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MPS 1: Optic Mount Report

All equations and variables can be found at: Functional Requirements

Functional Requirements

Name	Description	Value	Range	Justification
Height Post Thermal Expansion	Max height (worst case)	120 mm	±80mm	Can't be above the specified limit or below the thickness of the materials
Center Axis Alignment	Center alignment	0	±0.2mm	Eye resolution when using a ruler. The thickness of a paper makes a notable difference
Tilt	Platform does not allow tilt	0	±0.5 degrees	Experiment with slightly crooked lines and seeing who could recognize it
Translational Stiffness	Ensuring 6 DOFS	111,250 N/m	± 1112 N/m	5lbf moving it 0.2mm
Torsional Stiffness	Ensuring 6 DOFS	350 Nm/rad	± 35 Nm/rad	Torque to turn a doorknob is about 3 Nm, rotating it 0.5 degrees
Stress limits	A and B can´t handle more stress	0	±7 kPa	Electronics wont work otherwise
Flexure range of motion	Allow relative thermal expansion between A and B	0	± 0.4mm	Allows for thermal expansion of either material rounded to 0.1mm

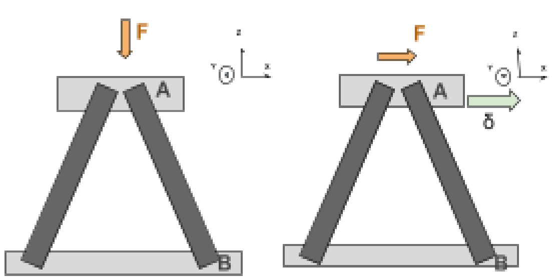

Figure 1: Theoretical flexure set-up with a load in the z direction. Each flexure makes a 45-degree angle with part B. Figure 2: the shear force and resulting displacement experienced by a pair of flexures under a torsional load in z.

Lateral stiffness calculations

We modeled the theoretical flexure setup to calculate the lateral stiffness in Z as seen in Figure 1. When subjected to a vertical force F of 5 lbs, the two fixtures will counteract x-displacement so it only displaces in the Z direction. Assuming small deformations and perpendicular flexures, this Z-displacement can be calculated as $\sqrt{2} * \delta_a$

$$\delta_a = \frac{F_a * L}{EA} = \frac{\frac{F}{\sqrt{2}} * L}{E*A} \qquad \delta_z = \sqrt{2} * \delta_a$$

$\delta_z = \frac{FL}{EA} = \frac{FL}{Etw}$ therefore for the mount with all 6 flexures: $k_z = 6 * \frac{twE}{L}$

The lateral stiffness in Z in the acrylic flexures is 4.38E+05 N/m. If the flexures were to be made of aluminum, then the lateral stiffness in Z is 4.4E+08 N/m.

Torsional stiffness calculations

When a torsional load is applied to A in the z direction, each pair of flexures experiences a shear force F as drawn in figure 2. This force is written as $F = \frac{M}{3r}$ where M is the moment applied and r is the radius from the center of A to the flexures. Similarly, the displacement can be written as $\delta = r\phi$ where ϕ is the angular displacement of A around the z axis. Given that the flexures make a 45 degree angle with B, the displacement under the shear force F can be approximated as $\delta = \frac{LF}{EA}$. Rearranging these equations, we get that the torsional stiffness is $\frac{M}{\phi} = \frac{3r^2 Etw}{L}$. With acrylic flexures, this torsional stiffness in z is 424 Nm/rad. With aluminum flexures, it would be 4.3E+05 Nm/rad.

Stress calculations

$\sigma_{max,\,flexure} = \frac{M \times c}{I}$ By relating the moment (which we simplified to $F \times L$) we were able to relate the force, width and length. We considered that the yield strength of the acrylic, according to the specs, is 72 MPa. We calculated that our design would induce a lower stress into the flexures, satisfying this requirement.

$\sigma_{max,\,part\,A} = \frac{F}{A} = \frac{12EI\delta}{AL^3} = 62.1\,Pa < 7kPa$ We first assumed that part A expanded thermally unconstrained causing a deflection in the flexure, $\delta = \alpha L \Delta T$. A fixed guided model finds the force on part A using δ. To find the max stress we used the area of the flexure interface in part A.

Discussion

This project was our first experience with designing with constraint based design. We learned how to formally define functional requirements and how to apply 2.001 concepts to calculate a flexure geometry that would satisfy the functional requirements.