Adjustments and flexures

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

Like stages, 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

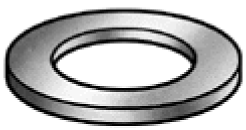
Classes of adjustments

- 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

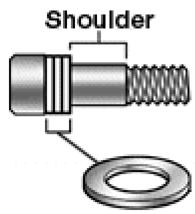
1420 products match your selections

Shim Type



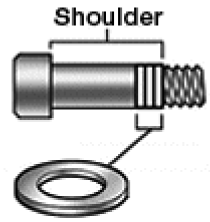
Standard

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



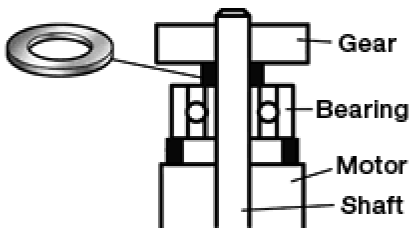
Shortening Shims for Shoulder Screws

These shims fit the shoulder diameter to shorten screw length.



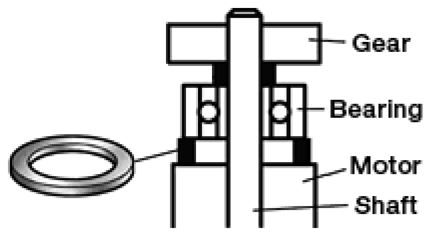
Lengthening Shims for Shoulder Screws

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



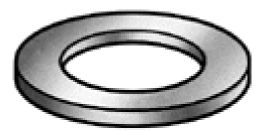
Shaft

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



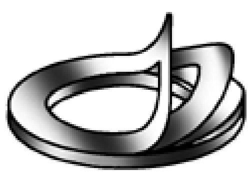
Bearing

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



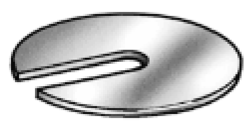
Die Punch Shims

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

Danger

This provides only 1 degree of freedom controlled constraint.

It weakly constrains the lateral dimensions

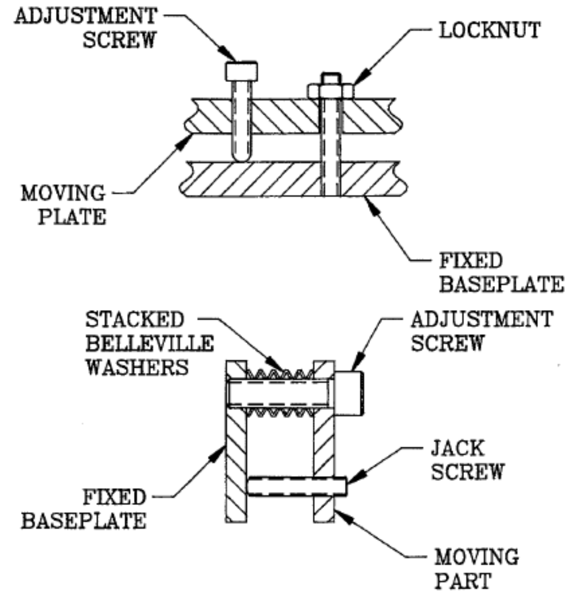
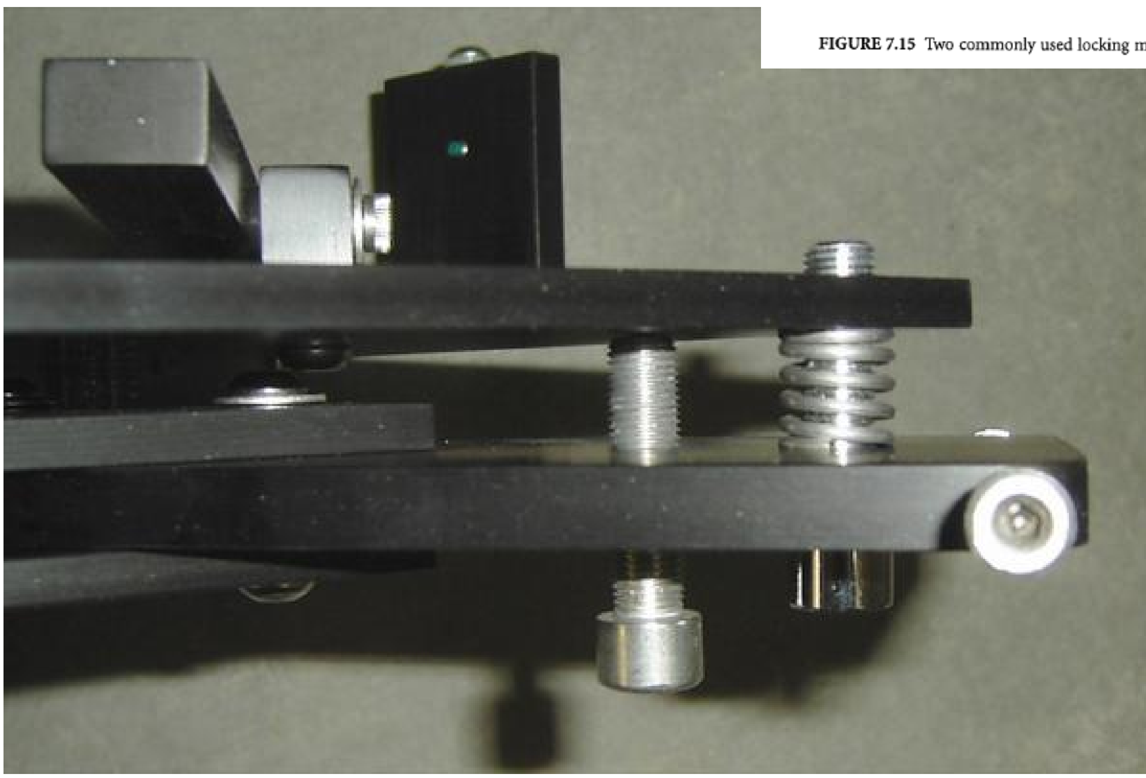


FIGURE 7.15 Two commonly used locking methods. (a) Locknut; (b) jack screw.



