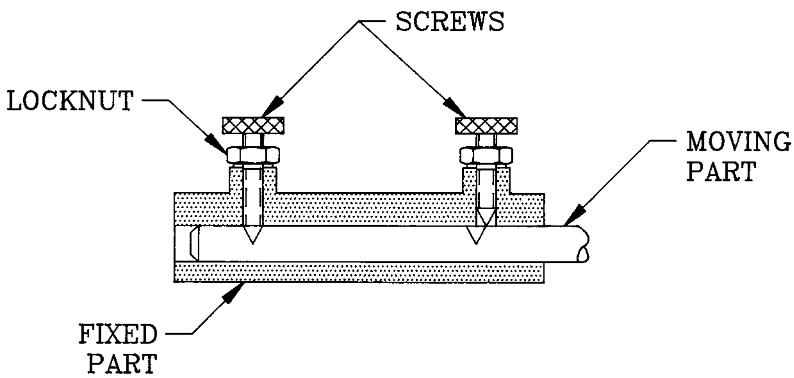


FIGURE 7.17 Two conical screws are used to slide a rod in this linear mechanism.

The linear mechanism, shown in Figure 7.17, has a sliding rod in a slightly oversized bore in the fixed structure. Two thumb screws with conical points engage the corresponding tapered holes in the rod. The center-to-center distance between the screws is slightly smaller than the corresponding spacing between the holes. When the screw on the right is threaded in while the left screw is moved out, the rod slides to the right. The opposing translation can be achieved by reversing the motion of the screws. The locking action of each screw, when threaded in, produces a side thrust on the threads. The screws must have long-enough engagement length to prevent cocking of the screws. The locknuts are provided to secure the adjusted position achieved through an iterative process. This mechanism also has friction and wear problems.

Another simple linear mechanism proposed by Elliot[11] is shown in Figure 7.18. The kinematic design of this mechanism employs two parallel cylindrical rods as guides. The moving plate has two vee and one flat contact with these rods. Simple screws or micrometers with opposing compression springs can be incorporated into this mechanism to produce linear sliding motion. Although this mechanism offers a long travel range for heavy load applications, it is too bulky to be incorporated in most of the practical optical systems.

A two-axis linear translation mechanism by Kittell[12] is illustrated in Figure 7.19. Two round-tip screws are acting on the moving part through flexures. The moving part is preloaded against the flexures by using in-line compression springs. This mechanism is suitable for low precision adjustment over short travels, and can have stiction problems.

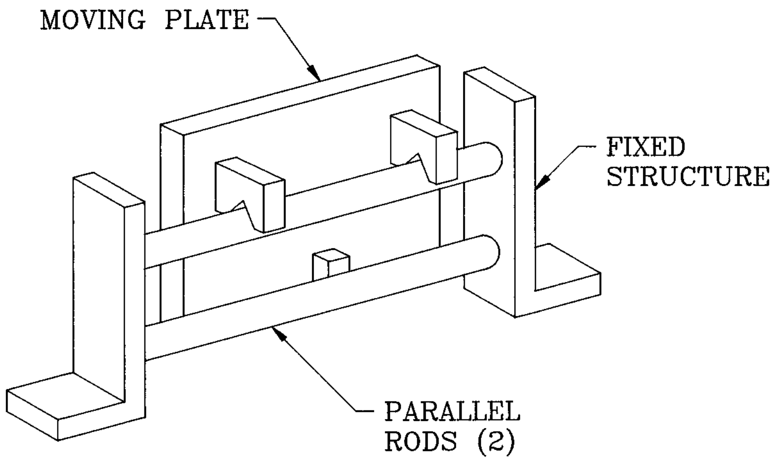


FIGURE 7.18 A linear mechanism with two parallel rods for long travels.

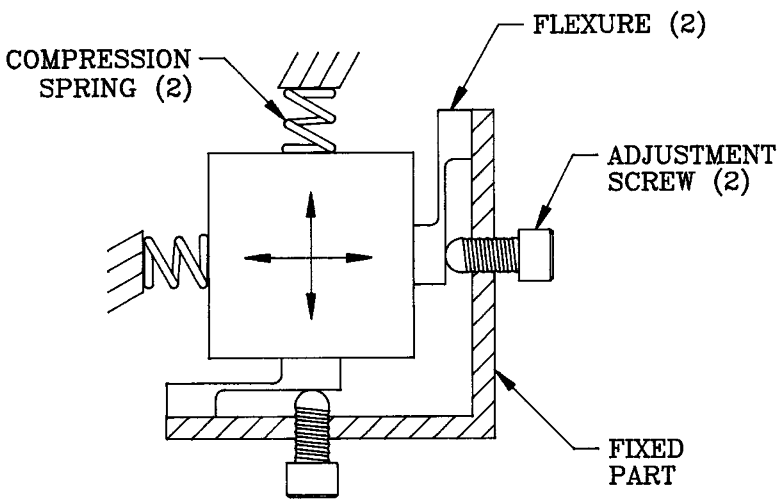


FIGURE 7.19 A two-axis linear mechanism using flexures and screw actuators.

A simple height adjustment mechanism by Ahmad,[1] depicted in Figure 7.20, has a regular screw threaded into a fixed baseplate. The moving part is spring-loaded against the screw by using a compression spring. A pair of swivel washers is provided to compensate for any nonparallelism between the moving and fixed plates. A typical mirror mount can be designed with three such adjustments on a triangular pattern. If the screws are moved equally, a pure height (or centration) adjustment is achieved. When only one of the screws is moved, a tilt results about an axis defined by the other two screws. Once the desired adjustment has been achieved, the jack screw is tightened to lock the mechanism. If an excessive force is used to tighten the jack screw, the adjusted position

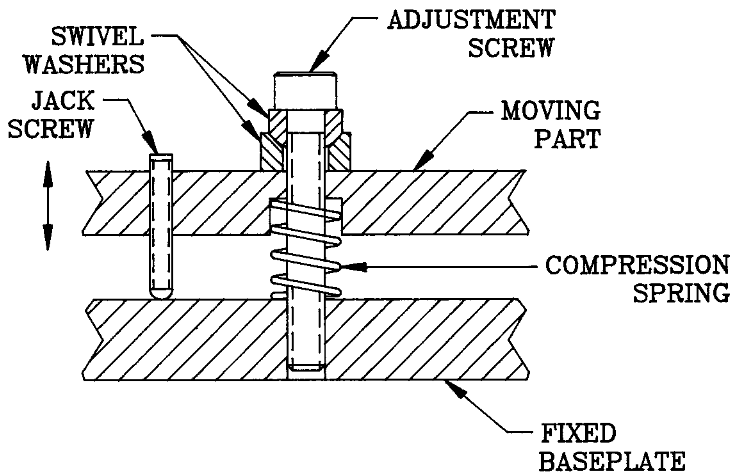


FIGURE 7.20 A height adjustment mechanism using spring-loaded screws.

may change slightly. This mechanism uses standard off-the-shelf commercial parts and is therefore, a good low cost option for linear adjustments with moderate resolution and travel.

The linear adjustment mechanism by Ahmad,[14] as shown in Figure 7.21, illustrates the principles of epoxy locking. This mechanism is essentially the same as the previous mechanism except that an extension spring, instead of a compression spring, is employed for preloading the moving part against the fixed structure. Here a fine threaded commercial screw with a ball tip is used for a finer resolution. The tip of the screw makes a line contact with a cone machined into the fixed plate to minimize the contact stresses for heavier load applications. The ends of the extension spring are retained by dowel pins going through them, thereby producing a force pulling the moving part toward the fixed plate. The locking in this mechanism is achieved by a locking pin, which is threaded into the fixed plate. The other end of the pin goes through an oversized counterbore in the moving plate, which is filled with a suitable viscous epoxy adhesive after the desired adjustment has been achieved. The clearance between the through hole and the pin must be controlled tightly so that it is only large enough to provide a free movement of the pin through the hole, but not too large so that the epoxy will run out of it during curing. Although this mechanism is more expensive than the previous one, it offers several advantages for some special applications. It is more suitable for high shock and vibration environments because the moving plate is rigidly locked against the fixed plate rather than merely relying on spring force to hold these two plates together. Moreover, the fine adjustment screws and springs can be removed for reuse after the epoxy has cured.

A simple linear adjustment mechanism,[15] which can be disassembled without destroying the adjustment, is shown in Figure 7.22. The adjustment bushing is threaded through the moving plate until the desired height is achieved. The clamp nut is then locked to retain this adjusted position. The clamp screw holding the parts together can be removed to disassemble the moving plate without losing the adjustment. For a finer resolution, this mechanism can be modified by incorporating differential screw threads as illustrated in Figure 7.23. The middle bushing has slightly different pitch threads on each section. The coarse adjustment is made by rotating the lower nut alone. When the middle bushing is rotated, a fine adjustment results due to a differential rotation in the two pairs of threads. This mechanism can also be disassembled without destroying the adjustment by removing the clamp screw. These simple mechanisms are suitable for applications requiring disassembly frequently.

Another linear mechanism employing differential threads by Kittell,[12] which is suitable for precision adjustments, is illustrated in Figure 7.24. The frictional force has been minimized by using spherical nuts in conical seats to achieve a line contact. The coarse adjustment is obtained

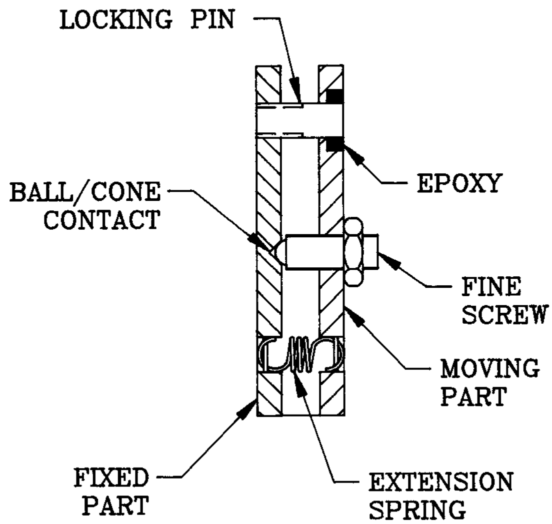


FIGURE 7.21 A linear mechanism using a fine threaded screw with epoxy locking and a ball–cone contact for a higher load capacity.

