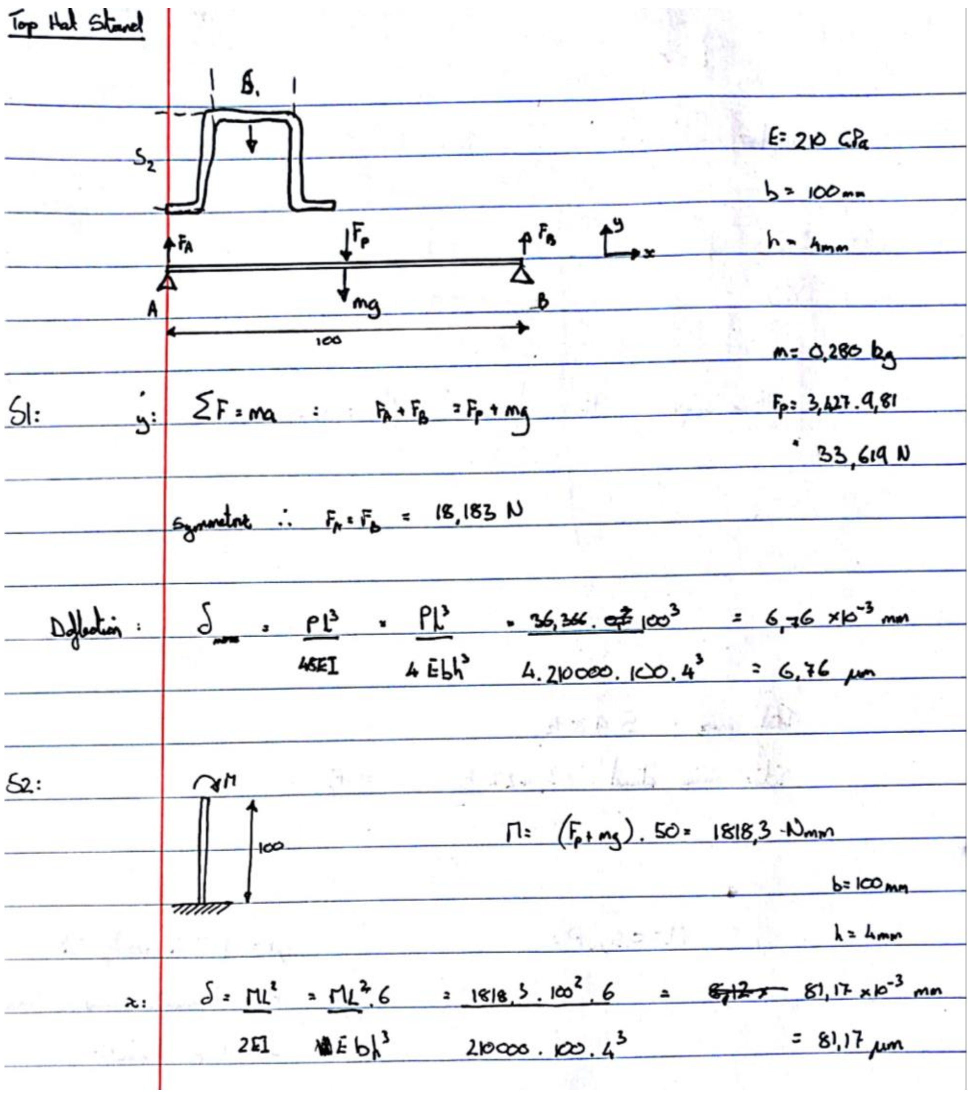
Mechanical Design Calculations

In this document, scans are taken from the author's notebook of design calculations that were made for various load bearing parts designed for use in the ISP assembly.

Body Mounting Stand

The body mount supports the entire mass of the gimbal assembly. These calculations estimated deflection in the part when manufactured from 4 mm steel.