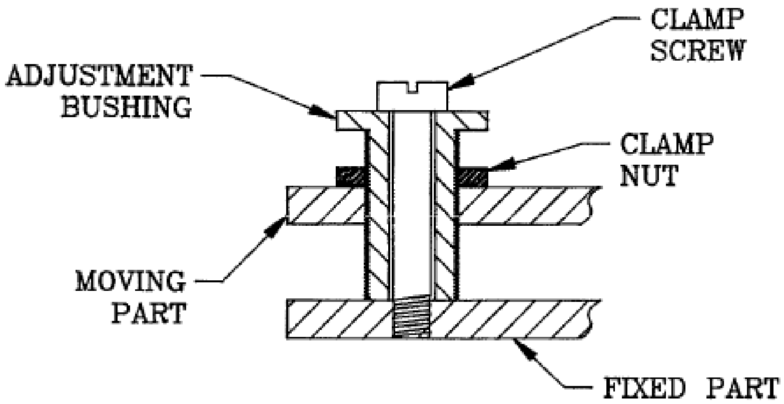


FIGURE 7.22 A linear mechanism with a threaded bushing and clamp screw suitable for applications requiring disassembly.

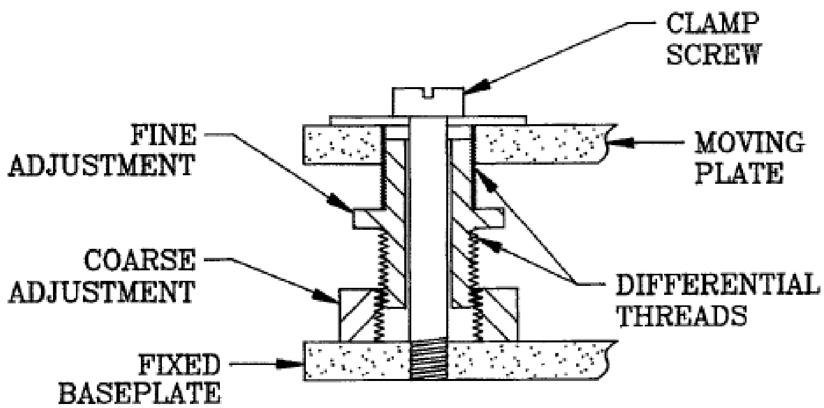
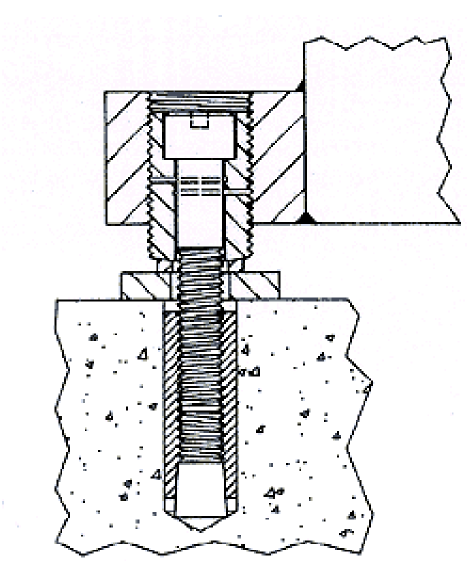
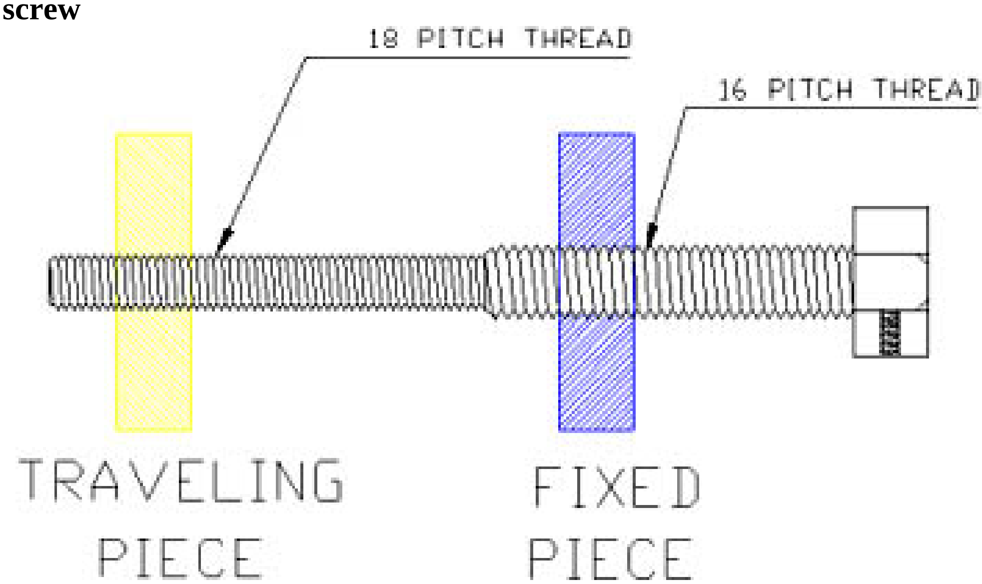


FIGURE 7.23 A linear mechanism with a bushing using differential threads for finer resolution.