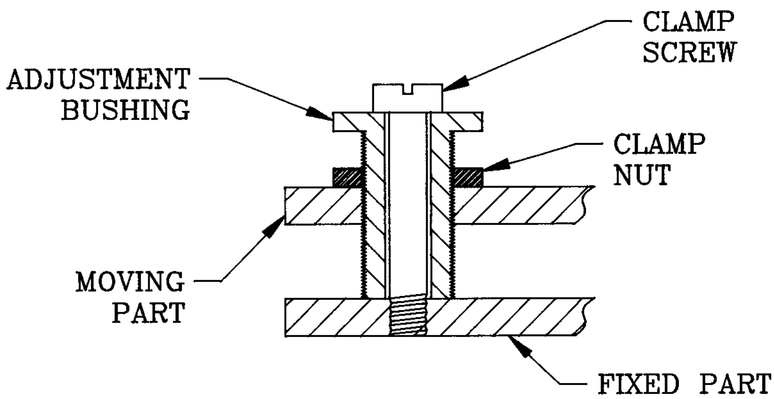


FIGURE 7.22 A linear mechanism with a threaded bushing and clamp screw suitable for applications requiring disassembly.

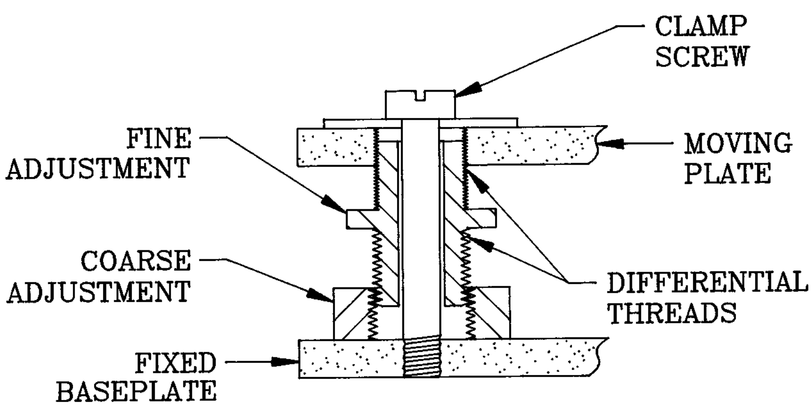


FIGURE 7.23 A linear mechanism with a bushing using differential threads for finer resolution.

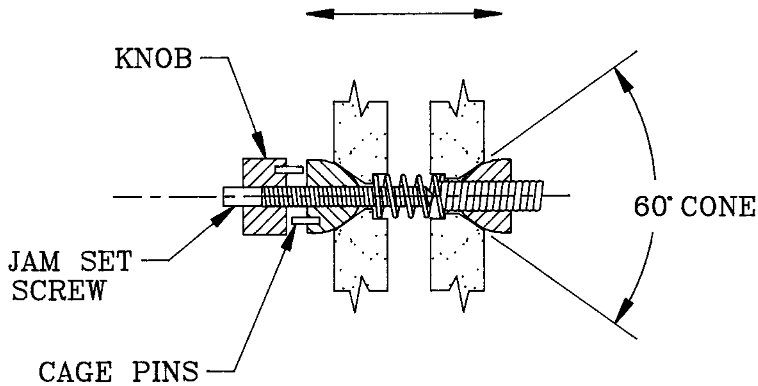


FIGURE 7.24 A high resolution linear adjustment mechanism using differential threads and spherical nuts to reduce friction.

by rotating the right side nut. When the left knob is rotated, a differential motion is achieved until the two cage pins engage. When this engagement of the pins occurs, the left nut rotates with the knob, thereby resulting in a coarse adjustment again. This clever design minimizes the time required to achieve very fine adjustments over a long travel range. A compression spring has been used in this mechanism to preload the spherical nuts into conical seats. Due to the precision required in machining the various parts of this mechanism, it is quite expensive and is recommended for high end applications only.

Ahmad[16] has described a high performance real-time focus adjustment mechanism, which is shown conceptually in Figure 7.25. An off-axis aspherical mirror is mounted in its cell through three tangent bars. The mirror cell is suspended by two pairs of flat blade rectangular flexures. A moving coil linear motor pushes at the center of the mirror cell. Since the rectangular flexures are much longer as compared to their height, they are very stiff in tension and torsion but comparatively quite compliant in bending. The flexures are arranged to be parallel to each other and orthogonal to the optical axis of the mirror. When the actuator pushes or pulls on the mirror cell, a pure axial movement of the mirror occurs along its optical axis. A closed loop control system monitors the image quality at the focal plane, and adjusts the focus in real time to compensate for environmental changes. This focus mechanism is simple in design but has a high resonance, and therefore, is suitable for high performance lithography applications requiring micron-level adjustment capabilities.

Lens Centration and Focus Mechanisms

This section describes some simple mechanisms suitable for centration and focus adjustments of single and multiple lenses. These mechanisms are used for low precision applications such as ordinary cameras and binoculars. For high precision applications, such as lithographic lens assemblies, the lens cells are normally diamond machined, and special assembly tooling, such as rotary air bearing tables, air gages, and interferometers, is employed to make the optical axis of the lenses collinear with the mechanical axis of the diamond machined cells.[17] These assembly methods are not discussed here because they do not fall into the category of adjustment mechanisms.

A simple double eccentric mount, shown in Figure 7.26, is frequently used in binocular objectives.[18] The lens is bonded into a ring with an eccentric outer diameter, which rotates inside another ring whose inner diameter is eccentric relative to the lens optical axis. Since the axis of ring rotation is offset from the axis of the lens, a centration adjustment results when the lens cell is rotated. The two eccentric rings are bonded together after making the adjustment by injecting epoxy through the radial holes. This adjustment mechanism provides a coupled low resolution two-axis centration adjustment.

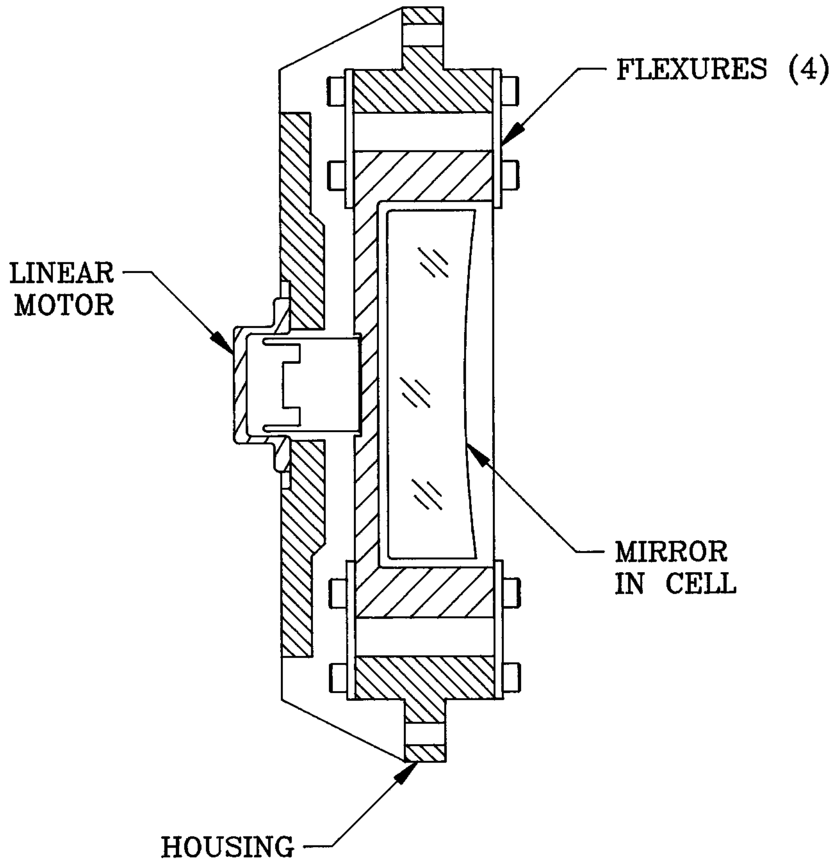


FIGURE 7.25 A real-time focus adjustment mechanism using flat blade flexures and a linear motor.

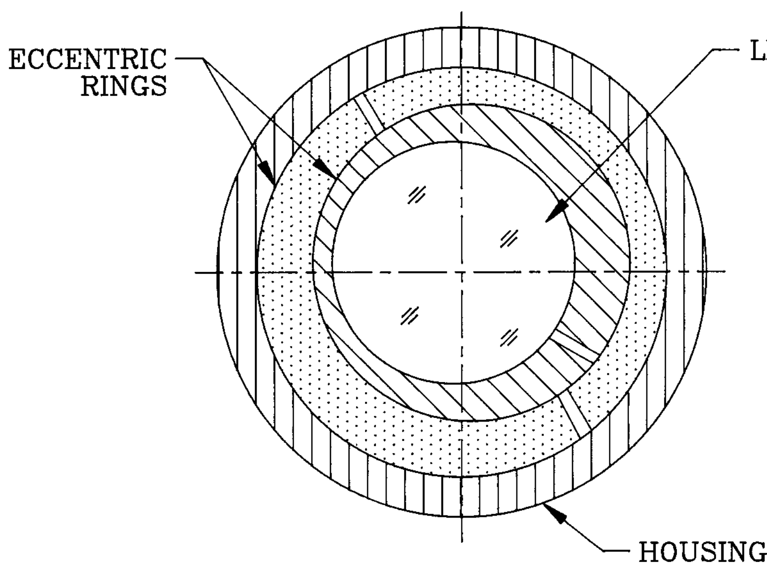


FIGURE 7.26 A pair of eccentric rings are used to adjust the lens centration.

A lens cell with four equally spaced radial set screws is shown in Figure 7.27.[18] A pair of opposing set screws is used to move the lens radially by moving one screw out while threading the other screw in. This arrangement provides an independent centration adjustment in two directions. Nylon screws can be used to avoid the risk of damage to the lens. This mechanism is suitable for lenses with thick edges, which are strong enough to withstand a compressive stress, without distortion or breakage. The desired centration is achieved through an iterative process, and its accuracy is dependent upon the pitch of screws used. Once the lens has been adjusted, it can be bonded to the cell by using dabs of epoxy, or by filling the gap between the lens OD and the cell ID with a suitable RTV elastomer. Again, this low cost lens centration mechanism is suitable for low accuracy applications.

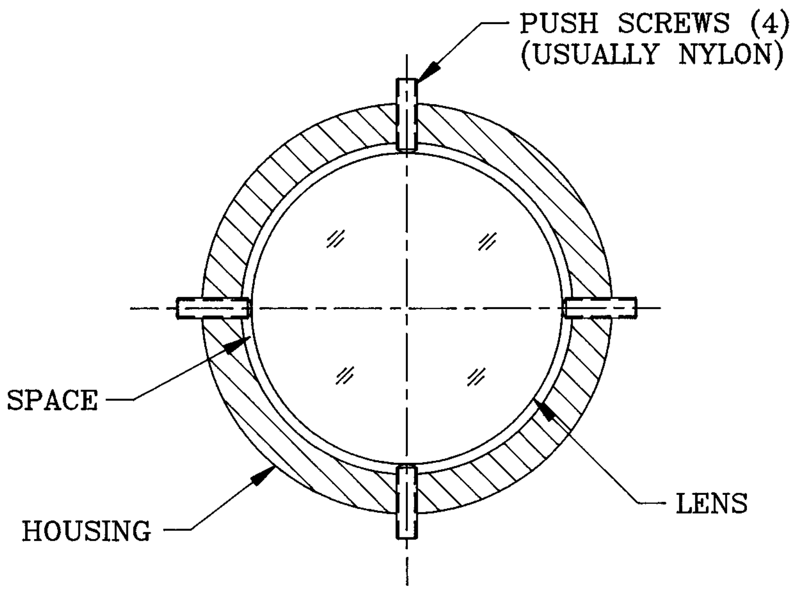


FIGURE 7.27 Lens centration using two pairs of opposing radial screws.

