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Mounting of Optical Components

General approaches to designing optical mounts

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The Design Process – applied to optomechanics

1. **Know the requirements:**
   - Position, 6 DoF accuracy, stability
   - Allowable distortion or internal stress
   - Operational environment
   - Survival environment
   - Interfaces
2. **Choose design form to constrain all degrees of freedom**
   - Systematically check all , place special emphasis on important ones.
3. **Design for operational environment**
   - Thermal changes cause shift and stress
   - Vibration, dynamic loading
4. **Design for survival**
   - Extreme temperatures should not break or misalign critical components
   - Must survive shock loading from shipping and handling

General rules of design:

- Start with the simplest concept, use rules of thumb and hand calculations.
- Identify "independent parameters" in the design and make sure that you know the impact of these
- Systematically evaluate important degrees of freedom
- Perform initial evaluation of manufacture and assembly before investing in detailed models or calculations

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Special requirements for mounting of optical elements

- Tight positional and angular tolerances, two basic strategies:
  1. Design so parts can be simply assembled, system tolerances controlled by component tolerances
     - Provides lowest cost, most stable, most robust solution
     - Understand which dimensions are critical
     - Manufacturing issues can dominate
     - Effective GD&T becomes critical
  2. Design so that adjustments or compensations are made during assembly
     - Each adjustment is costly:
       – Time, equipment required to make the adjustment
       – Limitation from adjustment resolution
       – Limitation from measurement that is used to guide the adjustment
       – Cross coupling is a killer for efficiency
     - Can effect stability, which limits performance
- Small distortions or stresses can limit performance
  – Small distortion in mirror surface creates optical aberrations
  – Stress creates birefringence in optical materials
- Limited materials, most have poor mechanical properties
  – Thermal expansion causes stress and stability problems
  – Optical materials are fragile
  – Many materials and coatings degrade optical performance
- Environments can be terrible!
  – Window to the cruel world
  – Cryogenic, vacuum, or caustic environment

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Control of position and tilt for optics

- **Optics that are in the wrong place will degrade image or wavefront quality**
  - Errors in the relative position for lenses and powered mirrors degrade optical performance
  - Tolerancing of the system : develop an error budget using computer simulation
    - Define figure of merit for performance
    - Simulate perturbation of each degree of freedom for the system and investigate the effect on performance
    - Create an error budget that includes everything that affects performance
    - Monte Carlo simulation to allow everything to go wrong at the same time
  - Developing appropriate tolerances must be iterative --
    - Optical engineer assigns initial tolerances
    - Mechanical engineer develops concept, estimates manufacturing limitations
    - Adjust to balance cost, performance
  - Best scenario : optical engineer knows the mechanical issues and designs system to limit sensitivity to errors. Sometimes this doesn't happen
  - Design multi-element lenses in groups. Hold tight tolerances for elements within a group. Looser tolerances between groups

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General approach for mounting of optical elements

Two options for mounting an optical element:

- Bond it
- Clamp it

Issues with Bonding

- One time assembly, difficult to take apart
- Easy, stiff in normal direction
- Can be compliant in shear direction
- CTE differences can cause large stresses, especially for
  - Large temperature ranges
  - Large CTE difference
  - Large dimensions
- Use UV curing cement to allow adjustment before curing
- Can provide seal
- Provides some compliance, mitigates stress from shock loading
- Requires careful preparation of surfaces
- May require special jigs and procedures
- Possibility of outgassing, affecting coatings
- Does not require preload

Issues with clamping

- Allows easy separation of constraint and preload
- Allows disassembly
- Clamping forces should be controlled
- Can cause distortions, affect performance
- Can cause large stresses, affect survivability
- Can allow for thermal expansion
- Lens barrels with threaded retainers, consider thread size, assembly torque

Design issues for clamping

- Control stress!
  - Interfaces
  - Preload force
- To control position and limit the stress, define clamping force separately from the constraint. (More about this later)