Micposi
http://www.harbingerengineering.com/



Differential screw



Newport product:

DM-13 Series Differential Micrometer

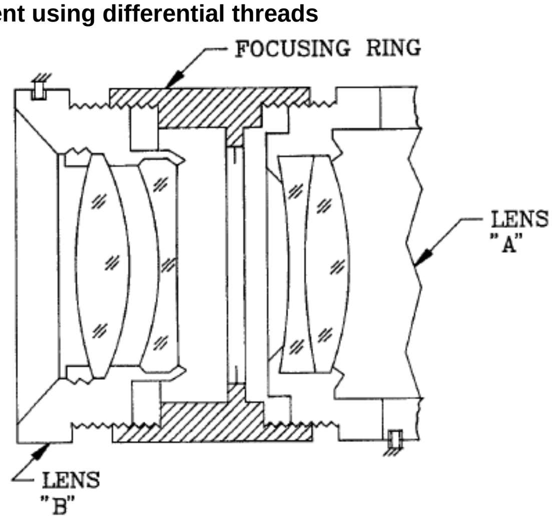
13 mm coarse travel with 0.07 µm sensitivity

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

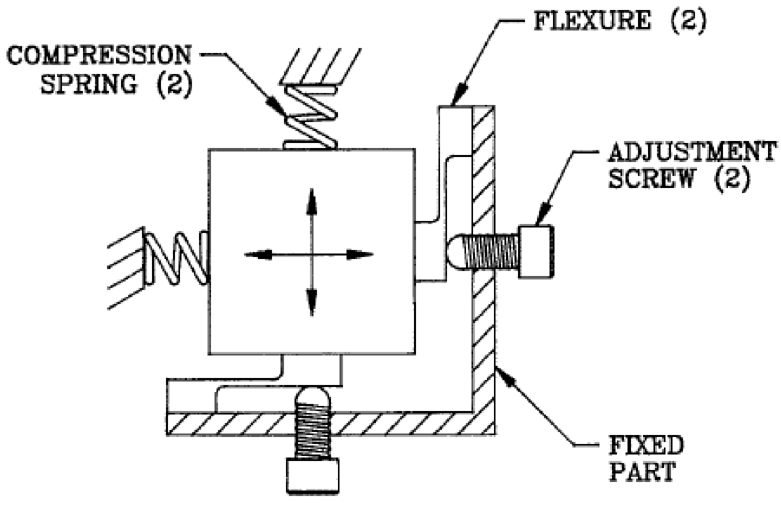
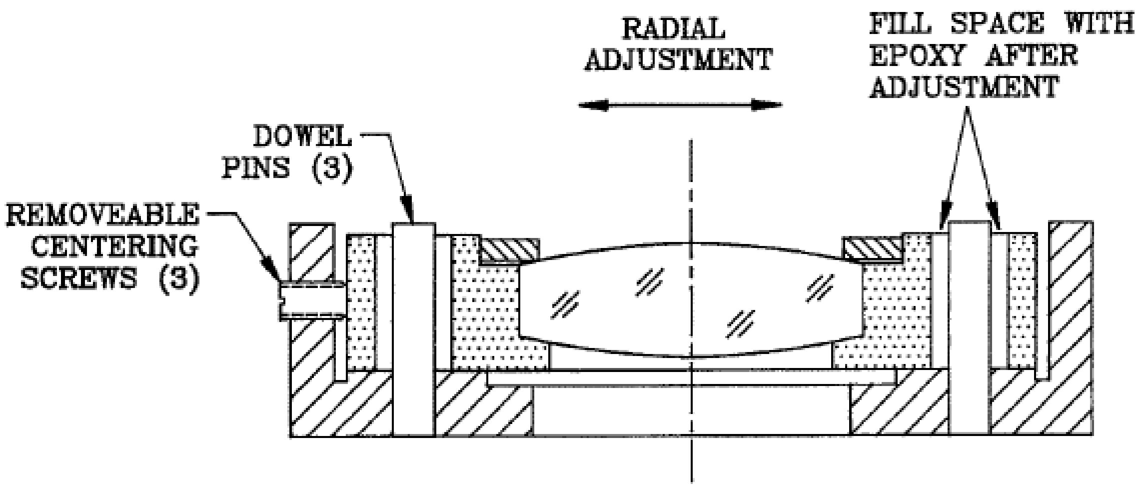
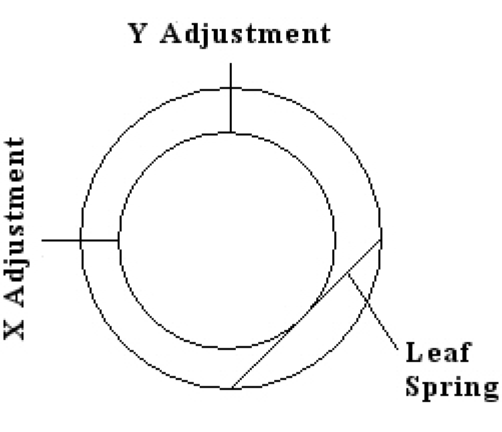
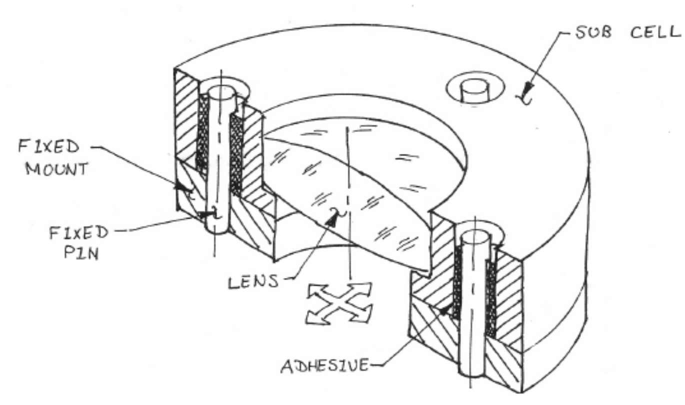
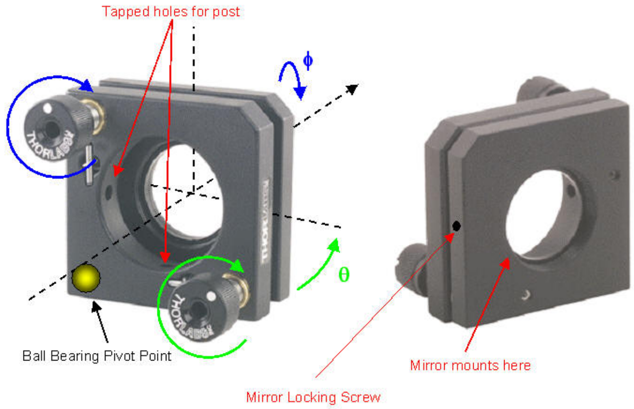


