Adjustments and flexures

Optical instruments and systems often require the ability to make small mechanical adjustments.

The design must be carefully considered to

- Provide resolution for the adjustment in the desired degree of freedom
- Fully constrain all other degrees of freedom

Other important considerations

- Total range of adjustment
  - Sometimes, you require both coarse and fine adjustments
- How often the adjustment must be made
- Required stability <u>for all degrees of freedom</u>
- Required stiffness (from static or dynamic requirement)

Constraints

- All degrees of freedom must be constrained
- As one degree of freedom is moving, the others must remain constrained
  - Kinematics (balls on hard plane, V, or cone)
  - Pivot (suffers from friction)
  - Flexures
    - No friction
    - Limited range
    - Use geometry, materials to provide compliance in adjustment DoF, and stiffness in others

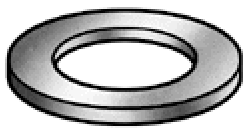
Classes of adjustments

- Shims and spacers
  - Most stable
  - Used for one-time adjustment
  - Need to have a good way to determine the necessary spacer thickness
  - Details of the hardware are critical (follow the load path)
- Push-pull screws
  - Not as stable as shims(can add locking jam nuts.)
  - Resolution limited by thread pitch, friction
  - For one time adjustments, use potting epoxy to make it permanent
- Push against a spring load
  - Can be kinematic
  - Most common for small tilt stages for fold mirrors
  - Details of point loads must be considered
  - Preload must not be exceeded in dynamic environment
  - For one time adjustments, use potting epoxy to make it permanent

Shims and Shim Stock

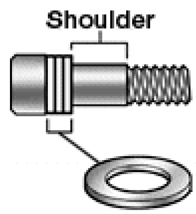
1420 products match your selections

Shim Type



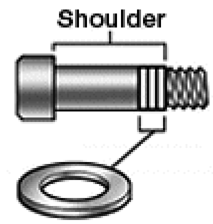
Standard

Specially designed precision washers that align, level and adjust parts in a wide variety of applications.



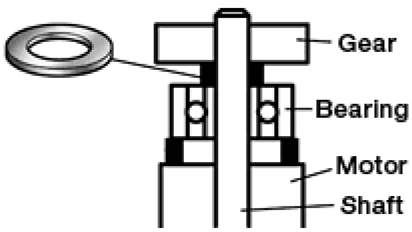
Shortening Shims for Shoulder Screws

These shims fit the shoulder diameter to shorten screw length.



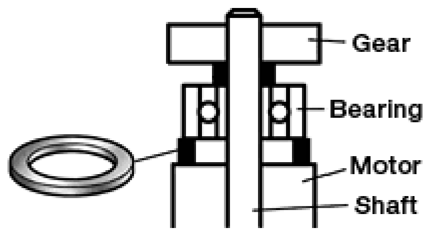
Lengthening Shims for Shoulder Screws

These tight-fitting shims thread onto the screw to extend shoulder length.



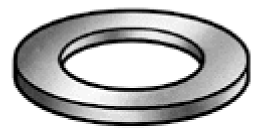
Shaft

Also known as inner-race shaft spacers or washers, they fit snug to shafts and are used for spacing between bearings and gears.



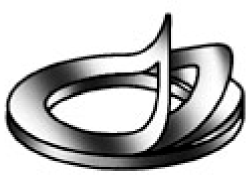
Bearing

These shims are also called outer rim spacers or washers and are designed for spacing between motors and bearings.



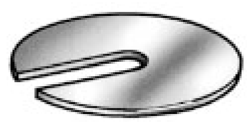
Die Punch Shims

Place these shims under resharpened dies to restore their original height and extend the life of the dies.



Laminated Peel-Away

Custom fit the shim right on your job site. Just peel off the extraordinarily thin laminated layers to get the thickness you need.



Slotted

All have an extra-wide bearing surface.

Color-Coded Shims with Holes

Designed for use with housings and cases for bearings, gears, and pumps, these shims adjust clearance on rotating equipment. Holes allow you to fasten the shim to a housing or flange.

Leveling wedges

Push-pull screws

- Provide large range of motion
- Can achieve fine resolution

  Limited by thread pitch, interface details
- Self locking – jam nuts or epoxy
- Can causes stress, distortion

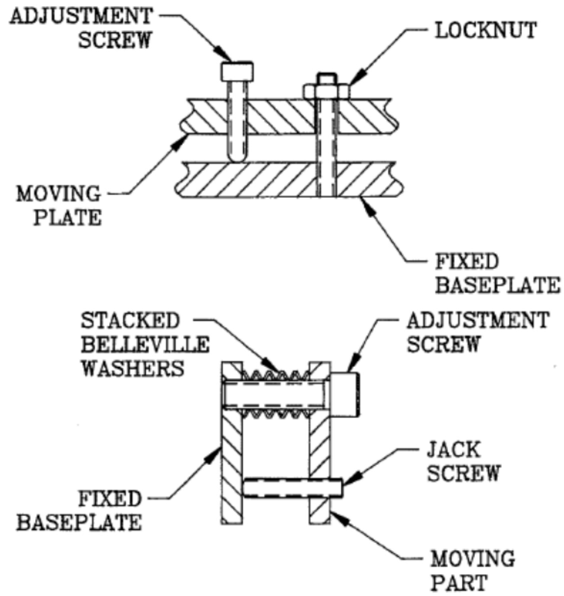
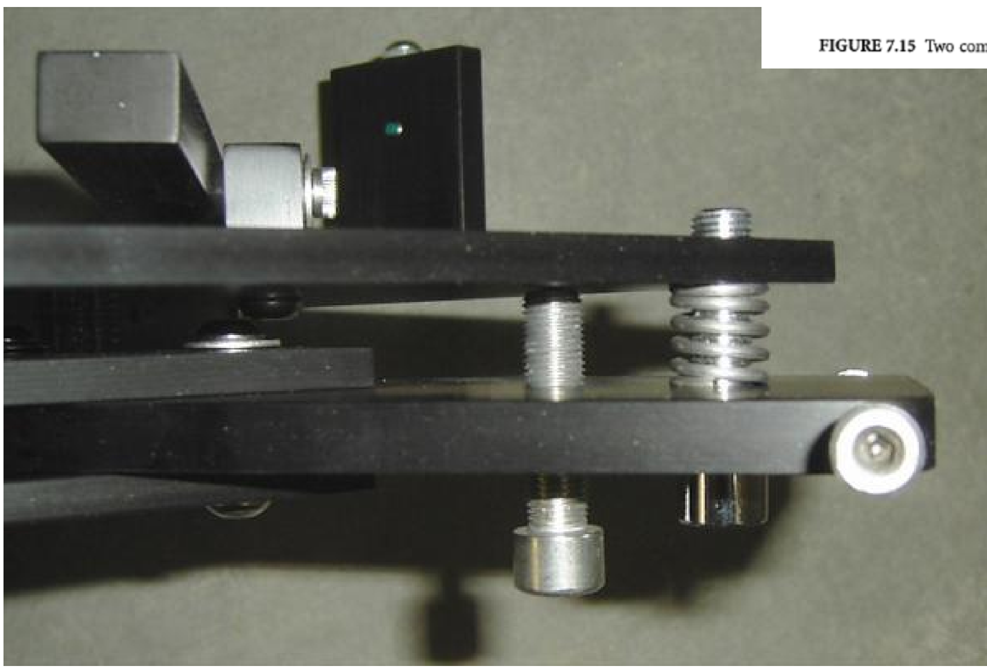


FIGURE 7.15 Two commonly used locking methods. (a) Locknut; (b) jack screw.