Another centration mechanism, suitable for higher precision applications with shock and vibration environment, has been proposed by Vukobratovich,[19] as illustrated in Figure 7.28. The lens is first assembled into its cell by bonding it in place, or by using a threaded retainer. Three equally spaced oversized holes for dowel pins are provided in the cell. The cell is then placed in a housing, which has a sufficient clearance for the lens cell OD, and has three equally spaced radial centering screws and three pressed-in dowel pins. The lens cell can be moved and centered by threading in one of the screws, which pushes the cell against the two other screws. Once the adjustment has been made, the clearance between the pins and oversized holes in the cell is filled with a suitable epoxy. The advantages of this mechanism as compared to the previous one are quite obvious. First, the lens is protected from any direct compressive loads, and the lens and cell ODs do not need to be machined precisely. The lens is not affected by the shrinkage of epoxy, also. The accuracy of the adjustment can be improved by using fine pitch screws. The negative features are a higher cost due to a number of extra parts and the permanent nature of the assembly because it cannot be disassembled easily if the lens was not adjusted correctly.

A simple focus (axial spacing) mechanism for lens assemblies has been proposed by Ginsberg[20] (Figure 7.29). The right-side lens is assembled into a cell, whose OD is threaded. This lens assembly can be translated axially with respect to the other lens group by rotating it, thereby moving it in or out of the housing. Once the correct focus is achieved, the threaded clamp ring is tightened against the housing to lock the adjustment in place. If the clearance between the internal and external threads is not controlled, excessive slop may introduce tilt errors. As no provision has been made for tilt adjustment, the common interface between the two housings must be machined square with the optical axes of both lens assemblies.

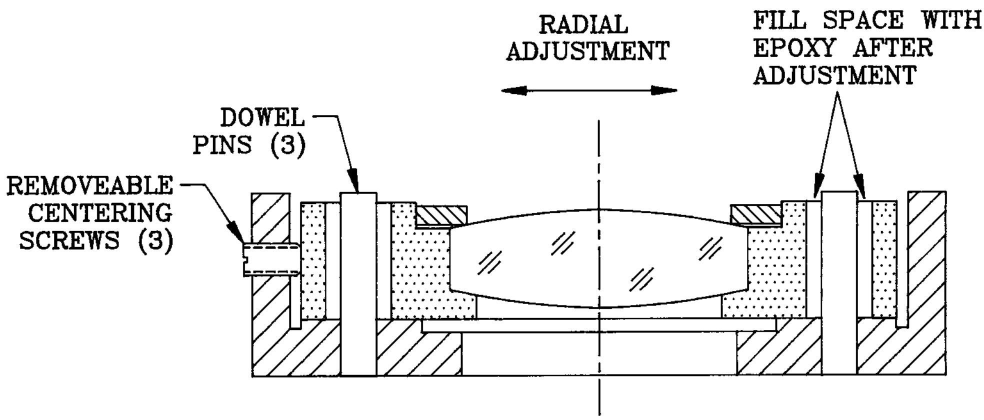


FIGURE 7.28 A lens centration scheme using epoxy and pins for precision applications.

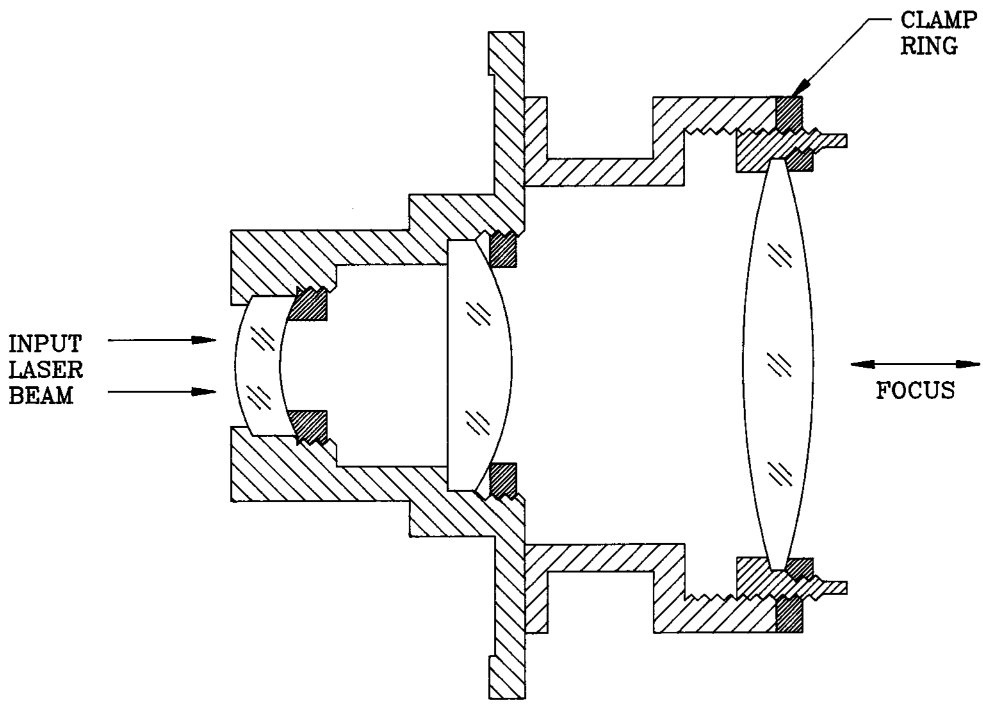


FIGURE 7.29 A lens focusing mechanism using a threaded lens cell.

Another fine-focus adjustment mechanism utilizing differential threads has been proposed by Jacobs[21] as depicted in Figure 7.30. The lens assemblies A and B are threaded into a focusing ring using two slightly different pitch threads. When the focusing ring is rotated, the relative axial spacing between the two lens assembly changes. Each lens cell must be constrained against rotation by providing a slot in the cell OD, which is guided on a fixed pin to result in a pure translation without rotation of the cell. Very fine-focus adjustments can be made by such a simple and inexpensive mechanism by selecting the right type of thread pitches, which differ only slightly. This type of focusing mechanism is commonly used in the camera objective lenses.

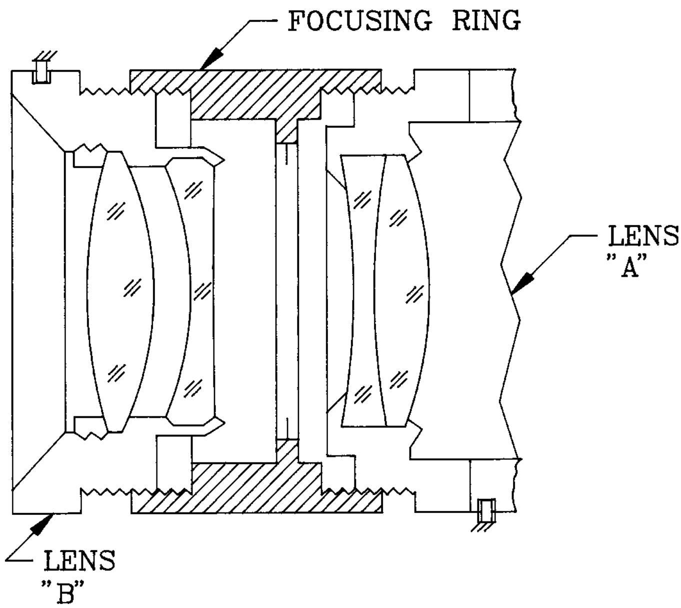


FIGURE 7.30 A fine-focus mechanism using differential threads for camera objective lenses.

7.4 Tilt Adjustment Mechanisms

General Description

Tilt adjustments of optical components such as mirrors, lenses, prisms, and diffraction gratings are frequently required to optimize the image quality in optical systems. A tilt mechanism can be designed to provide an adjustment about one axis or two mutually orthogonal axes. As mentioned earlier, tilt and linear adjustment mechanisms are quite similar in design and basically use the same kind of components. The design factors for selecting a suitable type of component for tilt mechanisms are the same as for the linear mechanisms. If an adjustment mechanism is designed with three mutually orthogonal adjustment points, it can be used to perform linear as well as tilt adjustments. When the three actuators are moved equally, a linear movement results. However, if only one of the actuators is moved, a tilt adjustment about an axis defined by the two others is achieved. The parts, which can be used as the basic building blocks for tilt adjustment mechanisms, are discussed briefly in this section. Also, some examples of tilt mechanisms are presented to demonstrate the basic design principles.

Interfaces for Tilt Mechanisms

The tilting component can be attached to a fixed structure through rotary bearings (journal, ball, roller, air), flexures (Bendix, flat blade), or a traditional kinematic interface. The trade-offs for these types of interfaces have already been discussed under linear mechanisms. A number of commercial single and two-axis tilt stages are available. These tilt stages employ a semikinematic interface between the fixed base and the moving tilt platform. The construction of a typical two-axis tilt stage is shown in Figure 7.31. The three-point interface between the tilt and fixed plates consists of hemispherical balls, which locate into a cone, a v-groove, and on a flat surface of the fixed plate. Two of the balls are rounded tips of the micrometers or fine pitch screws, and are positioned on mutually orthogonal axes with respect to the fixed ball. The interface between the two plates is preloaded by two extension springs located midway between the fixed and moving contact points.

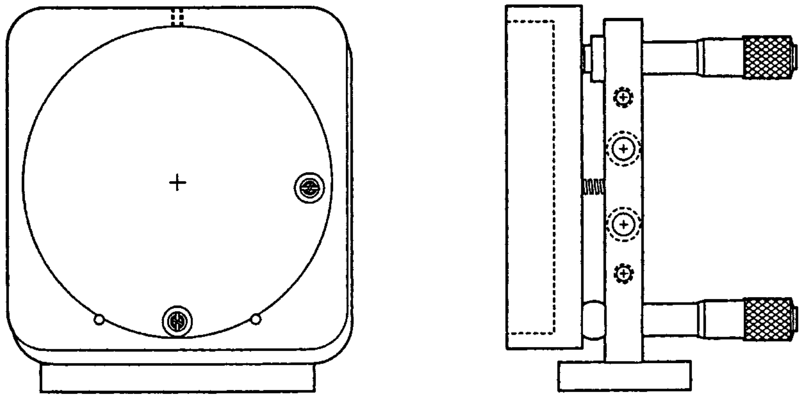


FIGURE 7.31 A typical two-axis commercial tilt stage (Courtesy of Newport Corporation, Irvine, CA.)