Top Hat Stand

$E = 210$ GPa

$b = 100$ mm

$h = 4$ mm

$m = 0{,}280$ kg

$F_p = 3{,}427 \cdot 9{,}81 = 33{,}619$ N

S1: y: $\sum F = ma$: $F_A + F_B = F_p + mg$

Symmetric ∴ $F_A = F_B = 18{,}183$ N

Deflection: $\delta_{max} = \frac{PL^3}{48EI} = \frac{PL^3}{4Ebh^3} = \frac{36{,}366 \cdot 100^3}{4 \cdot 210000 \cdot 100 \cdot 4^3} = 6{,}76 \times 10^{-3}$ mm $= 6{,}76$ µm

S2: $M = (F_p + mg) \cdot 50 = 1818{,}3$ Nmm

$b = 100$ mm

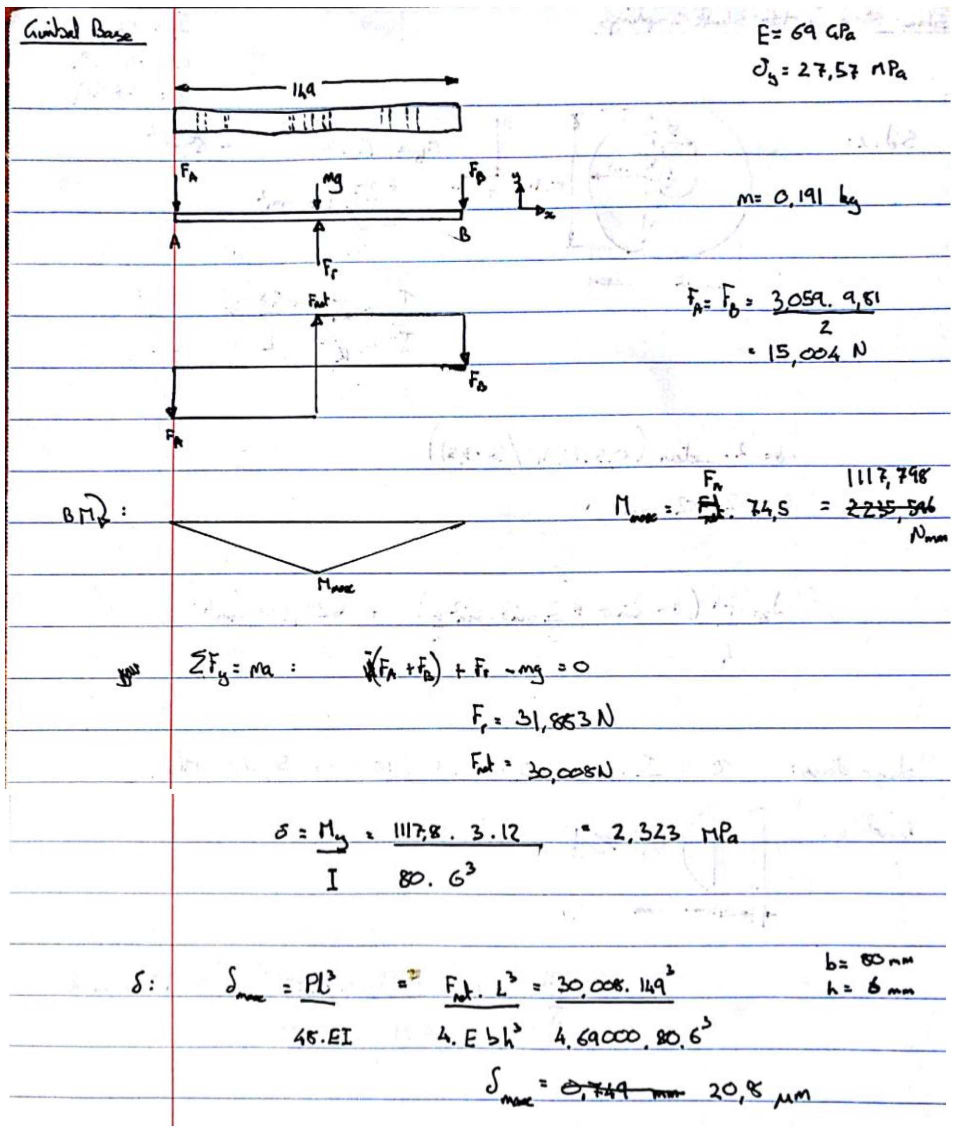
$h = 4$ mm

x: $\delta = \frac{ML^2}{2EI} = \frac{ML^2 \cdot 6}{Ebh^3} = \frac{1818{,}3 \cdot 100^2 \cdot 6}{210000 \cdot 100 \cdot 4^3} = 81{,}17 \times 10^{-3}$ mm $= 81{,}17$ µm

This part bore the load of the gimbal assembly, at the time of the design calculations, this part had a mass of 3.427 kg. The deflection of the horizontal section was estimated at only -6.76 µm in the y direction indicated, whilst calculations revealed that an x deflection of 81.17 µm would be present if the entire load was supported by one side of the vertical arms. Two arms clearly result in a smaller deflection. These estimates were very small and were acceptable values for the project. Therefore, the mounting stand was chosen to be made from 4 mm thick steel.