Danger

This provides only 1 degree of freedom controlled constraint.

It weakly constrains the lateral dimensions



Quality of adjustment with screws

Thread pitch: 80 pitch screw (0.3 mm/thread) gives about 1 µm resolution *IF*

- quality of threads is good
- preload is appropriate (< 5 lb)
- end is screw is well defined (ideal is a ball)
- surface being pushed on is hard and flat

Resolution scales approximately with thread spacing

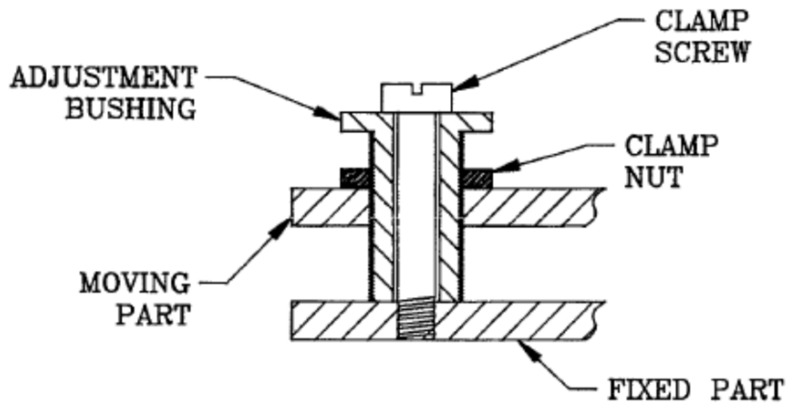


FIGURE 7.22 A linear mechanism with a threaded bushing and clamp screw suitable for applications requiring disassembly.

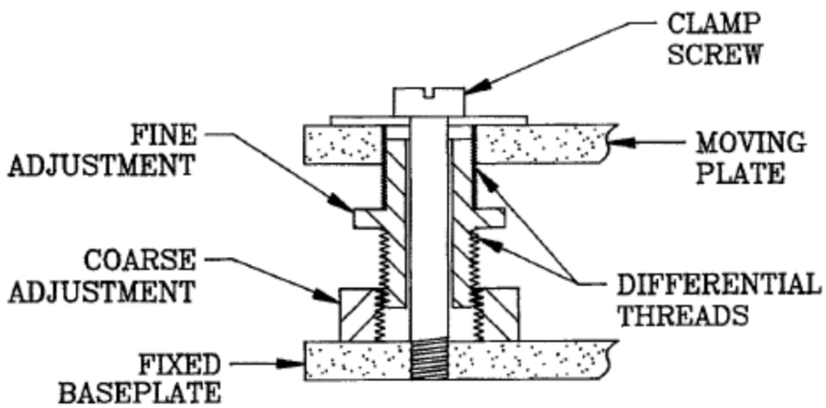
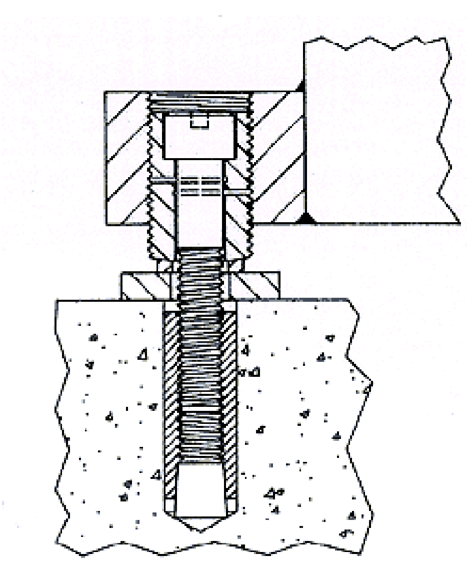


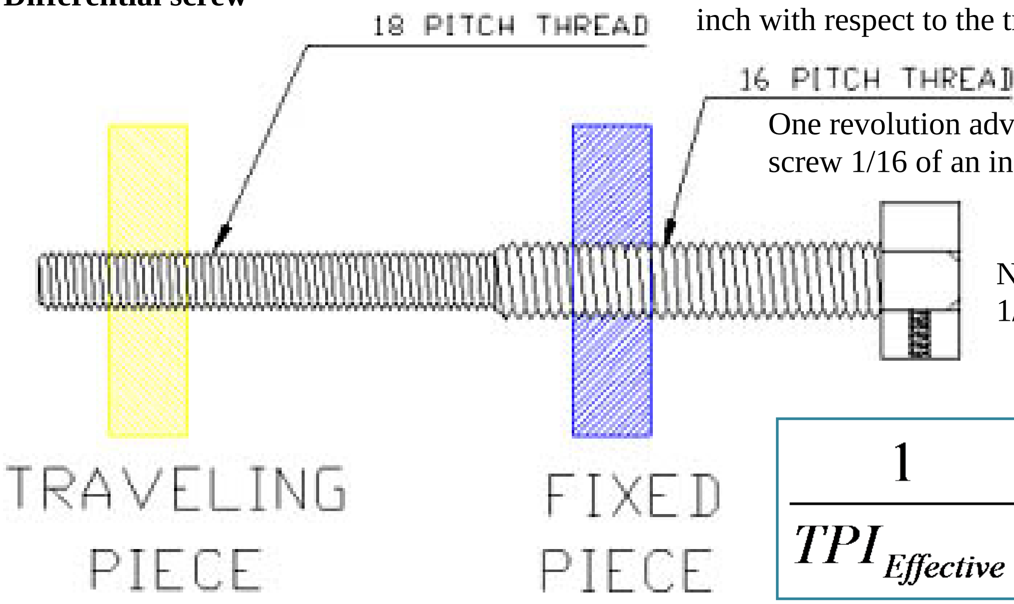
FIGURE 7.23 A linear mechanism with a bushing using differential threads for finer resolution.