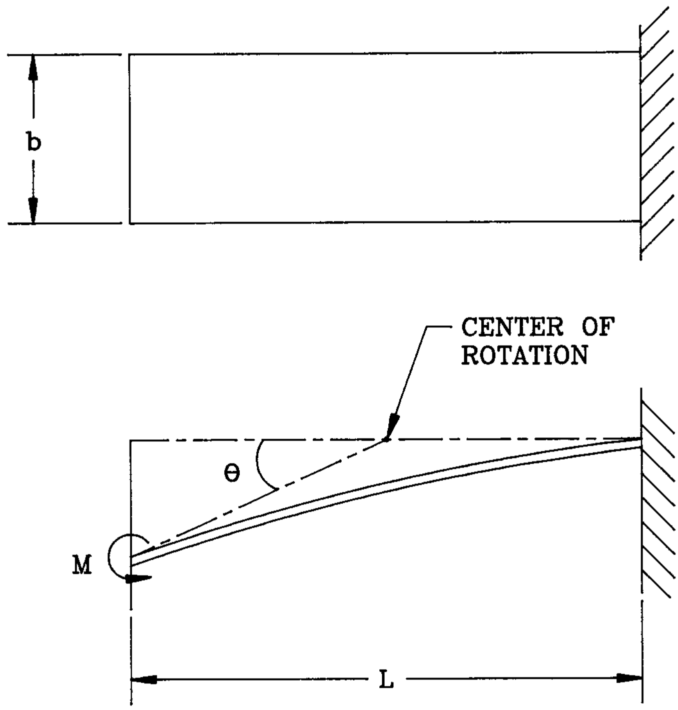


FIGURE 7.32 A flat blade flexure for small tilt angles.

Various configurations of flexures can also be used to attach the tilting part to the fixed structure. For small angles of rotation, a flexure strip shown in Figure 7.32 can be analyzed by the same equations used for beams that have end loads. If the axial load at the end is zero, the tilt angle θ is given by:

$$\theta = \frac{12ML}{Ebt^3} \tag{12}$$

where M = the applied bending moment
L = the length of the flexure
E = the elastic modulus of flexure material

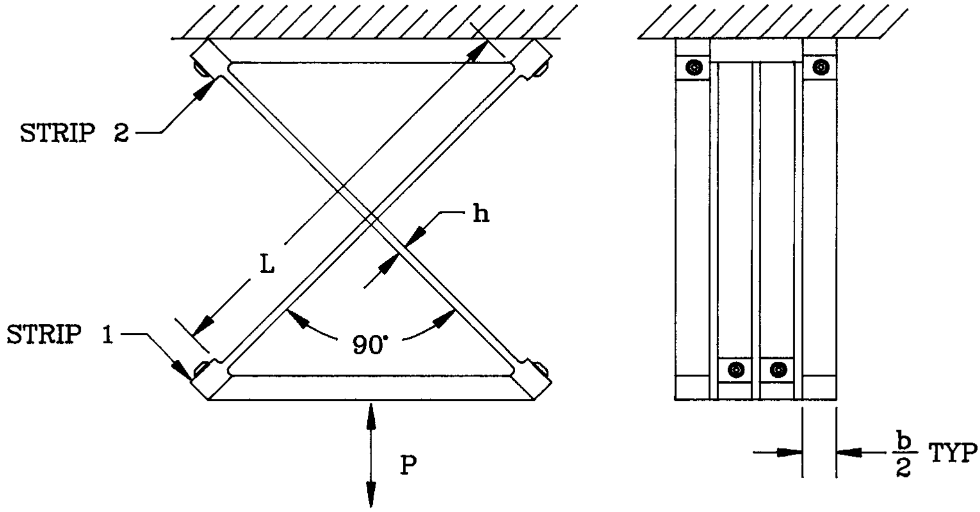


FIGURE 7.33 Schematic of a two-strip (cross) flexure for small tilts.

b = the width of the flexure
t = the flexure thickness

A two-strip flexure pivot, shown in Figure 7.33, can be used for small tilt angles.[3] This type of flexure is commercially available in the form of a pivot bearing. For a given tilt angle θ, the required bending moment M for a compressive load P can be calculated from the following equations:

$$M = \frac{EI\lambda}{2}\left[\frac{L\lambda}{2} + \cot\frac{L\lambda}{2}\right]\theta \tag{13}$$

If P is a tensile load, then the moment M is given by:

$$M = \frac{EI\lambda}{2}\left[\coth\frac{L\lambda}{2} - \frac{L\lambda}{2}\right]\theta \tag{14}$$

where I = the moment of inertia of the single strip
E = the elastic modulus of strip material

The parameter λ in these equations is defined by the following expression:

$$\lambda = \sqrt{\frac{P}{EI}}$$

A right circular flexure, shown in Figure 7.34, can be used in applications requiring a well-defined center of rotation and high stiffness.[22] For this special configuration of the flexure, the center of the cutting radius R lies on the edge of the flexure. The bending stiffness of this flexure can be estimated by:

$$\theta = \frac{9\pi MR^{0.5}}{2Ebt^{2.5}} \tag{15}$$

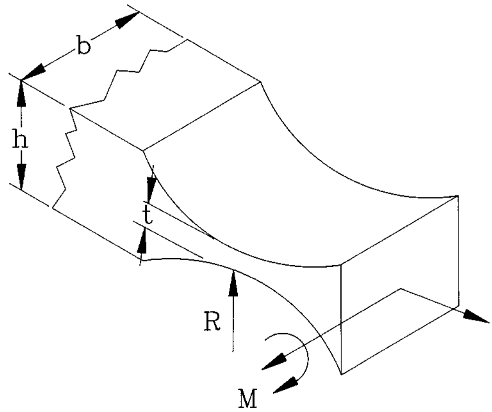


FIGURE 7.34 A typical right-circular flexure for high resonance applications.

where M = the applied bending moment
R = the cutting radius
E = the elastic modulus
b = the width of the flexure
t = the flexure thickness

Actuators for Tilt Mechanisms

The actuators suitable for linear mechanism can also be used in tilt mechanisms. The screws and micrometer adjustments are used in manual mechanisms, and are extensively employed in commercial tilt stages and mirror mounts. Once again, the motorized actuators are large in size and expensive, and are used when real-time and frequent adjustments are required.

Coupling Methods for Tilt Mechanisms

As in linear mechanisms, the rounded tip of the actuator can bear against a flat or a conical surface in tilt mechanisms. The flat surface has high contact stresses, while a conical seat provides an increased contact area to minimize the contact stresses in those tilt mechanisms, which are adjusted frequently or require a higher load capacity.

Preloading Methods for Tilt Mechanisms

The preloading methods employed in linear mechanisms can also be used in tilt mechanisms. These include springs and washers as discussed in Section 7.3. The extension or compression springs can be centered between two adjacent actuators to achieve a uniform preloading. For smaller tilts and higher preloads, stacked Belleville or curved washers can be used in place of springs. The advantages and disadvantages of these preloading methods have already been discussed under linear adjustment mechanisms.

Locking Methods for Tilt Mechanisms

The locking methods for tilt mechanisms are similar to those employed in the linear adjustment mechanisms. These include set screws, jack screws, locknuts, and epoxy. The set screws and locknuts are used for temporary locking in coarse adjustment tilt mechanisms. The epoxy locking is economical and simple in design but is used in those mechanisms which are adjusted at initial assembly and alignment only, and do not require to be disassembled. As in linear mechanisms, a closed-loop control system can be used for motorized actuators to retain the desired tilt adjustment in real-time position control applications.

Examples of Tilt Adjustment Mechanisms

The conceptual designs of a number of tilt adjustment mechanisms are described in this section. As mentioned earlier, these examples are meant as guidelines only and a designer will most likely have to size the components of the mechanism according to the desired performance requirements. Moreover, a number of components from different mechanisms presented here can be combined to optimize the design for the application on hand. In some cases, the design of a single adjustment point is presented. As mentioned earlier, three mutually orthogonal adjustment points are needed to achieve a tilt adjustment along two mutually orthogonal axes. These three adjustment points must be spaced as far apart as possible on the optical mount to improve the angular resolution of tilt adjustment.

A very simple single-axis tilt mechanism for a flat mirror is depicted in Figure 7.35[15] for nonprecision applications. The lower edge of the mirror is rounded and sits in a v-groove in the mount. A single round tip adjustment screw threaded through the mount acts at the center of top edge of the mirror causing the mirror to pivot about its lower edge. A sheet metal spring clip is employed to preload the mirror against the adjustment screw and the v-groove. A locknut is used to retain the adjusted position and to secure the spring clip to the mount. The lateral shift of the mirror can be minimized by making the length of the v-groove approximately the same as the width of the mirror. Moreover, the in-plane movement of a flat mirror can be tolerated in most applications. The negative features of this low-cost simple adjustment mechanism are the special machining features required at the top and bottom edges of the mirror, and a low accuracy of the adjustment. Since the mirror is retained by frictional force only, it is not suitable for shock and vibration environments, where it may shift and lose its alignment.

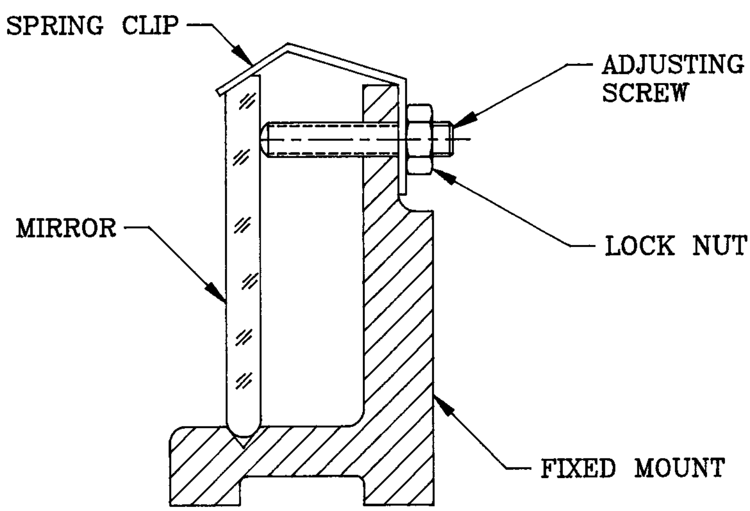


FIGURE 7.35 A low-precision single-axis tilt mechanism for a mirror.