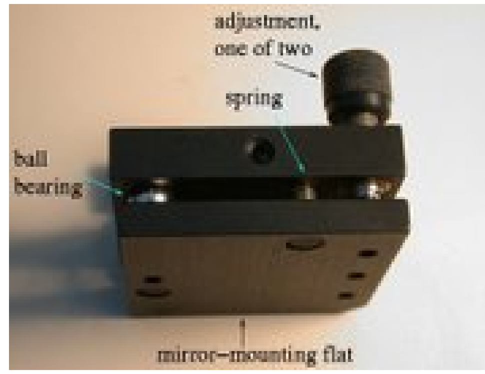
FIGURE 7.19 A two-axis linear mechanism using flexures and screw actuators.











Kinematic tip/tilt mounts

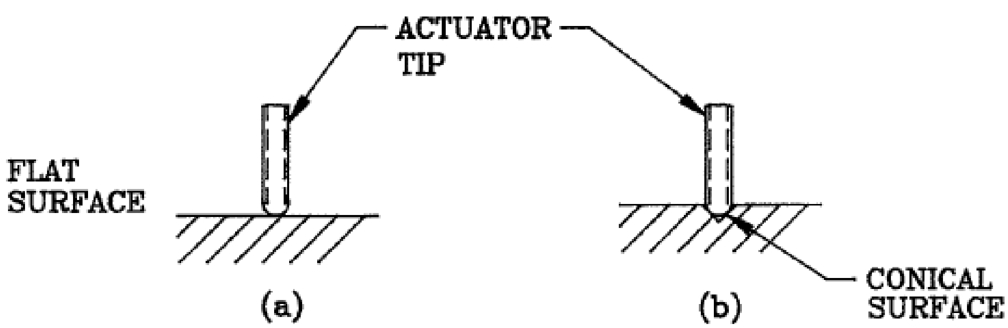


FIGURE 7.11 Two types of common interfaces for the round tip of an actuator. (a) Point contact with a flat surface; (b) line contact with a cone.

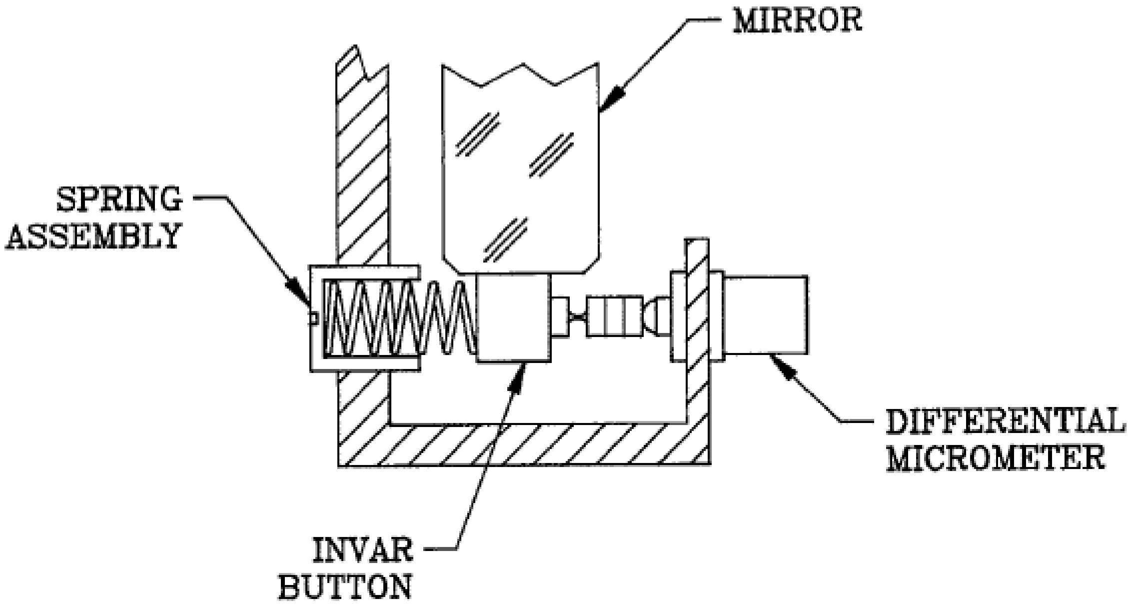
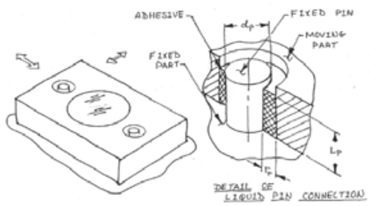


FIGURE 7.41 A tilt mechanism suitable for high resonance adjustable mirror mounts.