Micposi
http://www.harbingerengineering.com/



Differential screw

One revolution advances the screw 1/18 of an inch with respect to the traveling piece



One revolution advances the screw 1/16 of an inch

Net motion per rotation is 1/16 – 1/18 = 0.007” or 1/144

$$\boxed{\frac{1}{TPI_{Effective}} = \frac{1}{TPI_1} - \frac{1}{TPI_2}}$$

Newport product:

DM-13 Series Differential Micrometer

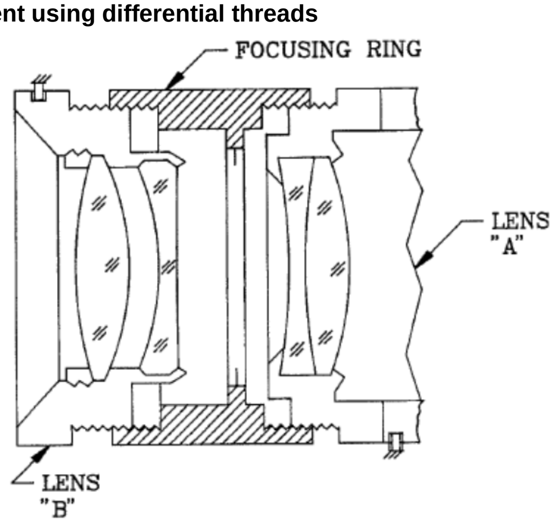
13 mm coarse travel with 0.07 µm sensitivity

Graduated increments of 0.5 µm

Accuracy better than 1%

Direct upgrade for SM Series Micrometers

Focus adjustment using differential threads



Centration, rotation of A wrt B must be constrained

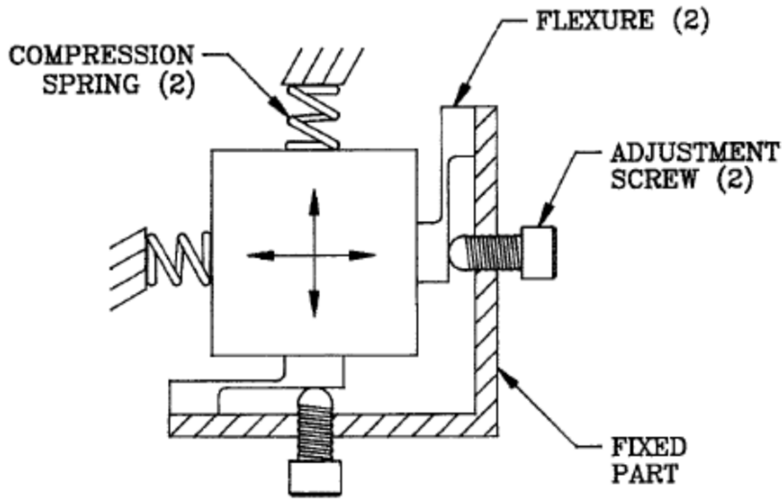
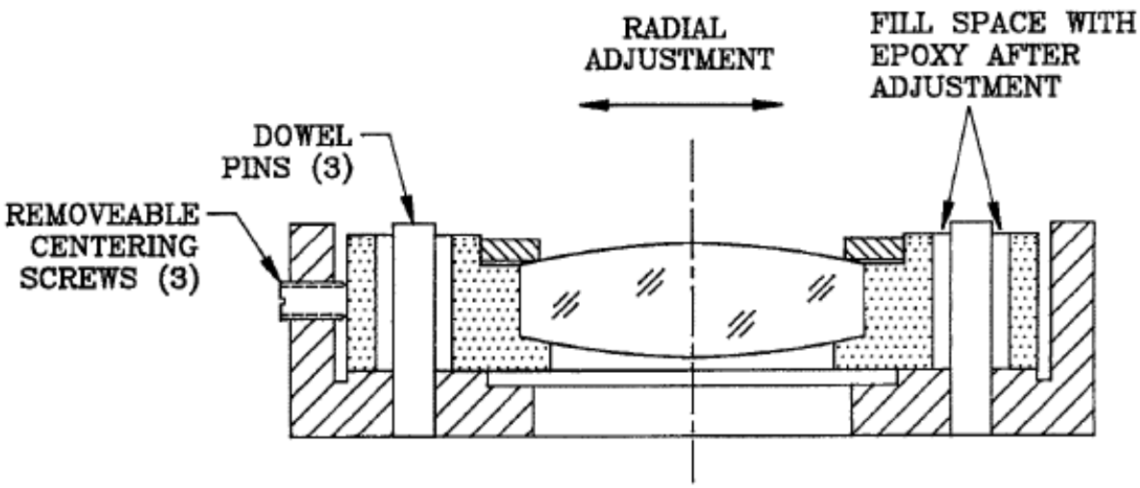
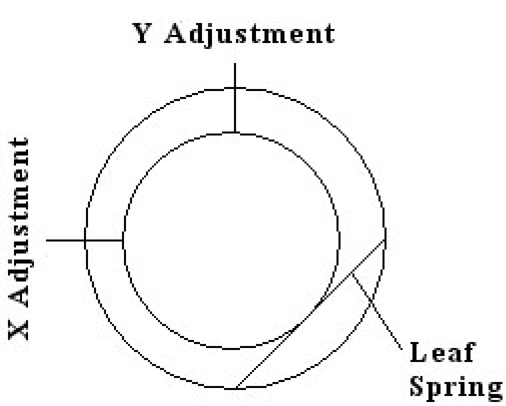
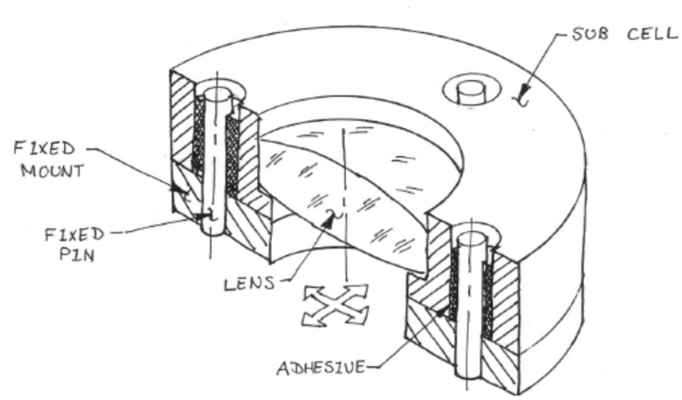
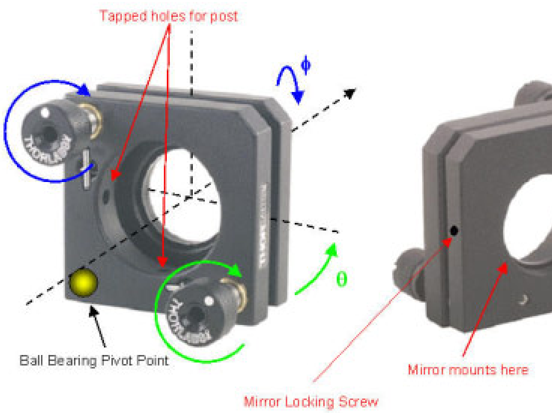


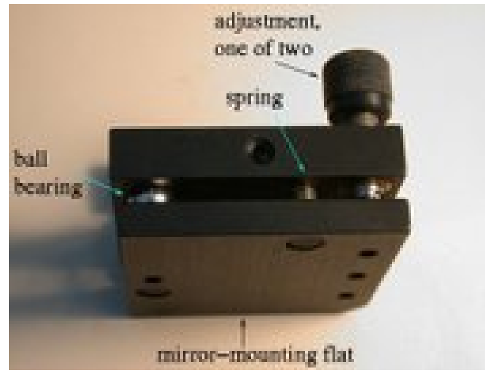
FIGURE 7.19 A two-axis linear mechanism using flexures and screw actuators.