The design of a simple two-axis tilt mechanism, similar to a commercial tilt stage described earlier, has been reported by Walsh.[23] The piano wire acts as a universal pivot between the moving and fixed plates (Figure 7.36). The two plates must be spring loaded against each other by providing two extension springs located between the adjustment screws and the piano wire. Fine threaded screws with round tips can be used to improve the resolution of tilt adjustment. One screw tip sits in a v-groove, while the other screw tip contacts the flat surface of the moving plate. It may be

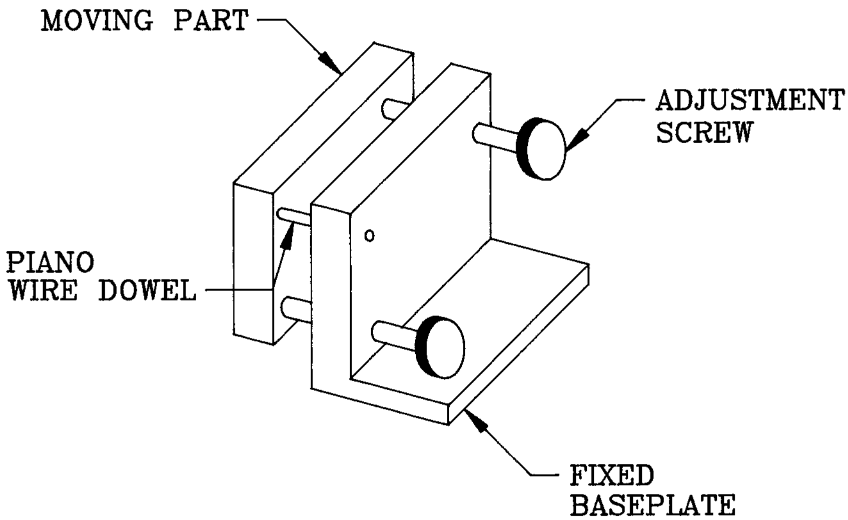


FIGURE 7.36 A simple two-axis tilt mechanism with a universal pivot.

more economical to buy a commercial tilt stage as compared to fabricating this tilt adjustment mechanism.

The design of a high precision single-axis tilt adjustment mechanism has been reported by Nemirovsky.[24] The interface between the tilt platform and the fixed base is through steel balls (Figure 7.37[A]). The top ball on the right side is spring loaded to eliminate backlash. A fine-threaded screw pushes on the lower ball, which is wedged under the middle ball, resulting in a tilt motion about the left-side ball. This mechanism has tilt resolution of the order of a fraction of an arc-second. A number of variations of this design are possible. The balls can be replaced by precision centerless ground shafting with tapered ends. The adjustment screw can have a round tip, thereby eliminating the need for a separate lower ball (Figure 7.37[B]). The screw can also be replaced by a rod with a tapered end to produce a jacking action (Figure 7.37[C]). This mechanism is useful in applications requiring single-axis tilt adjustments of a high accuracy over a limited angular travel.

James and Sternberg[25] have described the design of a simple two-axis tilt mechanism as shown in Figure 7.38. Three mutually orthogonal fine screws with ball tips provide a kinematic interface between the mirror cell and fixed structure. The mirror cell is spring loaded against the screws using a compression spring at the center. After making the tilt adjustment, the adjustment screws are locked by means of locknuts. Care must be taken to ensure that the adjustment is not lost due to excessive locking force. This is a good low cost mechanism for low accuracy applications.

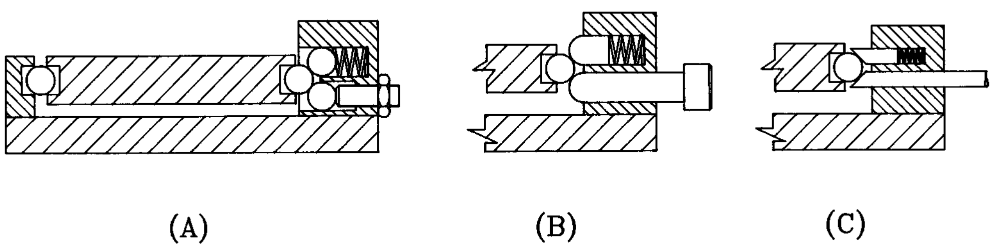


FIGURE 7.37 A high-precision mechanism for small tilt angles using: (A) two spring-loaded steel balls; (B) a round tip screw and a spring-loaded ball; (C) a tapered rod and a spring-loaded ball.

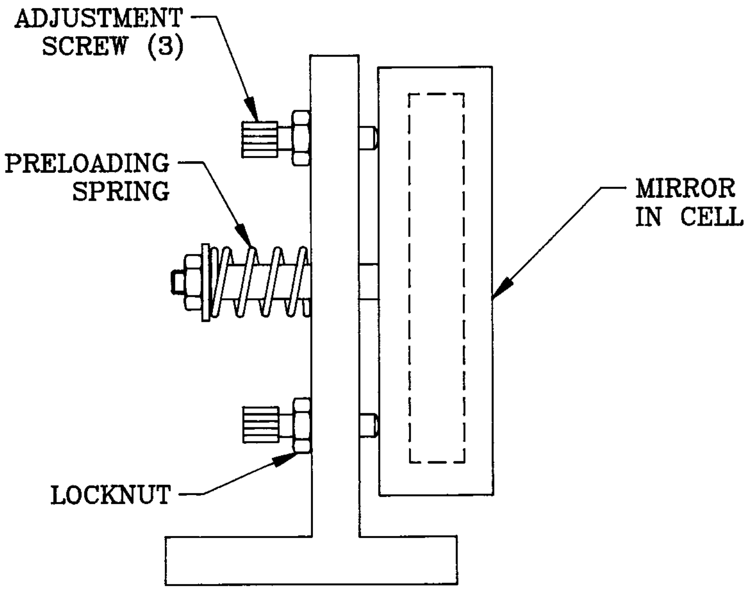


FIGURE 7.38 A low-cost tilt mechanism for a mirror using a single spring at the center for preloading.

The adjustment mechanism shown in Figure 7.39[15] can be designed to achieve very fine tilt adjustments due to a lever arrangement with a high mechanical advantage. It employs two flat blade flexures. The lower flexure is compliant in bending, while the vertical flexure is very stiff in tension. Coarse adjustment is provided by a regular nut threaded to the top end of vertical flexure. A locknut is provided to retain this coarse adjustment. Fine adjustment is made by a spring-loaded screw, which bends the horizontal flexure. By making R/r ratio equal to 10, 5 µm of vertical travel is obtained for every 50 µm of travel of the fine adjustment screw. The cost of this mechanism is comparatively higher due to the number of parts in it.

Another tilt mechanism using flat blade flexures is illustrated in Figure 7.40.[15] The two flexures are riveted or welded together to fix their lower edges. A spring-loaded adjustment screw is threaded into the fixed structure through a clearance hole in the flexures. The top edge of smaller flexure is rigidly attached to the fixed part using two screws and a pin to prevent any slippage. The top edge of the longer flexure is fixed to the tilting part in the same fashion. When the screw is threaded in to bend the flexures, a very small relative motion is produced between the free edges of flexures, thereby causing the moving part to tilt. A very fine adjustment can be achieved for a relatively large travel of the adjusting screw by properly sizing the flexures. This mechanism is rather expensive to fabricate, and is suitable for applications requiring high stiffness and accuracy over a limited travel range.

Figure 7.41 shows the design of a tilt adjustment mechanism reported by Ahmad[16] for a high resonance adjustable mirror mount. The mirror is suspended in its cell by three tangent bars with a pair of circular cross-flexures at each end. One end of each bar is secured to the invar buttons bonded directly to the mirror, while the other end is attached to the fixed structure. These tangent bars are very stiff in tension and compression, while they are relatively quite compliant in bending normal to the plane of the mirror. A small differential micrometer pushes on each invar button through a compliant flexure to eliminate lateral loads and misalignments. Each invar button is spring loaded against the tip of the micrometers. A high resolution tilt adjustment can be made by moving one differential micrometer at a time. This expensive tilt adjustment mechanism is recommended for high resonance precision applications.

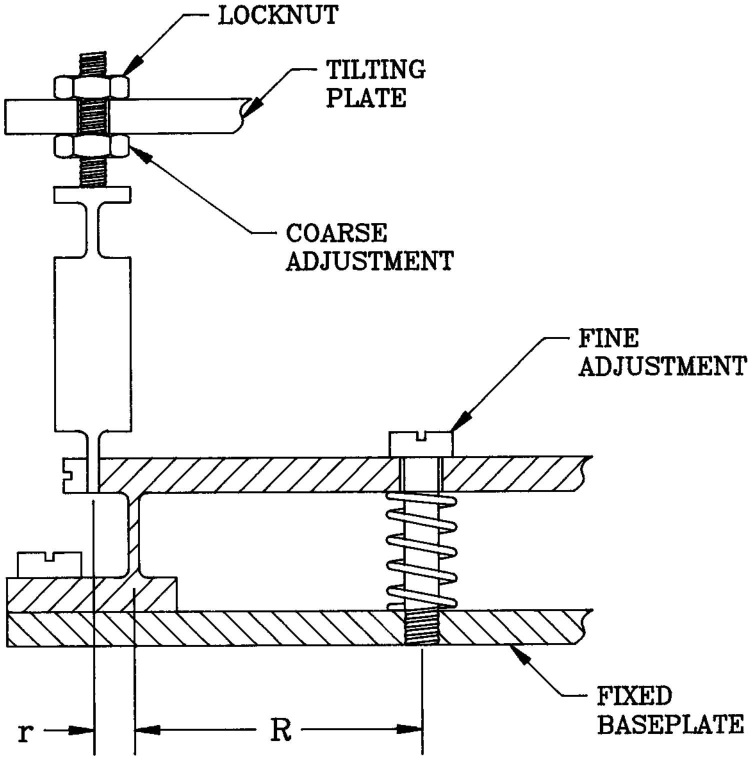


FIGURE 7.39 A tilt mechanism with coarse and fine adjustments using single blade flexures.

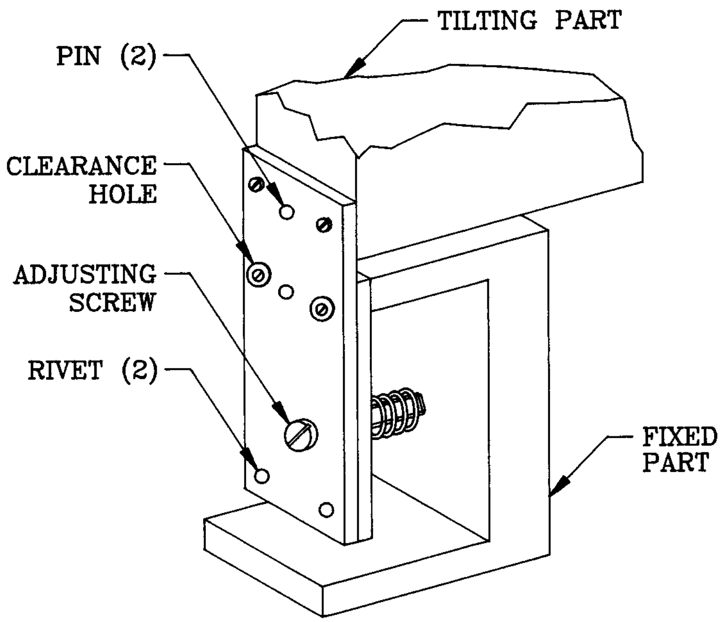


FIGURE 7.40 A tilt mechanism with a fine adjustment capability over small angles.