Liquid pinning

OPTOMECHANICAL DESIGN

- A special type of adhesive bonding used to adjust components is called liquid pinning. This uses a thin continuous constrained bond around a pin in a hole to provide adjustment, and means of locking the adjustment.



Bibliography References: 4.6.9

(Vukabratovich) 126

OPTOMECHANICAL DESIGN

- The radial stiffness of a liquid pin connection is given by:

$$k_r = \frac{\pi}{2} d_p \frac{L_p}{r_r} \left(\frac{E_r}{1-\upsilon_r^2} + G_r \right)$$

Where:

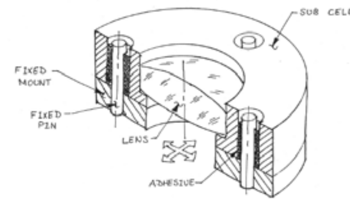
k_r	Is the radial spring rate (stiffness)
d_p	Is the pin diameter
L_p	Is the length of the adhesive bond in contact with the pin
r_r	Is the average radial thickness of the adhesive around the pin.
v_r	Is Poisson's ratio for the adhesive
G_r	Is the shear modulus of the adhesive

Bibliography References: 4.6.9

(Vukabratovich) 127

OPTOMECHANICAL DESIGN

- Liquid pinning is used to adjust the centering of lenses. The lens is placed in a sub-cell, and liquid pinning is used to adjust and lock the lens in place. If a release agent is placed on the pins, it may be possible to remove and replace the sub-cell and lens assembly from the mount without changing the adjustment.



The radial stiffness of a liquid pinned sub-cell and lens assembly ifs given by:

$$k_R = N\, k_r$$

Where:

k_R	Is the overall radial stiffness
N	Is the number of equally spaced liquid pin connections
k_r	Is the radial stiffness of a single liquid pin connection

NOTE: This assumes that the liquid pin connections are all identical, and equally spaced on a common diameter.

Bibliography References: 4.6.9

(Vukabratovich) 128

Flexures

OPTOMECHANICAL DESIGN

- For small rotation angles θ < 0.1 rad or small translations about 1 to 2 mm, flexures have the advantages of freedom from stick-slip, very low friction or hysteresis, and do **NOT** require lubrication. Flexures work well in adverse environments.
- Several types of flexures that are commonly used:
  - Single-strip flexure
  - Two strip rotational flexure
  - Parallel spring guide
  - Circular contour flexure hinge
- A good figure of merit for selecting a flexure material is the reduced tensile modulus. This is the ratio of the allowable bending stress σ_b to the elastic modulus (E) of the material. For a given flexure length, the greatest compliance is for the material with the largest σ_b / E.
- Flexures can pose severe fabrication problems. Very rigorous process control is usually required. Fabrication methods that leave very little residual stress in the flexure should be selected.
- Attention should be given to stress concentration in flexure design. Similarly, a smooth-surface finish is desirable for long life.

Bibliography References: 3.12.1, 3.12.3. 3.12.2

(Vukabratovich) 145

OPTOMECHANICAL DESIGN

SOME SELECTED METALLIC FLEXURE MATERIALS

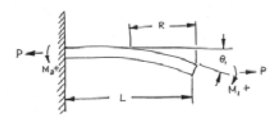
FLEXURE MATERIAL	σ_{ys} / E x 10^{-3}
ALUMINUM ALLOY 1100-H12	1.40
STAINLESS STEEL 17-4 PH COND. H1150-M	4.39
STAINLESS STEEL TYPE 304	1.25
TITANIUM 6AL-4V ELI	7.27
TITANIUM 5AL-2.5Sn ELI	5.94
MARAGING STEEL 18 Ni (250)	9.25
INVAR 36 Ni	4.75
BERYLLIUM COPPER ½ HARD	7.14

Bibliography References: 3.12.3

(Vukabratovich) 146

OPTOMECHANICAL DESIGN

- A single strip flexure is useful for small rotations:



- If p = 0

$$M_1 = M_2 = \frac{EI\,\theta_1}{L}$$

Bibliography References: 3.12.5, 3.12.6, 3.12.7, 3.12.8

(Vukabratovich) 147

OPTOMECHANICAL DESIGN

- If P is in compression

$$M_1 = \left(\frac{E\,I\,\gamma\,\theta_1}{\tan(L\,\gamma)}\right) \qquad M_2 = \left(\frac{E\,I\,\gamma\,\theta_1}{\sin(L\,\gamma)}\right)$$

$$R = \left(\frac{I}{\gamma}\right) \tan\left(\frac{L\gamma}{2}\right)$$

- If P is in tension

$$M_1 = \left(\frac{E\,I\,\gamma\,\theta_1}{\tanh(L\,\gamma)}\right) \qquad M_2 = \left(\frac{E\,I\,\gamma\,\theta_1}{\sinh(L\,\gamma)}\right)$$

$$R = \frac{I}{\gamma} \tanh\left(\frac{L\gamma}{2}\right)$$

Where:

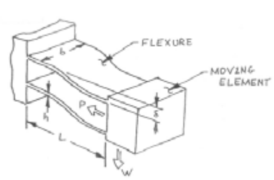
γ	((P / EI))1/2
I	Is the flexure section moment of inertia
E	Is the flexure material elastic modulus

Bibliography References: 3.12.5, 3.12.6, 3.12.7, 3.12.8

(Vukabratovich) 148

OPTOMECHANICAL DESIGN

- A parallel spring guide offers a small range of linear translation:



If p = 0

$$\delta = \left(\frac{W\,L^3}{2\,E\,b\,h^3}\right)$$

$$\sigma_{max} = \left(\frac{3\,W\,L}{2\,b\,h^2}\right)$$

Bibliography References: 3.12.13, 3.12.14

(Vukabratovich) 149

OPTOMECHANICAL DESIGN

- If P is in compression

$$\delta = \frac{W\,L}{P}\left[\frac{1}{\left(\frac{\psi\,L}{2}\right)}\tan\left(\frac{\psi\,L}{2}\right) - 1\right]$$

$$\sigma_{max} = \left(\frac{3\,W\,L}{2\left(\frac{\psi\,L}{2}\right) b\,h^2}\right)\tan\left(\frac{\psi\,L}{2}\right) + \left(\frac{P}{2\,b\,h}\right)$$

- If P is in tension

$$\delta = \left(\frac{W\,L}{P}\right)\left[1 - \frac{1}{\left(\frac{\psi\,L}{2}\right)}\tanh\left(\frac{\psi L}{2}\right)\right]$$

$$\sigma_{max} = \left(\frac{3\,W\,L}{2\left(\frac{\psi\,L}{2}\right) b\,H^2}\right)\tanh\left(\frac{\psi\,L}{2}\right) + \left(\frac{P}{2\,b\,h}\right)$$

Where:

$$\left(\frac{\psi\,L}{2}\right) = \left(\frac{3\,P\,L^2}{2\,E\,b\,h^3}\right)^{\frac{1}{2}}$$

E	Is the flexure material elastic modulus
σ_{max}	Is the maximum flexure stress

Bibliography References: 3.12.13, 3.12.14

(Vukabratovich) 150

OPTOMECHANICAL DESIGN

- There is a change in length of foreshortening of a parallel spring guide flexure due to motion in translation. Due to this change in length the motion of the flexure in translation is not truly linear. There is cross-coupled motion between translation and vertical motion of the flexure. The amount of cross-coupled motion or foreshortening is given by:

$$\Delta L = L\left[1-\frac{L}{2\delta}\sin\left(\frac{2\delta}{L}\right)\right]$$

Where:

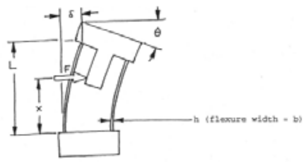
ΔL Is the cross-coupling or foreshortening of the flexure due to translation
L Is the parallel spring guide length
δ Is the linear translation of the parallel spring guide

Bibliography References: 3.12.16

(Vukabratovich) 151

OPTOMECHANICAL DESIGN

- Parallel spring guides are vulnerable to error. If the translating force (F) is not applied midway between the flexures, the moving carriage pitches or tilts as it is displaced. The pitch angle (θ) is estimated from



$$\theta \cong \left(\frac{\delta}{L}\right)\left[\frac{6\,(L-2x)\,h^2}{3b^2L-2h^2L-6xh^2}\right]$$

- If h << L, b THEN

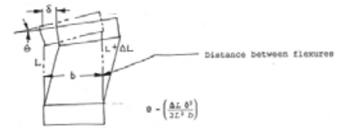
$$\theta \cong \frac{2\,\delta\,h^2\,(L-2x)}{b^2\,L^2}$$

Bibliography References: 3.12.16

(Vukabratovich) 153

OPTOMECHANICAL DESIGN

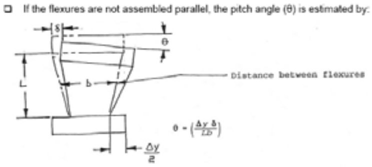
- If a parallel spring guide has assembly errors, the moving carriage pitches or tilts as it is displaced.
- If the flexures differ in length, the pitch angle (θ) is estimated from:



Where:

L Is the flexure length
ΔL Is the flexure length difference
δ Is the carriage displacement
b Is the distance between flexures

- If the flexures are not assembled parallel, the pitch angle (θ) is estimated by:

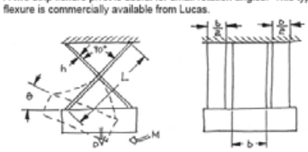


Bibliography References: 3.12.16

(Vukabratovich) 154

OPTOMECHANICAL DESIGN

- A two strip flexure pivot is useful for small rotation angles. This type of flexure is commercially available from Lucas.



- If P is in compression

$$M=\left(\frac{EI\gamma}{2}\right)\left[\left(\frac{L\gamma}{2}\right)+\cot\left(\frac{L\gamma}{2}\right)\right]\theta$$

- If P is in tension

$$M = \left(\frac{E\,I\,\gamma}{2}\right)\left[\coth\left(\frac{L\,\gamma}{2}\right)-\left(\frac{L\,\gamma}{2}\right)\right]\theta$$

Where:

I Is the single strip moment of inertia
E Is the elastic modulus
$\gamma = (P/(EI))^{1/2}$

- The shift of instant center of rotation (ΔR) as the pivot is rotated is estimated by:

$\Delta R \approx 0.24\,\theta\,L$

Bibliography References: 3.12.4, 3.12.17, 3.12.18, 3.12.11

(Vukabratovich) 155

OPTOMECHANICAL DESIGN

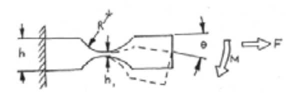
- Lucas Free-Flex pivots; the diameter of the largest pivot is about 25 mm, and its load capacity is approximately 1000 kg normal to the axis.