Kinematic tip/tilt mounts

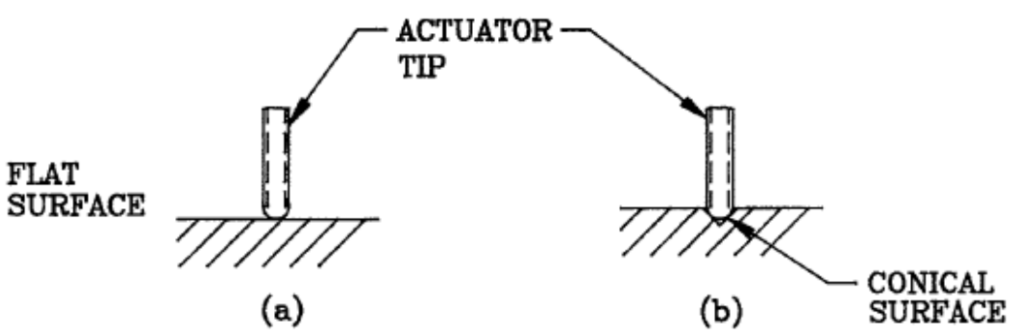


FIGURE 7.11 Two types of common interfaces for the round tip of an actuator. (a) Point contact with a flat surface; (b) line contact with a cone.

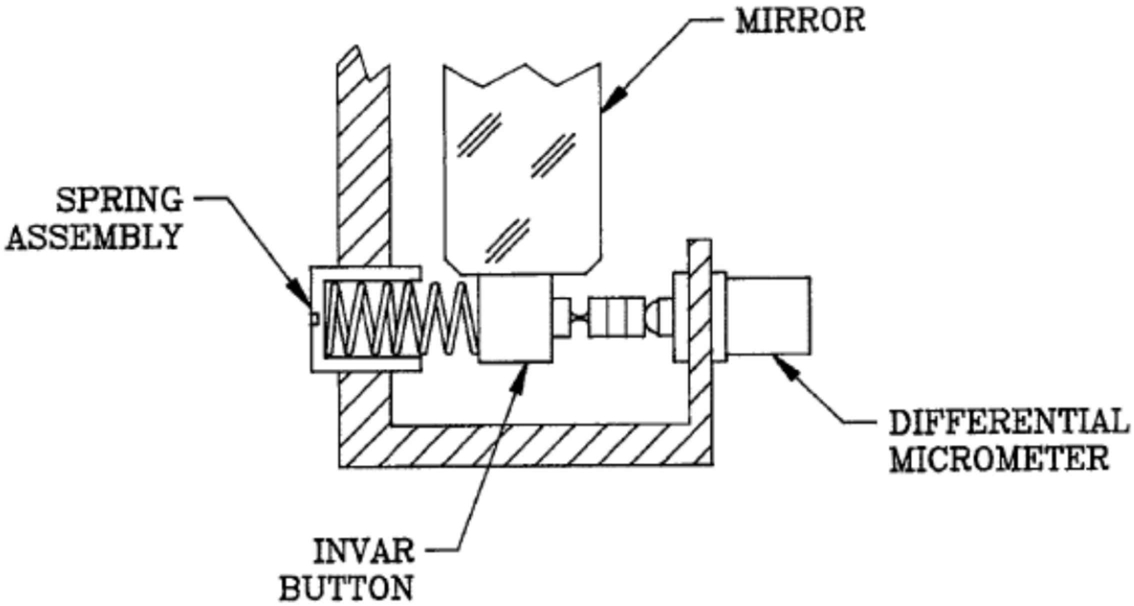
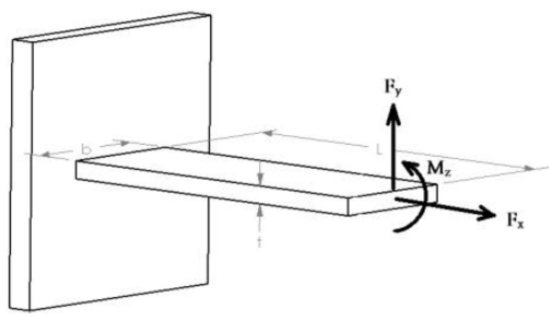


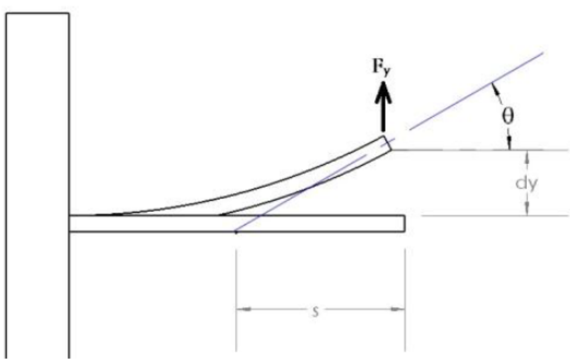
FIGURE 7.41 A tilt mechanism suitable for high resonance adjustable mirror mounts.

Flexures

Flexures provide a means of precise adjustment using the elastic deflection of materials due to an applied force; they can provide very rigid constraints in certain directions while still maintaining compliance in others. Flexures have low hysteresis, low friction, and are suitable for small rotations (< ~5 deg) and translations (< ~2 mm). They can also provide mechanical and thermal isolation of an optical element from its housing. Flexures typically cannot tolerate large loads, and there must be low residual stress in the flexure from fabrication. Large tensile loads may be tolerated in one direction, based on the geometry of the flexure.

The most simple and common type of flexure is the **single-strip** or **leaf type**, which is useful for small rotations.





We define the blade length L, Young's modulus E, thickness t, and width b; the moment of inertia I is:

$$I = \frac{1}{12} bt^3$$

Stiffness Relations

For simple loading, the **stiffness relations** are:

$$\kappa_{\theta_z} = \frac{M_z}{\theta_z} = \frac{EI}{L} \qquad k_y = \frac{F_y}{dy} = \frac{3EI}{L^3} \qquad k_x = \frac{F_x}{dx} = \frac{Ebt}{L}$$

$$\frac{F_y}{\theta_z} = \frac{2EI}{L^2} \qquad \frac{M_z}{dy} = \frac{3EI}{L^3}$$

The maximum stress in the flexure is:

$$\frac{6M}{bt^2} \text{ or } \frac{6FL}{bt^2}$$

The stiffness relations change in the presence of a tensile (T) or compressive (C) force F_x, which is positive for T and negative for C:

$$\kappa_{\theta_z} = \frac{M_z}{\theta_z} = \begin{cases} \dfrac{EI}{L}\omega\coth\omega & \text{(T)} \\ \dfrac{EI}{L}\omega\cot\omega & \text{(C)} \end{cases} \cong \frac{EI}{L}\left(1 + \frac{F_x L^2}{3EI}\right)$$

$$k_y = \frac{F_y}{dy} = \begin{cases} \dfrac{F_x\omega}{L[\omega - \tanh(\omega)]} & \text{(T)} \\ \dfrac{F_x\omega}{L[\tan(\omega) - \omega]} & \text{(C)} \end{cases} \cong \frac{3EI}{L^3}\left(1 + \frac{2}{5}\frac{F_x L^2}{EI}\right)$$

$$\omega = \sqrt{\frac{F_x L^2}{EI}}$$

Critical buckling limit is $\omega = \pi/2$.