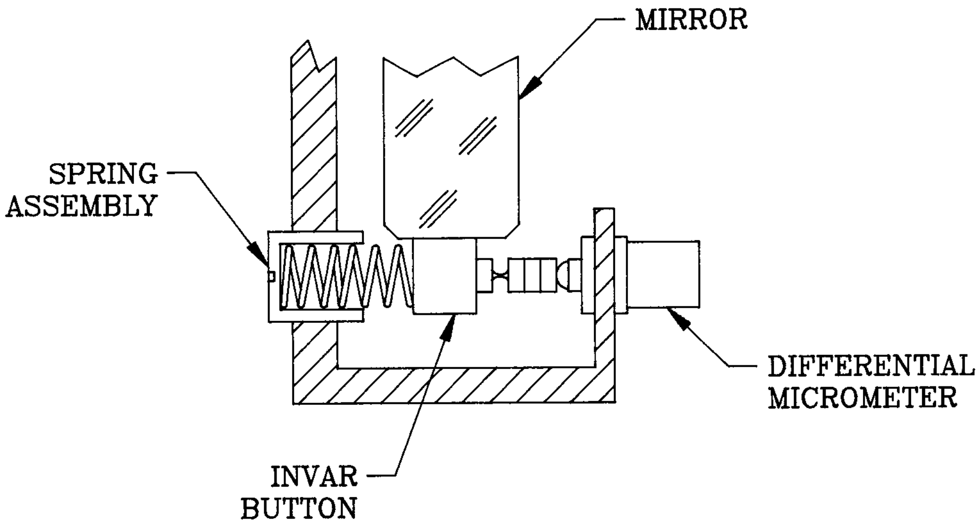


FIGURE 7.41 A tilt mechanism suitable for high resonance adjustable mirror mounts.

7.5 Rotary Adjustment Mechanisms

General Description

Rotary adjustment mechanisms are not as commonly used in optical systems as the linear and tilt adjustment mechanisms. For flat optics such as fold mirrors, windows, filters, and prisms, an in-plane rotation does not produce any change in alignment, and hence the system performance is unaffected. Similarly, for spherical optics such as lenses and mirrors, any rotation about the optical axis does not produce or correct any optical aberration. For off-axis and aspheric optics, normally linear and tilt adjustment mechanisms are employed to align the optical systems. In general, rotary mechanisms are used in scanning applications such as bar-code scanners in supermarkets and scanning telescopes and cameras for surveillance and remote monitoring, or for optical beam steering.

A number of choices are available to a designer for selecting the components of a rotary mechanism depending on the performance requirements such as frequency, range, and resolution of angular travel, shock, load capacity, cost, and size of the mechanism. A number of components that are used in linear mechanisms can also be used in rotary mechanisms, either in the same configuration or in their rotary version. These design choices along with some design guidelines are discussed in this section, followed by examples of some simple rotary mechanisms.

Interfaces for Rotary Mechanisms

The choice of a suitable interface between the rotating part and the fixed structure depends on such design criteria as the range and frequency of adjustment, shock, load capacity, cost, and size requirements. As in the case of linear mechanisms, rotary versions of the bearings discussed in Section 7.3 are commonly used to interface the rotating part to the fixed part. Spherical and journal bearings are only used for light duty cycles because of the friction and wear problems. These bearings are compact in size and are inexpensive as compared to rotary ball, roller, and air bearings, which are used in those applications where heavy loads are adjusted frequently or require continuous rotation.

A high speed rotary mechanism using a ball bearing has been described by Weinreb.[26] Figure 7.42 shows the radiometer spindle assembly for TIROS II meteorological satellite. The chopper mirror is mounted to a shaft supported by a ball bearing and is driven by a low power motor at a speed of 2750 rpm. Since this mechanism operates in space, lubrication of bearings is critical to minimize the wear for a long life and to maintain the alignment of the mirror. The ball bearing in this application is constantly lubricated in space by employing a lubricant reservoir of oil-impregnated sintered nylon.

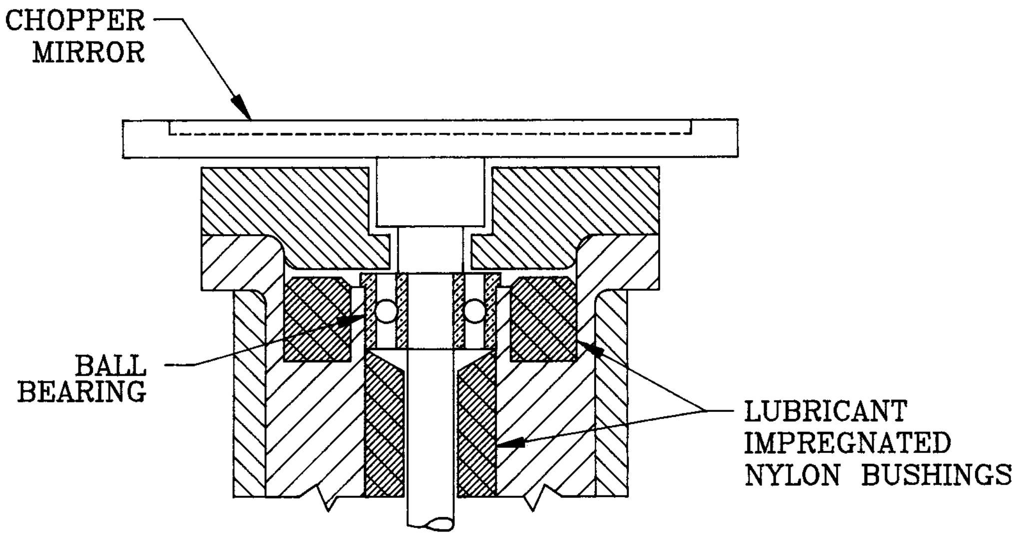


FIGURE 7.42 A high-speed rotary mechanism using ball bearings from TIROS II radiometer.

Rotary air bearings offer high stiffness and load capacity, accuracy, and cleanliness. Motorized air bearings are frictionless and use a thin film of clean dry air at a pressure of 80 to 100 psig. One such motorized air bearing by Dover Instrument Corporation is shown in Figure 7.43.[27] The bearing assembly consists of a load-carrying section, a spindle housing, and a motor housing. The air-bearing part consists of a shaft and two thrust plates. The shaft, thrust plates, and the inside surfaces of the housing are machined flat and cylindrical to tolerances of 10 micro-in. The air is fed through jeweled orifice restrictors into the clearance between shaft and the housing. The clearance between rotating elements and spindle housing is of the order of 5 to 10 μm. When compressed air passes through this clearance, a positive film pressure is created around the rotating shaft. This film of high pressure air acts like a very stiff spring to prevent any mechanical contact between the rotating parts and the housing, thereby creating a zero-friction condition. This compact bearing assembly comes with a built-in motor and encoder. These types of bearings are ideal for high speed applications requiring vibration-free rotation. The main disadvantage is that these bearings needed a supply of pressure-regulated clean air, which adds to the overall cost due to the equipment needed for air supply.

For small angular adjustments, a flexural pivot (Bendix type) offers the advantages of friction and backlash-free angular adjustment. These commercially available pivots are small in size and have low hysteresis. The application of these pivots in a mirror mount has been described by Rundle[28] in detail.

Actuators for Rotary Mechanisms

The actuators used in rotary mechanisms are very similar to those used in linear mechanisms discussed in Section 7.3. Screws are used in low cost applications for small angular adjustments. Micrometers and differential micrometers are used when a more precise adjustment with a readout

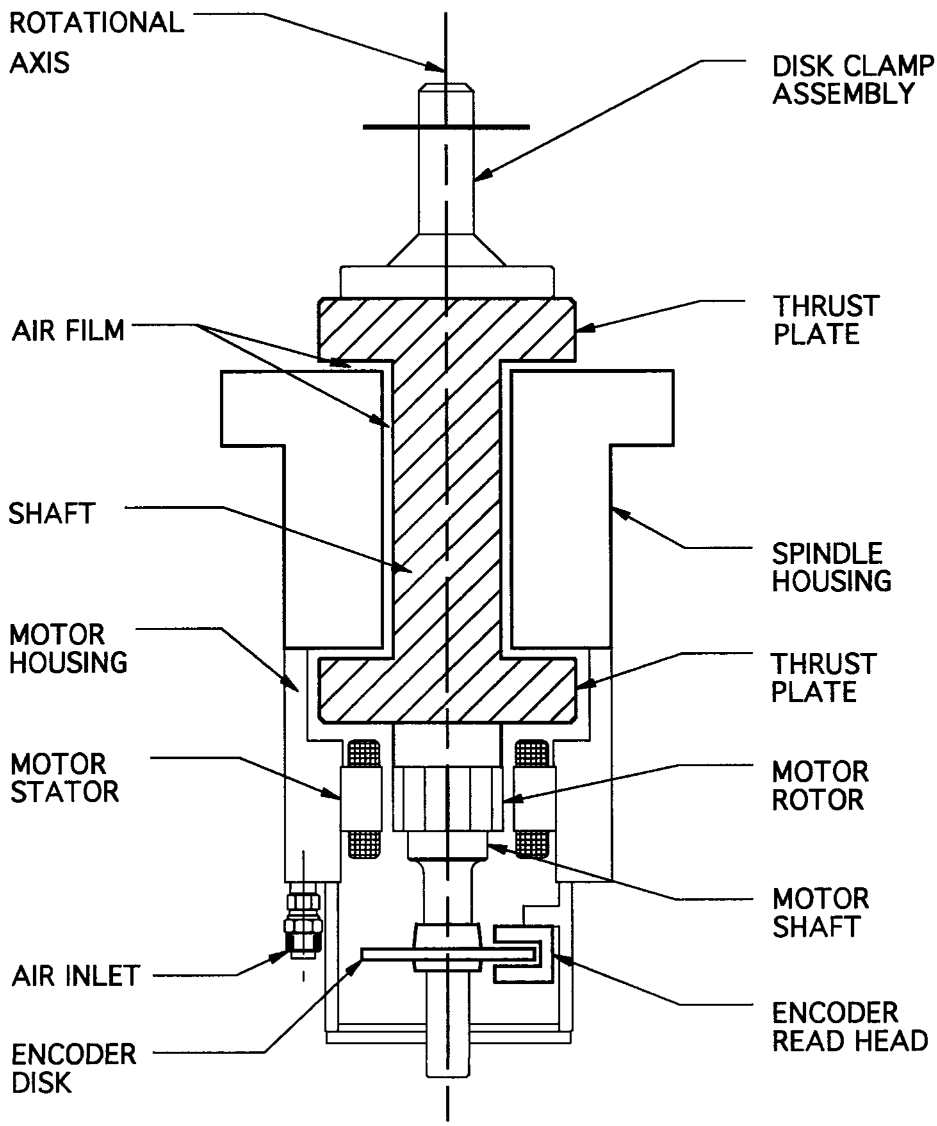


FIGURE 7.43 (A) The cross section of a high speed motorized rotary air bearing. (Courtesy of Dover Instrument Corporation, Westboro, MA.)

is required. Commercially available rotary stages use both types of micrometers extensively. A typical commercial rotary stage is illustrated in Figure 7.44. It provides full 360° rotation about a vertical axis, and typically runs on a ball bearing. The angular resolution depends on the type of actuator used. A thumb screw is used for locking the adjusted position. Once again, the motorized actuators such as DC and stepper motors are used in applications requiring large and frequent angular travels or for real-time adjustments.