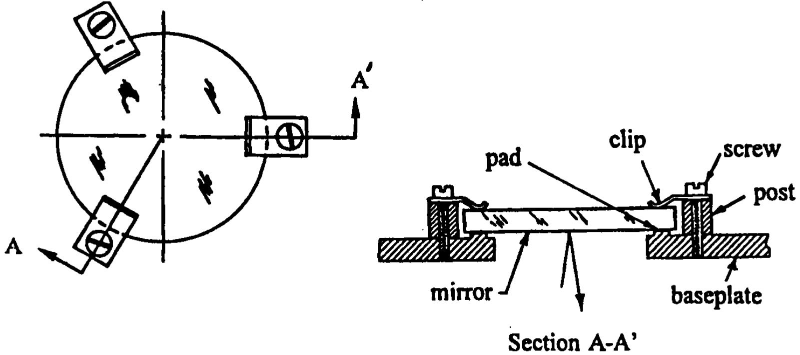

Fig. 9.1 A simple spring-clamped mirror mounting. (Adapted from Durie.[3])

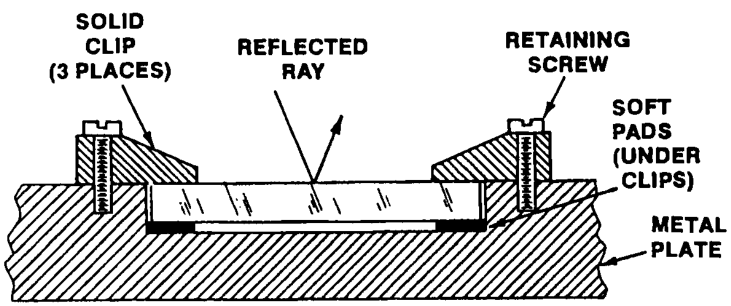

Fig. 9.3 Mirror constraint using resilient pads as springs. (From Yoder.[4])

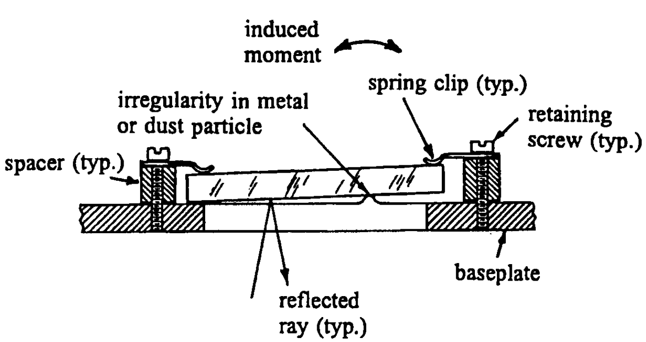

Fig. 9.2 Representation of pad irregularity or dirt particle in a mirror-to-mount interface. (Adapted from Durie.[3])

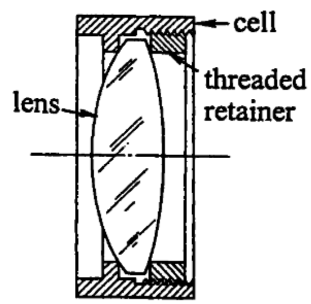

Fig. 3.17 Typical configuration of a lens secured in its mount with a threaded retaining ring. (From Yoder.[7])

Yoder, *Mounting Optics In Optical Instruments 2nd Ed.*

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Main issue with clamping, controlling the preload force

- Apply force in line with constraint to limit stress, possible distortion, and instability
- Analyze shock load:

  Force at constraint = Preload Force +/- mass * acceleration
- For shock, which is typically given in G's :
  - Metric, use mass in kg, 1 G = 9.8 m/s^2 get force in N

    1 kg optic that sees 10G shock, Force = 98 kg m/s^2= 98 N
  - English, use weight in pounds, 1 G = 386 in/s^2

    <u>but!</u>

    The weight is the really the force exerted by the optic in 1 G

    So 1 lb optic that sees 10 G shock, Force is 10 lb.
- Stress = Force/Area
  - Make sure that applied stress is not so large that it breaks the optic
  - Rule of thumb for glass, limit short term compressive stress to 50,000 psi
  - Special calculations to determine stress for point and line contacts
- Set preload to maintain contact between the optic and the constraint
  - Otherwise it can rattle, which causes very high local stress, possible fractrure
  - Position may not be stable if the contact is not maintained

Control of the preload force

- **Use compliance (springiness)**
- **Everything acts as a spring. Applied force will cause deflection. Measure spring stiffness in the *x* direction as:**