> As a rule of thumb, limit the compressive force to 20% of critical.

For small deflections, consider the end motion as a rotation around a virtual pivot at distance s from the end:

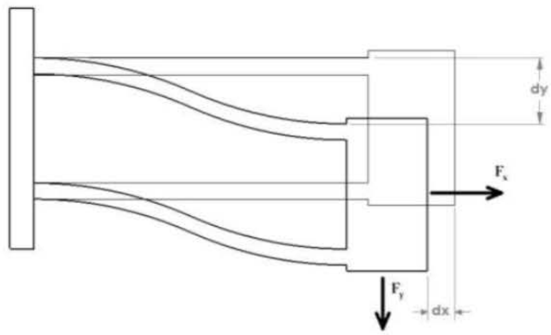
$$s = \frac{dy}{\tan\theta_z} \approx \frac{2}{3}L$$

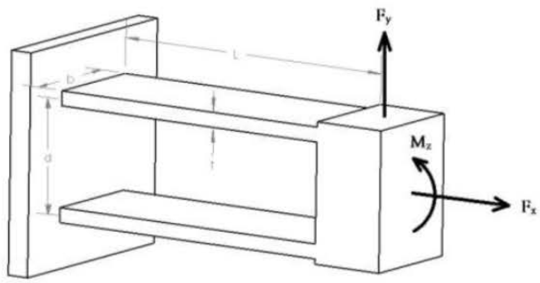
Parallel Leaf Strips

Two **parallel leaf strips** can be used for small translational motions in a **rectilinear** or **parallel spring guide**. The stiffness relations for simple loading:

$$\kappa_{\theta_z} = \frac{M_z}{\theta_z} = \frac{Ebtd^2}{2L} \qquad k_y = \frac{F_y}{\delta y} = \frac{24EI}{L^3} = 2Eb\left(\frac{t}{L}\right)^3$$

$$k_x = \frac{F_x}{dx} = \frac{2Ebt}{L}$$





The motion due to the bending of the blades is not purely parallel; the resulting axial motion is

$$\delta x = -\frac{2}{3}\frac{\delta y^2}{L}$$

The part will rotate if the flexures have different length ΔL or if the flexures are not parallel, with separation varying by Δd over the length.

$$\text{Flexures differ in length: } \theta = \frac{\Delta L}{2L}\frac{\delta y^2}{d}$$

$$\text{Flexures not parallel: } \theta = \frac{\Delta d}{L}\frac{\delta y}{d}$$

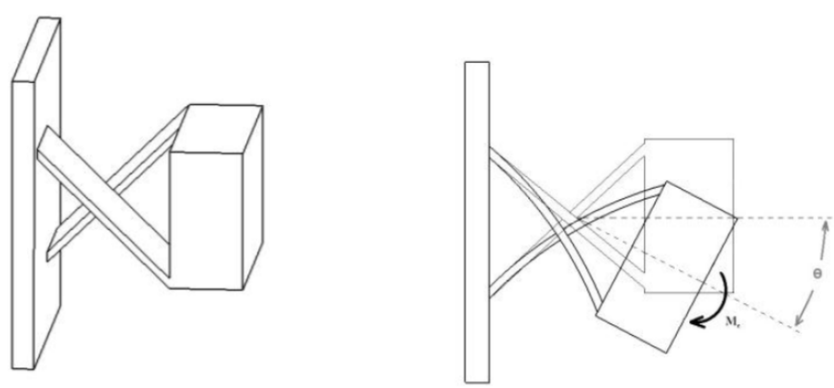
More on Stiffness Relations

The **stiffness relations** change in the presence of a tensile (T) or compressive (C) force F_x, which is positive for T and negative for C:

$$k_y = \frac{F_y}{\delta y} = \begin{cases} \dfrac{F_x}{L}\dfrac{1}{\left(1-\dfrac{\tanh\gamma}{\gamma}\right)} & \text{(T)} \\ \dfrac{F_x}{L}\dfrac{1}{\left(\dfrac{\tan\gamma}{\gamma}-1\right)} & \text{(C)} \end{cases} \cong \frac{2Ebt^3}{L^3}\left(1+\frac{3}{5}\frac{F_x L^2}{Ebt^3}\right),$$

$$\gamma = \sqrt{\frac{F_x L^2}{8EI}}$$

Cross-strip pivots allow rotation with two or more flat strips that attach between a fixed base and a moving platform. These are commercially available and are useful for applications that require larger angular deflections.



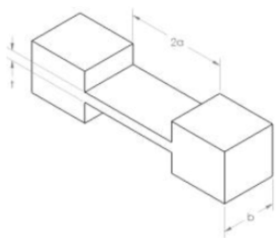
For small deflections, the axis of rotation is located at the intersection of the blades. The stiffness of the two-blade system under simple loading is given by:

$$\kappa_{\theta_z} = \frac{M_z}{\theta_z} = \frac{2EI}{L}$$

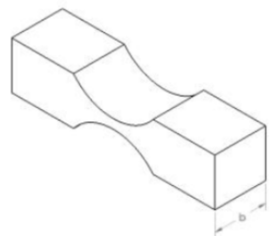
The stiffness is increased proportionally with additional blades; a three-blade design is common.

Notch Hinges

The **notch hinge** is also a common geometry for flexures. These provide added stiffness over a leaf hinge and have a better-defined center of rotation. Examples include the **leaf** (a), **circular/elliptical** (b), and **toroidal hinges** (c).



(a)



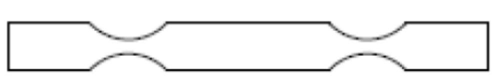
(b)

(c)

The bending stiffness $\kappa_\theta = (M_z / \theta_z)$, axial stiffness $k_x = (F_x / \delta x)$, and maximum bending stress σ_y are given below, assuming $t << a$.

	Leaf	**Circular**	**Toroidal**
κ_θ	$\dfrac{Ebt^3}{24a}$	$\dfrac{2Ebt^{5/2}}{9\pi a^{1/2}}$	$\dfrac{Et^{7/2}}{20a^{1/2}}$
k_x	$\dfrac{Ebt}{2a}$	$\dfrac{Eb^3}{12\pi a^2}\left(\sqrt{\dfrac{a}{t}}-\dfrac{1}{4}\right)^{-1}$	$\dfrac{Et^{3/2}}{2a^{1/2}}$
σ_y	$\dfrac{6M}{bt^2}$	$\dfrac{6M}{bt^2}$	$\dfrac{30M}{t^3}$

Hinge flexures are frequently used to control the line of action for a support member. If properly designed, the line of action of the force will go through the center of the hinges.



Flexure Materials

The material choice for a flexure will depend on a variety of factors, including the material's compliance, fracture toughness, thermal properties, corrosion resistance, and stability over time. The greatest compliance, given the same length flexure, is achieved by the material with the greatest **reduced tensile modulus**, defined as the ratio of the yield strength of the material to Young's modulus. The higher the reduced tensile modulus is, the more desirable the material for use as a flexure.