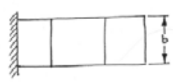
Bibliography References: 3.12.21, 3.12.22.

(Vukabratovich) 156

OPTOMECHANICAL DESIGN

- For bending applications where a well defined center of rotation is needed or where greater stiffness than a single strip flexure would give, a circular contour flexure can be used.





- If (h/2R) << 1 and (h_1/2R) << (h/2R), F = 0
- Then the bending stiffness is estimated by:

$$\frac{\theta}{M} \cong \left(\frac{9\,\pi\,R^{\frac{1}{2}}}{2\,E\,bh_1^{\frac{5}{2}}}\right)$$

Where:

E Is the elastic modulus

Bibliography References: 3.12.21, 3.12.22.

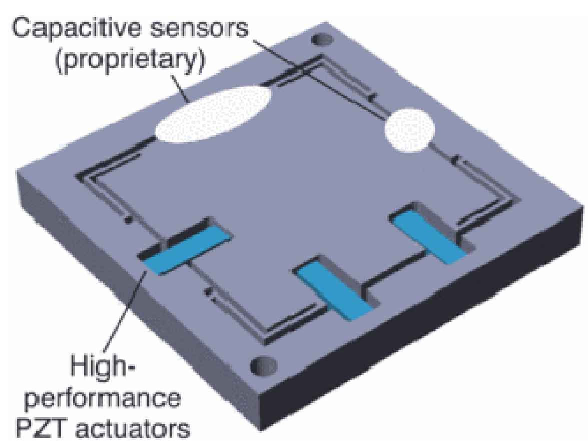
(Vukabratovich) 157

Flexure stages

Tilt stage

Translation stage

(Thor Labs)



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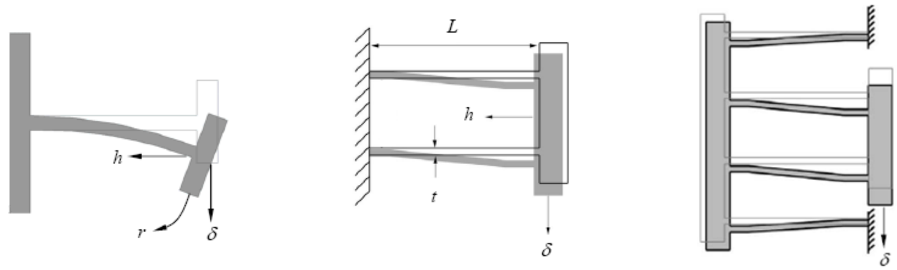
Special Issues with flexures

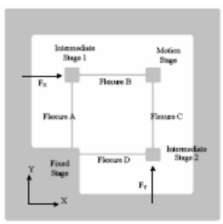


Simple Flexure
- rotational movement, *r*
- orthogonal displacement, *h*

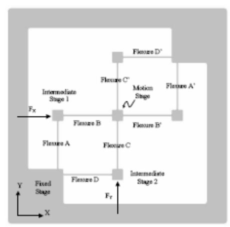
Double Flexure
- **no** rotational movement, *r*
- orthogonal displacement, *h*

Compound Flexure
- **no** rotational movement
- **no** orthogonal displacement

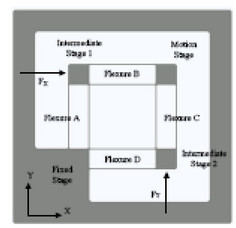
<u>Adding complexity to improve performance</u>



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



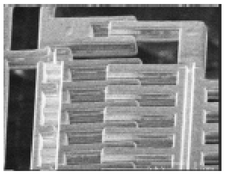
- In-plane rotation minimized
- Parasitic motion reduced or cancelled
- Less cross-talk

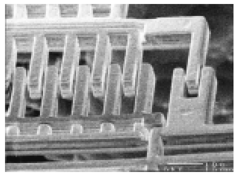


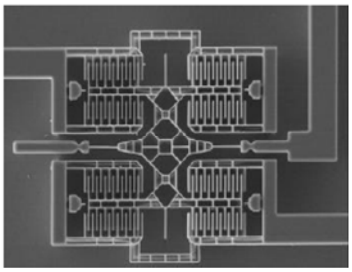
- In-plane rotation constrained
- Parasitic motion reduced
- As soon as the stage moved, Fx developed some “local” y component

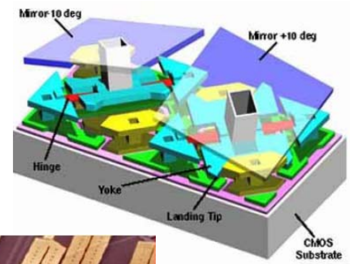
(from James Wu 2007)