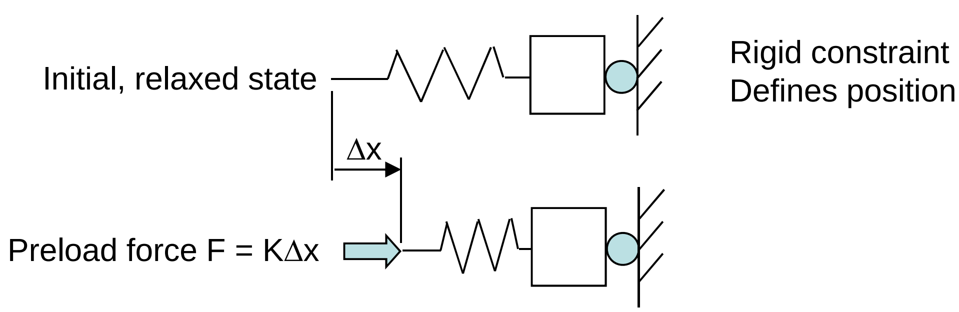
$$K = \frac{F}{\Delta x}$$

Where

F_x = Applied force in the x direction
Δx = deflection in the x direction due to the applied force

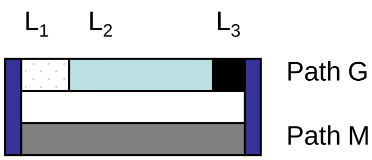
To control the preload force:

1. **Estimate the stiffness *K* of the compliant element that provides the preload**
2. **When mounting the optic, control the deflection Δx of this element as the assembly distorts it from its relaxed state.**



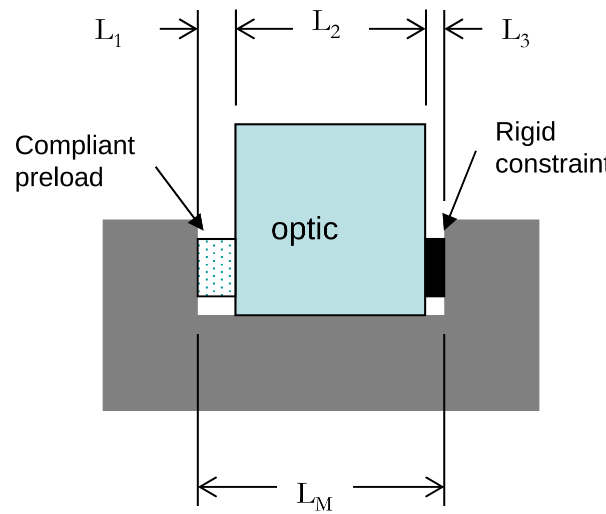
Control of thermally induced stresses

- **Design preload element to take thermal strain**
  - **preload stiffness K is << stiffness of constraint**
  - **Simplified model, use L_i, α_i, L_M, α_M**



$$L = L_M = \sum L_i$$

$$L_M = L$$



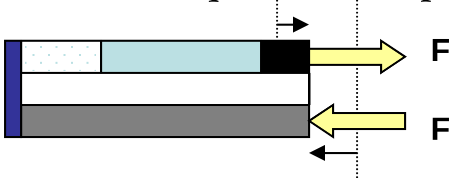
- **Use superposition to determine change in stress**

1. Allow unconstrained expansion, use equivalent CTE

$$\Delta L_{GT} = \alpha_e L \Delta T = \sum \alpha_i L_i \Delta T$$

$$\Delta L_{MT} = \alpha_M L \Delta T$$

2. Determine relationship between force and displacement
Use equivalent compliance C_e



$$\Delta L_{GF} = C_e F = \sum C_i F$$

$$\Delta L_{MF} = -C_M F$$

3. Solve for the force required to hold the thing together

$$\Delta L_G = \Delta L_M$$

$$\Delta L_{GT} + \Delta L_{GF} = \Delta L_{MT} + \Delta L_{MF}$$

$$\alpha_e L \Delta T + C_e F = \alpha_M L \Delta T - C_M F$$

$$F = \frac{(\alpha_M - \alpha_e) L \Delta T}{C_e + C_M}$$

Add this to the preload force

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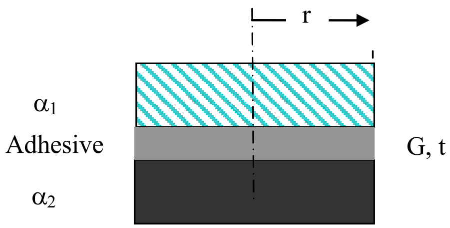
Design issues with bonding