Flexure material	E (Gpa)	σ_{ys}/E (10^{-3})
Stainless steel 17-4	193	4.39
Titanium 6AL-4V	108	7.27
Invar 36	148	4.75
Beryllium copper	115	7.14
Aluminum 6061-T6	68	3.85

$$\sigma_{max} = \frac{My_{max}}{I} = \frac{M}{I}\left(\frac{t}{2}\right)$$

$$\Delta\theta = \frac{ML}{EI} = \frac{L}{E}\frac{2\sigma_{max}}{t} = 2\frac{\sigma_{max}}{E}\frac{L}{t}$$

Elastic limit for σ_{max} is yield stress.

L/t is the aspect ratio(ratio of length to thickness of the flexure)

Total angular change for blade flexure simply $\Delta\theta = 2\frac{\sigma_{ys}}{E}\frac{L}{t}$

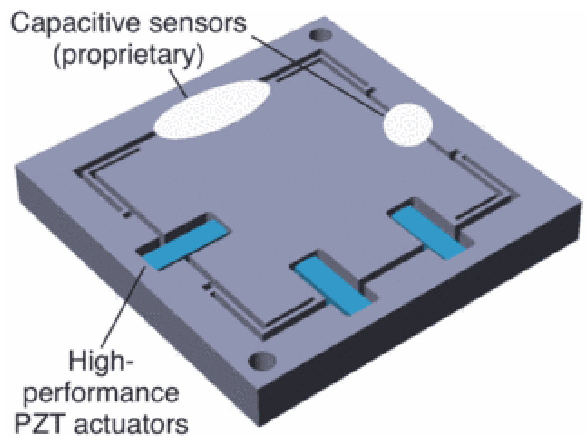
An aluminum flexure with aspect 10:1 allows rotation of 2*3.85E-3*10 = 0.077 radians or 4.4°. The same geometry using titanium allows 8.3°

Flexure stages

Tilt stage

Translation stage

(Thor Labs)



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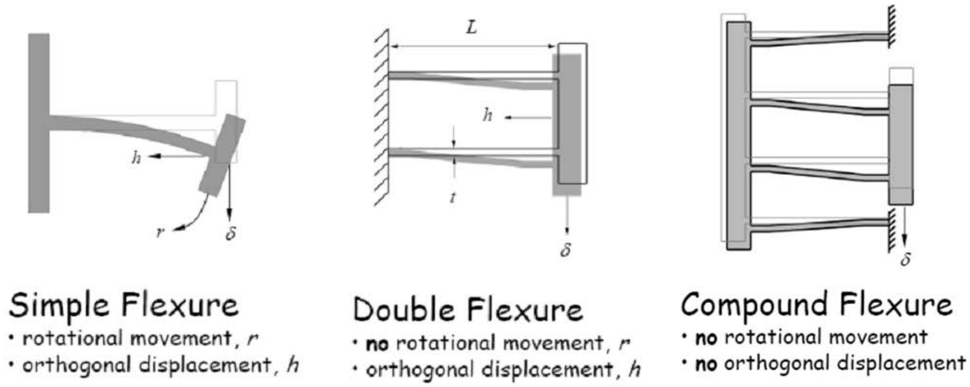
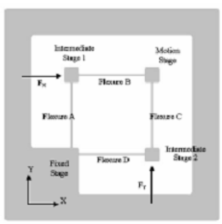
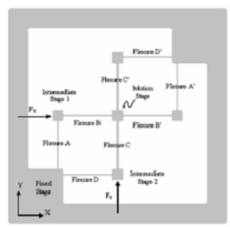
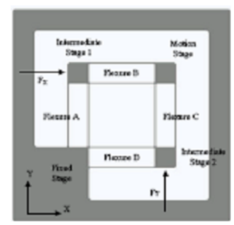
Special Issues with flexures



<u>Adding complexity to improve performance</u>







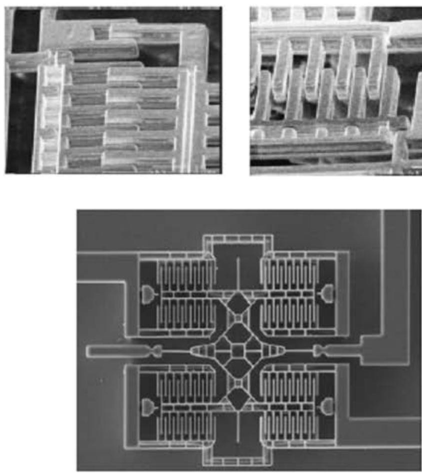
- In-plane rotation
- Parasitic motion not di-coupled
- As soon as the stage moved, Fx developed some "local" y component

- In-plane rotation minimized
- Parasitic motion reduced or cancelled
- Less cross-talk

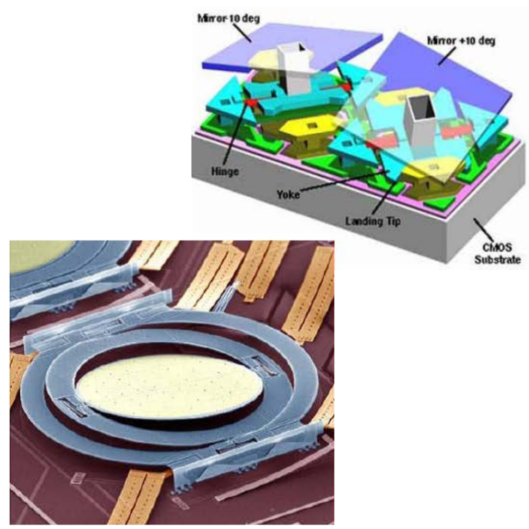
- In-plane rotation constrained
- Parasitic motion reduced
- As soon as the stage moved, Fx developed some "local" y component

(from James Wu 2007)