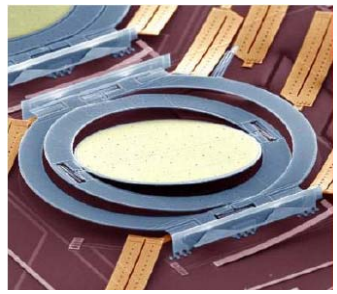
Micro Flexures







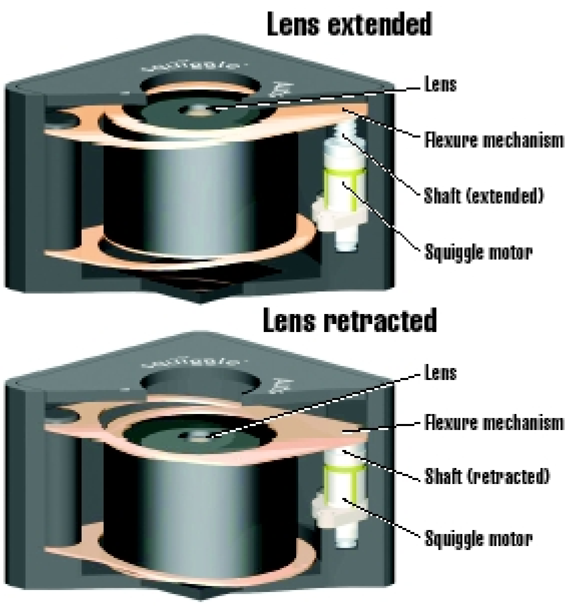


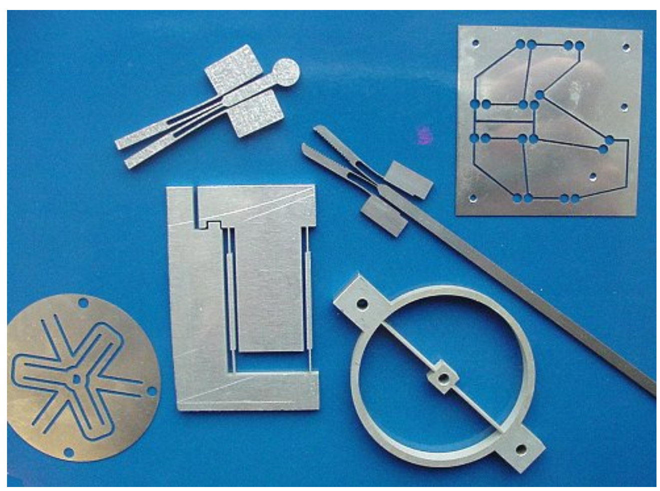


Comb drive

Tip-tilt mirrors

discrete vs analog





Use mechanical advantage to get finer resolution

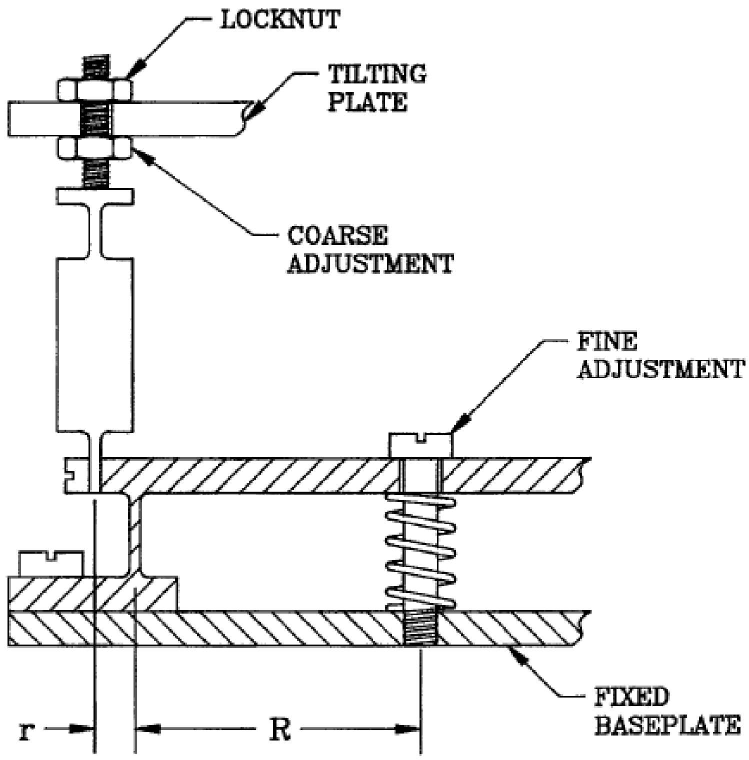


FIGURE 7.39 A tilt mechanism with coarse and fine adjustments using single blade flexures.

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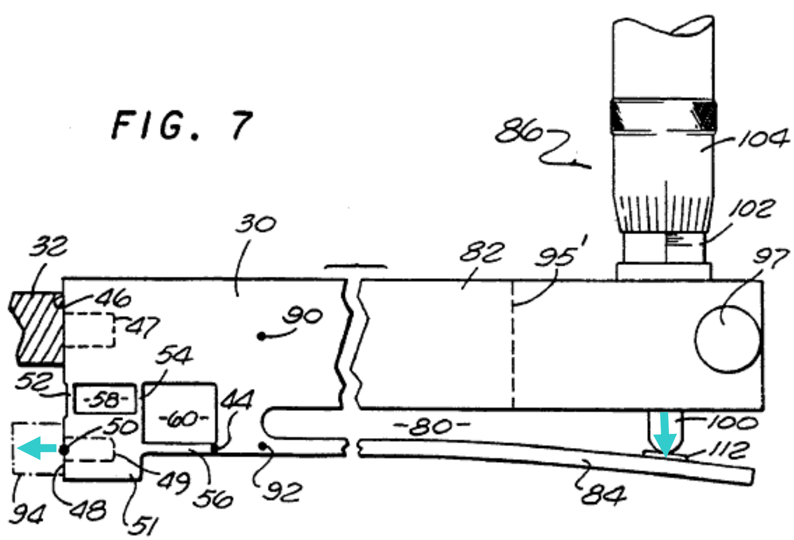
FIG. 7

0.001 µm here



10 µm here