Coupling Methods for Rotary Mechanisms

Some coupling methods for linear mechanisms discussed in Section 7.3 can also be used in rotary mechanisms. For small angular travels, the round tip of an actuator can bear against a flat surface, cone, or a spherical socket. The advantages and disadvantages of these arrangements have already been discussed earlier. These interfaces can only be used for very small angular adjustments, since

FIGURE 7.43 (B) Picture of a high speed motorized rotary air bearing. (Courtesy of Dover Instrument Corporation, Westboro, MA.)

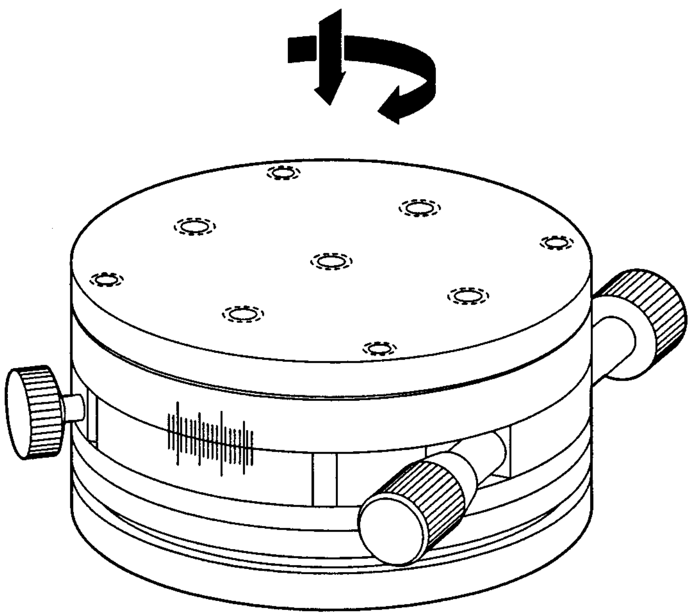


FIGURE 7.44 A typical single-axis commercial rotary stage. (Courtesy of Newport Corporation, Irvive, CA.)

the tip of the actuator slides relative to the rotating part and cannot maintain a good contact with the rotating part for larger angles.

For large, frequent, or constant angular motion, an actuator can be coupled to the rotating part through a flexible type of coupling. The choice of coupling type depends on the radial load, torque capacity, life expectancy, and maintenance.[29] Flexible-type couplings can tolerate relatively large misalignment between the axes of rotation of the actuator and the moving part. The performance of the systems using a flexible coupling depends on the inertia, backlash, friction, and linearity of the coupling used. It should be noted that couplings designed for motion control applications may not be suitable for power transmission and vice versa. The couplings illustrated in Figure 7.45 are suitable for rotary mechanisms and exhibit low inertia, zero backlash, and near-constant velocity. The principle disadvantages of couplings are their relatively large size, weight, and cost.

FIGURE 7.45 Some commonly used flexible couplings for rotary motion. (Courtesy of Renbrandt, Inc., Boston, MA.)

Sometimes linear actuators are used in rotary mechanisms because of their lower cost and compact size. In such cases the linear motion of the actuator is converted to a rotary movement through a worm and gear or a rack and pinion arrangement. In these mechanisms, the part to be rotated is attached to the rotating gear while the linear actuator moves the rack. These mechanisms are expensive because of their mechanical complexity and exhibit backlash, if not designed properly.

Preloading Methods for Rotary Mechanisms

The selection of a proper preloading method to obtain a backlash-free rotary adjustment is based on the same design factors that are discussed in Section 7.3 for the linear mechanisms. The tension or compression springs are extensively used for preloading purposes because of their low cost, and also because these can be used over a fairly large angular range. The Belleville washers are used for high preloads over small adjustment ranges. For larger angular rotations, a torsion spring can be used for preloading the rotating part. One end of the spring is attached to the rotating part, while the other end is attached to the fixed structure.

Locking Methods for Rotary Mechanisms

The design factors affecting the choice of suitable locking methods for rotary mechanisms are similar to those in the linear mechanisms as already discussed in Section 7.3. These factors include travel range and frequency, size and cost, and disassembly requirements.

The locking can be accomplished by set screws, clamps, epoxy, and locknuts. The relative advantages and disadvantages of all these locking methods have already been discussed. Epoxy locking is inexpensive, but it is more or less permanent, because the parts cannot be disassembled without a risk of damage. The other locking methods are nonpermanent and less expensive, but may introduce high stresses due to clamping force in the components being locked.

Examples of Rotary Adjustment Mechanisms

Tuttle[10] has suggested the designs of a number of simple rotary mechanisms for angular positioning as shown in Figures 7.46 to 7.49. For small angular adjustments, the lever and shaft arrangements shown in Figures 7.46 and 7.47 are simple and cost effective. Two opposing tangent screws are used in the mechanism shown in Figure 7.46. The round tip of the screw pushes on a lever that rotates on a precision journal or ball bearing through small angles, while the opposing screw is moved out. The screws are locked in place by locknuts to hold the adjusted position. This low precision, low cost mechanism is suitable for infrequent adjustments over small angular travels.

In the mechanism shown in Figure 7.47, the resolution of adjustment can be improved by using the fine pitch threads. In this mechanism, the threaded sleeve is spring-loaded against the stationary

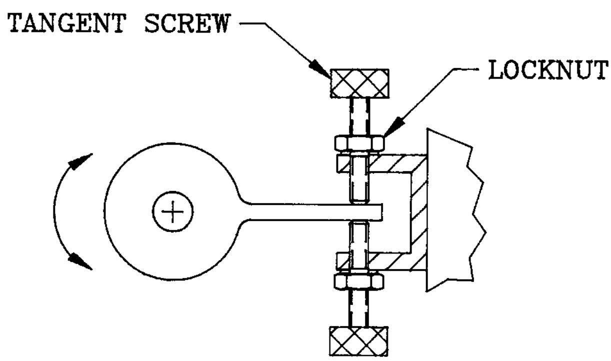


FIGURE 7.46 A simple rotary mechanism using tangent screws for small angular travels.

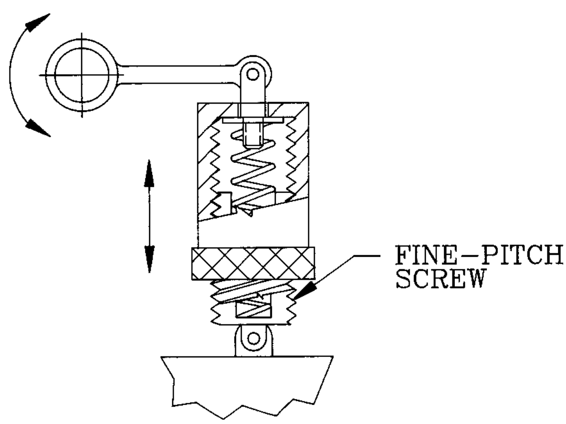


FIGURE 7.47 A rotary mechanism using a fine-pitch screw for a limited travel range.

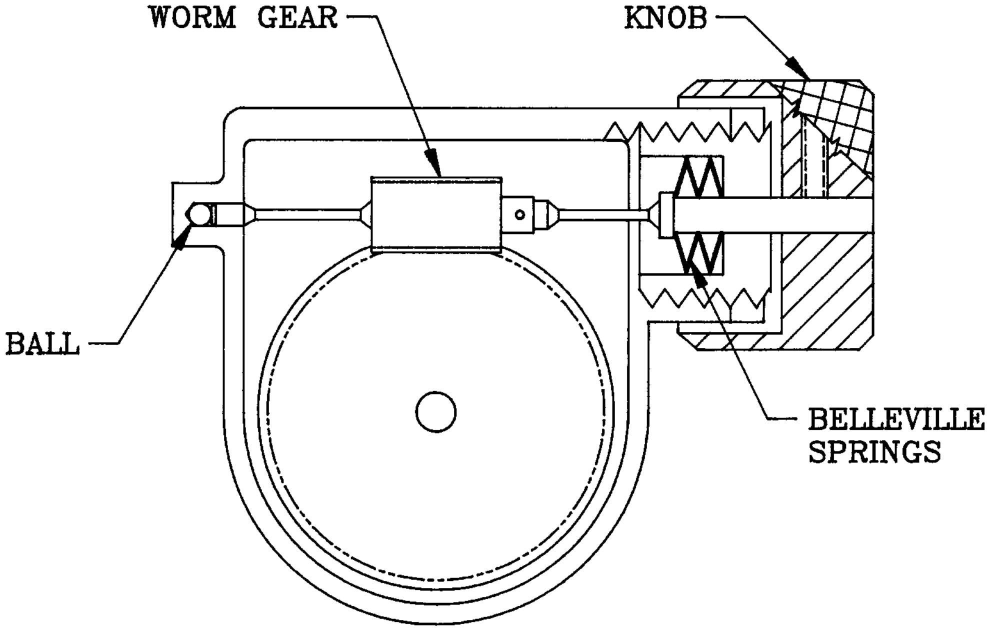


FIGURE 7.48 A rotary mechanism using a worm and gear for full 360° rotation.