Micro Flexures

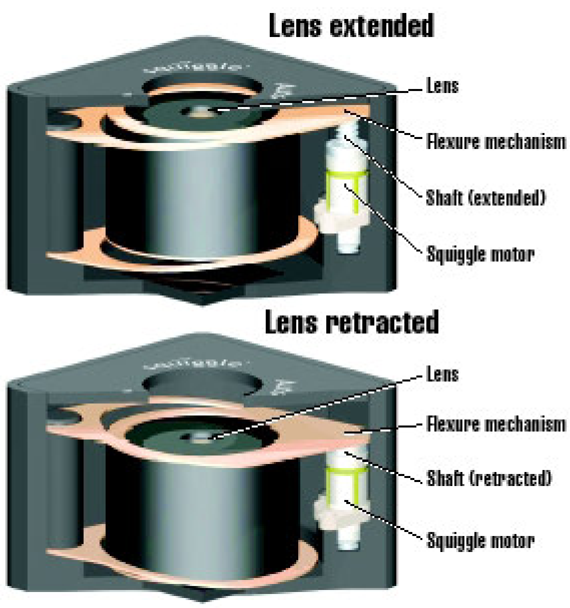


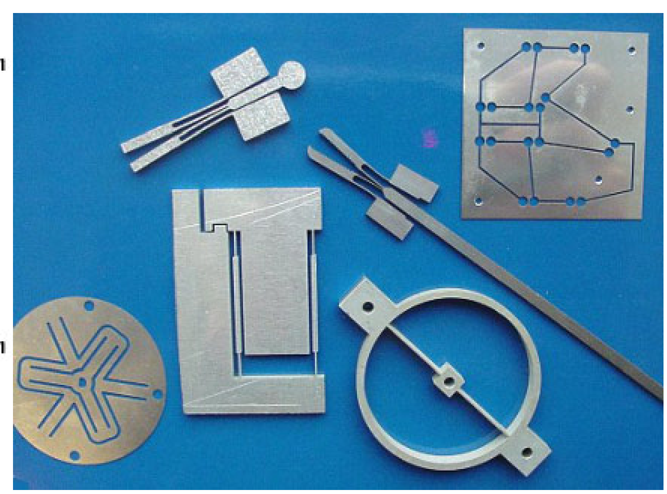
Comb drive



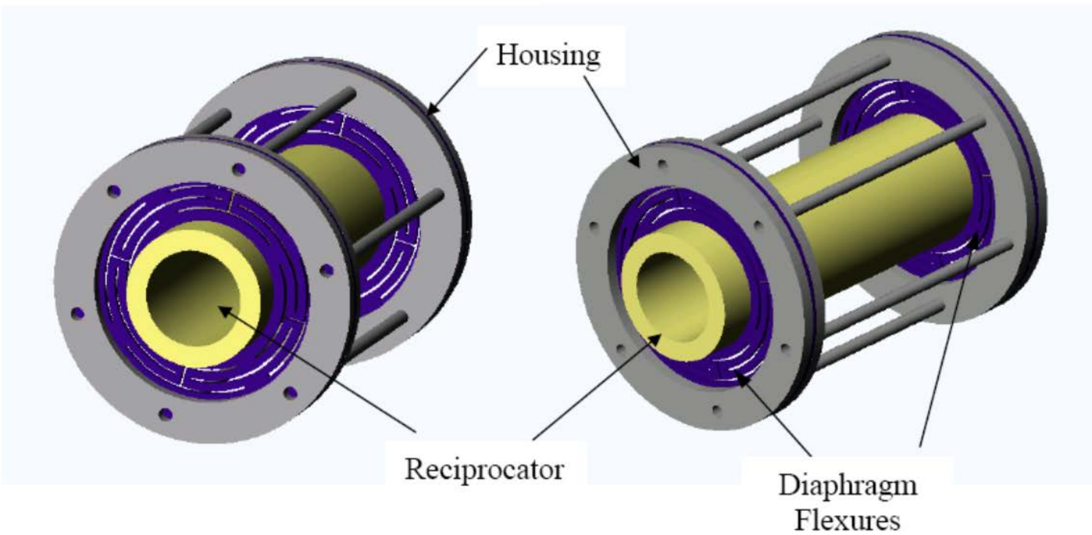
Tip-tilt mirrors

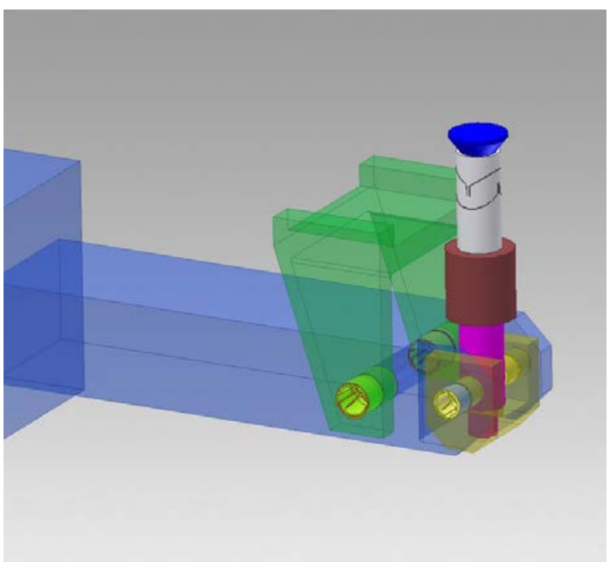
discrete vs analog

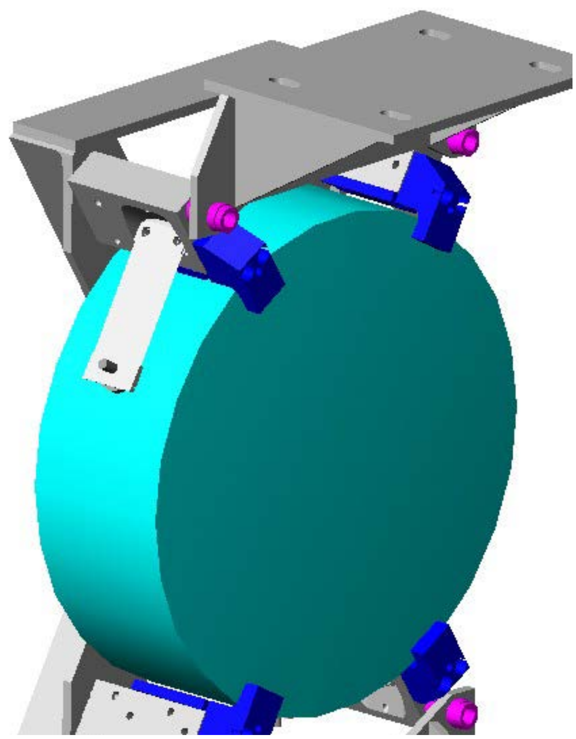


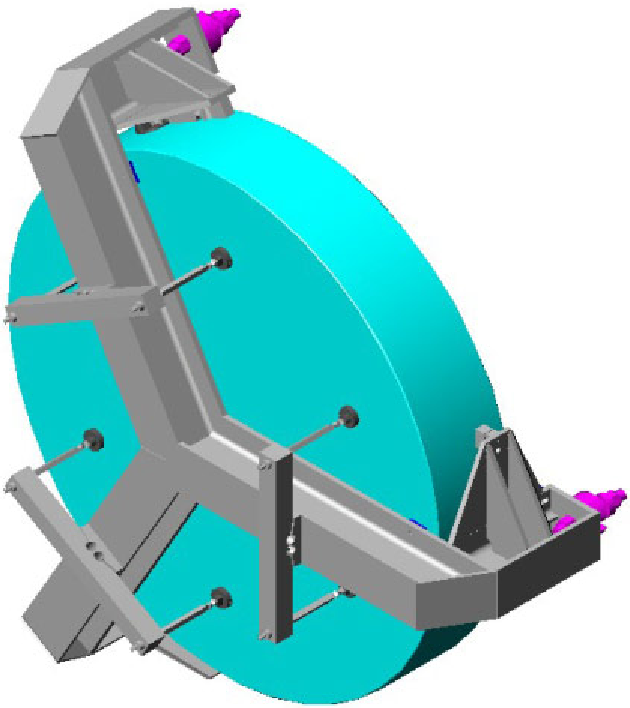


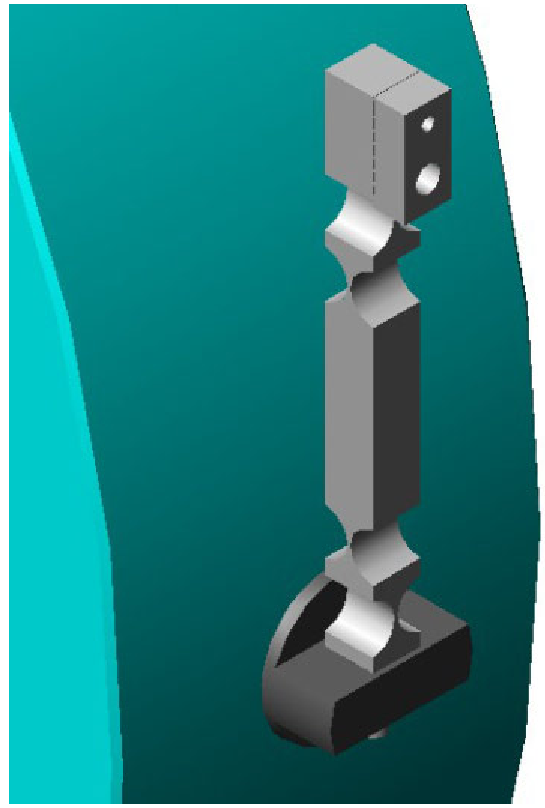
Applications of flexures in optical systems

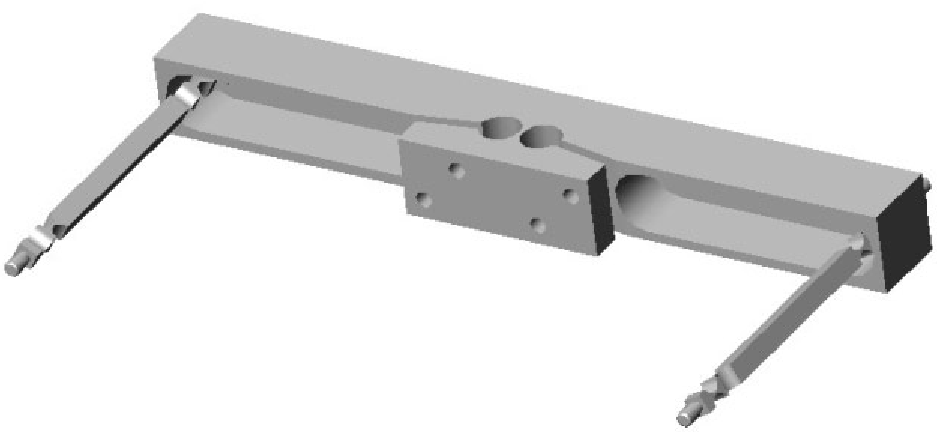








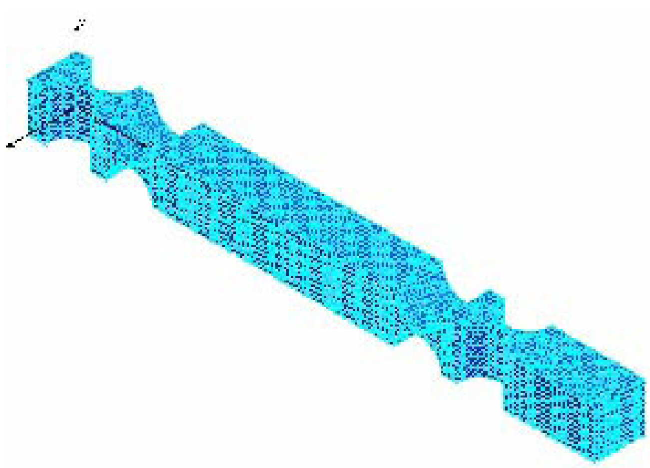


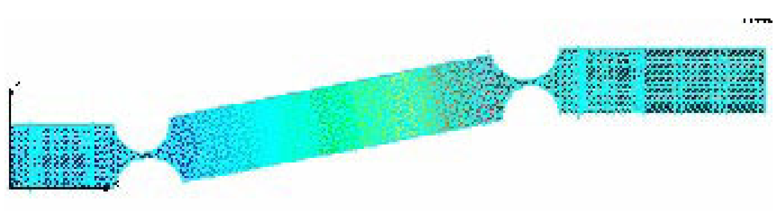


Flexure Analysis

- **Stiffness**
  - **Axial stiffness: 384K lb/in**