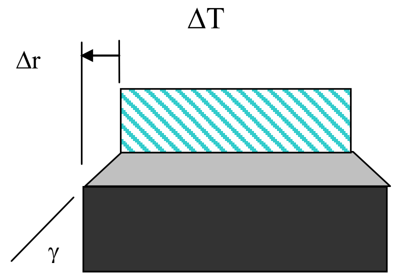
- The primary issues with bonded joints are the strength, stiffness, and sensitivity to thermal effects.
- Material properties vary widely for adhesives. Important properties are:
  - Shear modulus: Low modulus gives good isolation from thermal effects, high modulus provides good stiffness.
  - Strength: Properly prepared bonds will achieve either the bulk strength of the adhesive or the strength of the substrate
  - Optical transparency for some applications
  - Viscosity: Viscous (thick) adhesives limit the bond thickness
  - Sensitivity to environment: adhesives loose their strength in corrosive environments. Also, they swell in the presence of moisture
  - Volume change upon curing: The materials generally shrink as they cure, which can cause stress and deformation
  - Temperature range for survival: Adhesives soften and let go at high temperatures, they become stiff and brittle at low temperatures
  - Outgassing: very important for vacuum environments, but even at ambient pressures some adhesive release volatiles that contaminate optical surfaces, especially for UV applications
- Common adhesive types used in optomechanics
  - Elastomeric adhesive: *i.e.* RTV, very compliant, good sealant, poor strength
  - Epoxy: Good stiffness, strength
  - UV curing: Achieve low stress, low CTE

Thermal effects for bonded joints

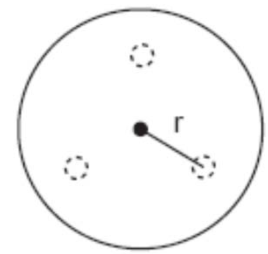
- **Bonding of materials that have different CTE creates stress in the joint.**
- **For the case where the adhesive shear compliance dominates, the shear stain (and stress) is easily calculated as below.**

For bond with maximum radial dimension r, thickness *t*





3 small bonds, distance *r* from center



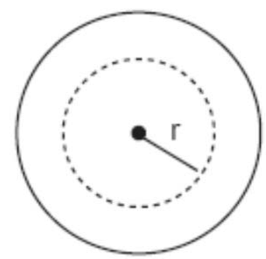
Assuming adhesive goes into pure shear, maximum shear strain, γ, at distance *r* from center is found from the differential expansion between the two materials:

$$\Delta r = r(\alpha_1 - \alpha_2)\Delta T$$

$$\gamma(r) = \frac{\Delta r}{t}$$

$$\tau_{\max} = \gamma(r) \cdot G = \frac{Gr}{t}(\alpha_1 - \alpha_2)\Delta T$$

One large bond, diameter 2*r*



This approximation will work most of the time. It is conservative. Actual stress is reduces by additional compliance of substrates. For the case of two circular plates bonded, such as doublet lenses, the maximum shear stress is

$$\tau_{\max} = \frac{(\alpha_1 - \alpha_2)\Delta T G \tanh(\beta r)}{\beta t}, \quad \beta = \left[\frac{G}{t}\left(\frac{1}{E_1 h_1} + \frac{1}{E_2 h_2}\right)\right]^{\frac{1}{2}}$$

(approximation, using 1-dimensional geometry)

where

G = shear modulus of the adhesive
r = size (radius) of the adhesive bond
t = thickness of the adhesive bond
$\alpha_1 \alpha_2$ = thermal coefficient of materials 1 and 2
E_1, E_2 = elastic moduli of materials 1 and 2
h_1, h_2 = thickness of materials 1 and 2

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Issues with design of bonded joints

- Adhesives are weakest for peal-type loads. The geometry should never allow these loads.
- The strength of the bonded joint is estimated by the shear strength of the adhesive times the bond area.
- Typical epoxy has 2000 psi shear strength. A bond with 0.1 in^2 area *should* handle 200 pound load.
- Safety factor of 3 is typical, which would allow a 67 pound load.
- Loads can come from thermal effects and from acceleration, coupled with the mass.
- Bond thickness is important:
  - Optimal thickness for strength ~0.005” for epoxy
  - Thicker bonds reduce thermal stresses, slightly reduce strength
  - Set bond thickness with fixtures, shims, or glass beads
- Many adhesives creep under shear load. This mitigates thermal stresses. Analysis uses shear modulus which is a function of time.