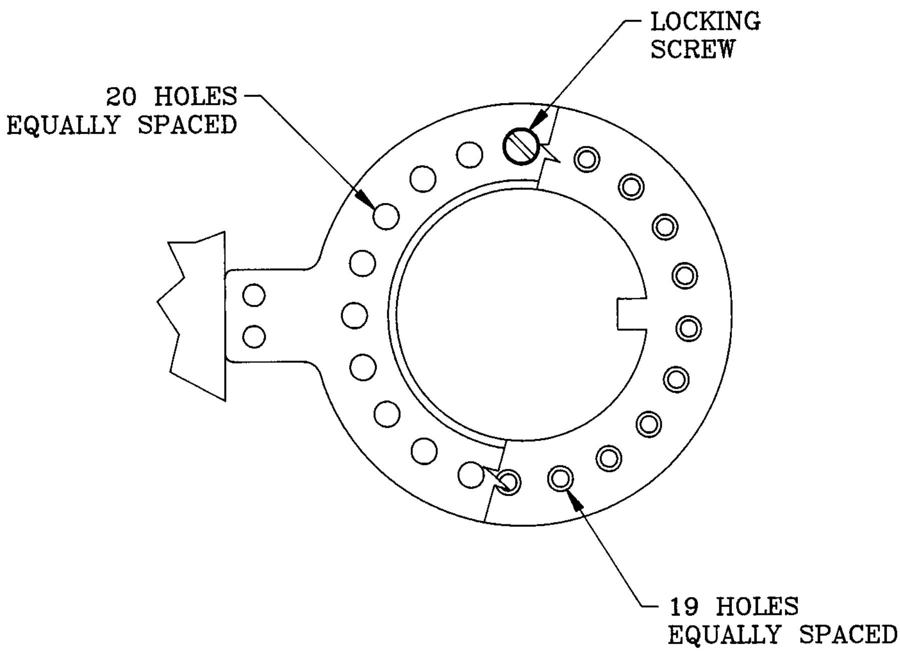


FIGURE 7.49 A very simple low-cost rotary mechanism using two concentric rings with fixed adjustment steps.

half to eliminate any backlash. When the sleeve is rotated in either direction, a change in the length of the screw produces a small angular rotation about the center of rotation. The angular resolution of this mechanism can be further improved by incorporating a differential screw. This mechanism is also suitable for applications requiring infrequent adjustments over small angles.

A worm and gear-type mechanism illustrated in Figure 7.48 provides an angular adjustment over full 360°. A worm mounted on an axially loaded shaft engages with a gear rotating on a bearing. The worm shaft with a reduced diameter is used intentionally to reduce its bending stiffness. The shaft is loaded axially with Belleville spring washers against a ball, which acts as a thrust bearing at the end of the shaft. Due to this spring load, the shaft operates in a slightly deflected shape to ensure a positive contact between the worm and gear. A knob attached to the shaft is rotated to produce the angular rotation of the gear. The angular resolution of this mechanism depends on the gear ratio selected. This mechanism is more expensive due to its mechanical complexity, but it does provide angular adjustments over full 360°, as stated earlier.

A very simple mechanism for one-time coarse angular adjustment is depicted in Figure 7.49. It consists of two concentric rings with a number of same-diameter holes on bolt circles of equal diameters. One of the rings has one hole less than the other ring. The locking is accomplished by lining up one set of holes in both rings, and then inserting a screw or pin through both rings. The number of holes n determines the angular adjustment step between any two adjacent locking positions, and is given by $360/n(n-1)$.

Figure 7.50 illustrates a rotary mechanism using a spherical bearing. The mirror cell has an integral hemispherical ball which sits in a spherical seat in the fixed structure. This arrangement not only allows the mirror to be rotated about its optical axis, but also allows it to be tilted. This mechanism is spring loaded at the center to hold the adjustment. This low precision mechanism has a high friction and is therefore suitable for infrequent angular adjustment.

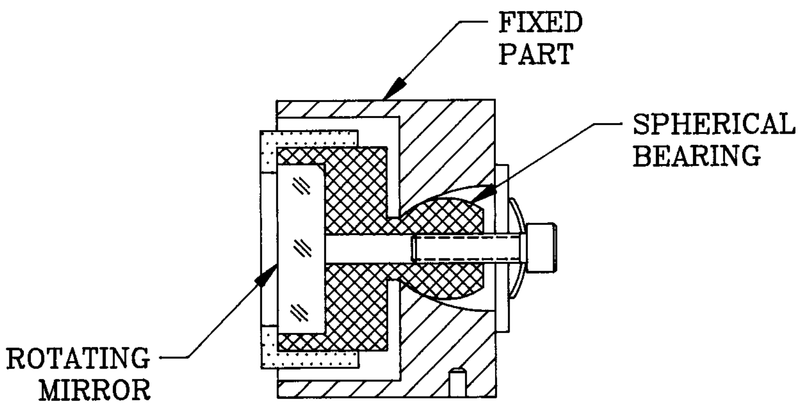


FIGURE 7.50 A spherical bearing provides a full 360° rotation in this rotary mechanism for a mirror.

7.6 Design Guidelines for Adjustment Mechanisms

An adjustable optical mount is inherently more complex, expensive, and less stable than a comparable fixed type of mount. Therefore, a careful trade-off design study must be conducted to weigh the benefits of an adjustable optical mount against the potential stability and cost drawbacks. A good summary of how and when adjustment mechanisms must be incorporated in optical systems has been presented by Ahmad[14] and Vukobratovich.[30]

The decision to use adjustable optical mounts is dictated by the sensitivity analysis performed by an optical designer. This analysis defines the assembly tolerances for an optical system to achieve its image resolution and quality requirements. For systems with high image resolution requirements, usually the alignment tolerances of the optical elements exceed the practical fabrication tolerances for the optical elements and the associated mechanical hardware. This problem is further compounded by the stack-up of machining tolerances and inspection uncertainties when an optical system has a large number of mirrors and lenses. In such applications, it is economical and time saving to manufacture the optical and mechanical parts to rather loose tolerances and provide adjustments for a few sensitive optical elements. These elements are then adjusted to compensate for the errors and uncertainties in the positions of other optical elements. Therefore, a rigorous optical sensitivity analysis must be performed to identify the optical elements that have the strongest effect on image quality of the system.

The required accuracy of adjustment is usually dictated by the type of an optical element and performance requirements of the optical system. Table 7.8 by Vukobratovich[30] gives general guidelines for the type of adjustment required and its sensitivity for different types of optical elements. Once the types of adjustments required for an optical element have been determined, the next step is to design an adjustable mount to provide the desired adjustment range and resolution. The glass optical elements are normally assembled in a metal frame, and an adjustment mechanism is provided between the frame and the fixed mounting structure. This way, a direct contact between the actuator and the glass element is avoided, and the element is protected from high contact stresses at the tip of the actuator. Providing an adjustment range, which is longer than that determined by the optical sensitivity analysis, can be very risky. Such mounts can be assembled far away from their required nominal position, and a lot of time may be wasted during alignment to obtain any image at all. In such cases, it is better to design the mount so that it can be shimmed at assembly, in case the range of adjustment is found to be inadequate. After an optical system has been aligned, the adjustable mounts must be positively locked to prevent any misalignment due to accidental tweaking or any drifts as a result of shock and vibrations. The adjustments, which are used only at initial assembly, can be designed to have lockable mounts with removable actuators. This approach not only saves on the cost of actuators, but also eliminates the risk of accidental misadjustment later on.

TABLE 7.8 Types of Adjustments Required and Their Sensitivity

Type of Optical Element	Adjustment Sensitivity	Type of Adjustment Required
Flat window	Very low	Tilt
Field lens	Low	Tilt and decenter
Flat mirror	Medium	Tilt
Prism	Medium	Tilt
Objective lens	High	Tilt, decenter, and despace
Relay lens	High	Tilt, decenter, and despace
Curved mirror	Very high	Tilt and despace

The adjustable mounts should be designed to tilt or rotate about the principal points to avoid cross-talk between axial and tilt adjustments. The cross-coupling of adjustments can be very frustrating because several iterations of tilt and axial adjustments may be needed to achieve the alignment. The adjustable mirror mounts must be designed to tilt a mirror about its vertex to avoid an unwanted image shift. The adjustment points in a tilt mechanism should be positioned in a mutually orthogonal pattern relative to the axis of the optical element. Such mounts are easy to adjust and produce predictable movements.

The adjustment mechanisms must be designed to have a large mechanical advantage such that a large axial movement or rotation of the actuator will incrementally move the optical element. This design feature can also save valuable alignment time because there is a less likelihood of accidentally overshooting the optimum position. If feasible, the mechanisms must be designed to have a coarse as well as a fine adjustment. The coarse alignment can be quickly achieved by using the coarse adjustment, while the fine part of the adjustment is only used to optimize the quality of the image.

A number of translation, rotary and tilt stages, mirror, lens, and gimbal mounts are available commercially. These mounts are economical, precise, and quite rugged, and are very suitable for prototypes and laboratory setups. Their main disadvantage is their bulky size and weight, which makes their use impractical in the systems with several optical elements that are packaged together tightly. If weight and size are not a problem in an application, it is far more economical and time saving to use commercial mounts rather than designing and fabricating the custom adjustable mounts. Locking can be added to these commercial mounts, if needed.

While the adjustment mechanisms offer several advantages, the disadvantages of the adjustable mounts must not be overlooked, and provisions must be made in their design to minimize their negative effects on the optical system. First of all, the number of adjustable optical elements in a system must be kept to an absolute minimum. This not only saves on the fabrication cost, but also maximizes the long-term stability of the system. The adjustable mounts are less rigid and less stable than the fixed type of mounts and therefore experience a drift with time. The adjustments often induce nonlinear and unpredictable effects in optical systems. The adjustable mounts are mechanically weak and are more susceptible to drifts due to shock loads, vibrations, and temperature variations.

7.7 Summary

The adjustment mechanisms play an important role in the integration and alignment of sophisticated optical systems which have tight positioning tolerances. By incorporating these mechanisms in the optomechanical design, it becomes feasible and economical to produce and assemble such optical systems to very high alignment accuracies, which in turn greatly enhances the image quality of these systems. The design guidelines for three basic types of adjustment mechanisms have been presented. The design choices for various components that make up the linear, rotary, and tilt adjustment mechanisms have been listed. The advantages and disadvantages of these choices have been presented to help a designer in choosing the most suitable parts of the adjustment mechanism for a particular application.

The designs of a number of sample linear, tilt, and rotary mechanisms have also been described in this chapter. These designs cover low cost and low resolution applications and the precision mechanisms with a fine resolution for high performance applications. Some mechanisms presented here may be employed without any modification to satisfy a particular need, while in other cases the components with desirable features can be selected from different mechanisms to design a custom mechanism with optimum features for a particular application. The size and shape of the components in a mechanism are dictated by the space constraints. Therefore, in most applications it may be necessary to select or design the components to satisfy the space requirements.

The design guidelines for how and when adjustment mechanisms must be incorporated into optical systems have also been discussed. By following these simple design guidelines, the performance and image quality of a system can often be optimized at a minimum cost. The intelligent use of adjustment mechanisms can result in lower fabrication costs and a considerable reduction in the time and effort involved in alignment of precision optical systems.

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