  - **Bending stiffness (x-dir): 139 lb/in**
  - **Bending stiffness (z-dir): 58 lb/in**
  - **Torsional compliance: 809 in-lb/rad**

- **Force & moment analysis**
  - **0.010” displacement of rod end**
    - **Shear force: 1.39 lb**
    - **Moment: 3.1 in-lb**
  - **1 deg. displacement of rod end**
    - **My: 8.83 in- lb**
    - **Mz: 7.42 in-lb**
  - **0.01 radian torsion**
    - **8.1 in-lb moment**





Use mechanical advantage to get finer resolution

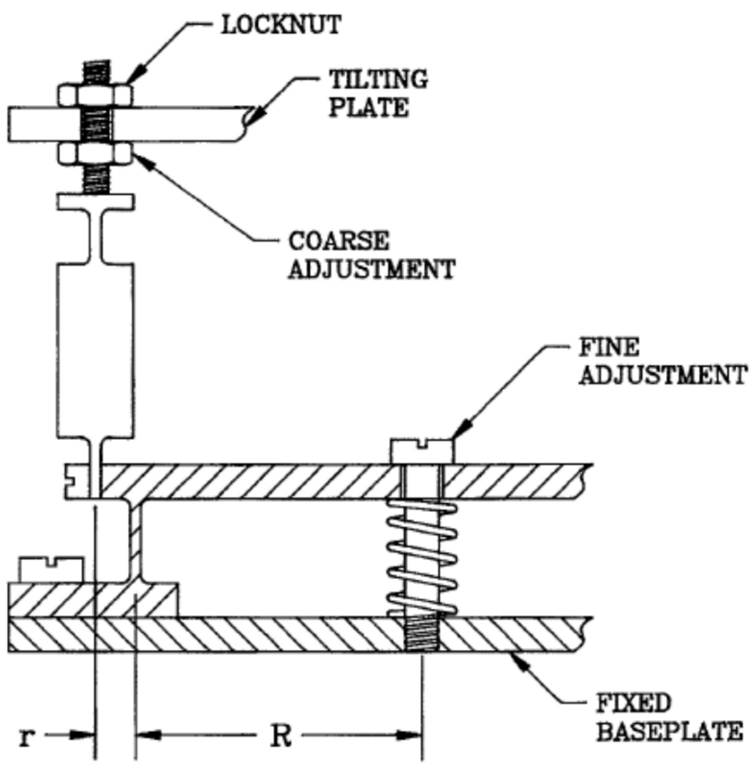


FIGURE 7.39 A tilt mechanism with coarse and fine adjustments using single blade flexures.