Gimbal Base

The gimbal base supports the vertical uprights of the yaw gimbal which in turn support the telescope platform



Calculations estimated an acceptable deflection of 20.8 µm being evident if the part was made from 6 mm Aluminum.

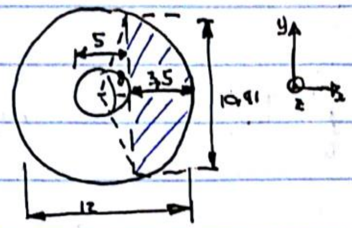
Shear Stress in Motor Shaft Couplings

The motor shaft couplings joined the motors to the inertia of the gimbal or the telescope platform

Shear Stress in Motor Shaft Couplings

$\sigma_y = 27{,}57$ MPa
$E = 69$ GPa
$G = 27$ GPa

Side 2:



$\cancel{A_b = \pi \cdot 6^2 - 85{,}66^2}$

$\cancel{= 27{,}44 \text{ mm}^2}$

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{L}$$

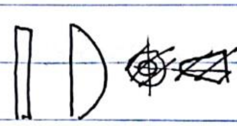
$$\theta = 2 \cdot \arctan\left(0{,}5 \cdot 10{,}91 / (6-3{,}5)\right)$$

$$\theta = 2{,}282 \text{ rad}$$

$$I_z = \frac{r^4}{4}\left(\theta - \sin\theta + \frac{2}{3}\sin\theta \cdot \sin^2\frac{\theta}{2}\right) = 629{,}138 \text{ mm}^4$$

Shear stress: $\tau = \frac{T}{J} \cdot R = \frac{0{,}539 \cdot 6 \cdot 1000}{629{,}138} = 5{,}140$ MPa

Bending:



4,5mm max

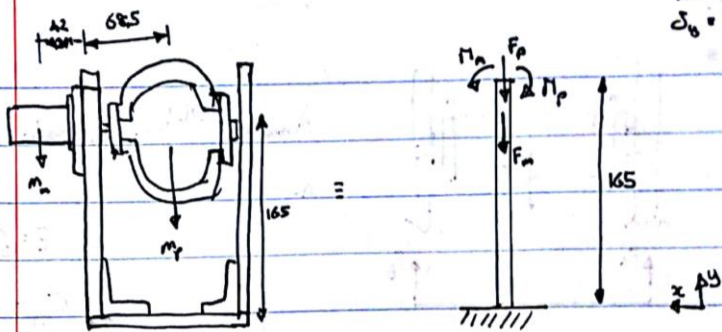
$$\theta = \frac{TL}{JG} = \frac{539 \cdot 1{,}5 \cancel{\times 10^3}}{629 \cdot 27000} = 47{,}6 \ \mu\text{rad}$$

The calculations above show a shear stress of 5.14 MPa in response to the maximum motor torque of 0.539 Nm. This is well below the yield stress of approximately 27.57 MPa for 1060 Aluminum from which the parts are made. In addition, a twist angle 47.6 µrad was expected and was deemed acceptable for the project

Gimbal Uprights

These parts support the pitch platform such that it can rotate freely.

Gimbal Arms



$E = 69$ GPa
$\sigma_y = 27{,}37$ MPa
$m_m = 0{,}242$ kg
$m_p = 1{,}899$ kg

$M_m = 0{,}242 . 42 = 10{,}164$ Nmm $. 9{,}81 = 99{,}709$ Nmm

$M_p = 0$ Nmm

$F_p = \frac{m_p . g}{2} = 1{,}899 . 9{,}81 . 0{,}5 = 9{,}315$ N

σ: BM = $M_m = 10{,}164$ Nmm $\quad y = 3$ mm

$\therefore \sigma_b = \frac{M_m y}{I} = \frac{M_m . y . 12}{bh^3} = \frac{10{,}164 . 3 . 12 . 9{,}81}{80 . 6^3}$ $\quad b = 80$ mm $\quad h = 6$ mm

$\sigma_b = \cancel{0{,}0212}$ MPa
$0{,}208$

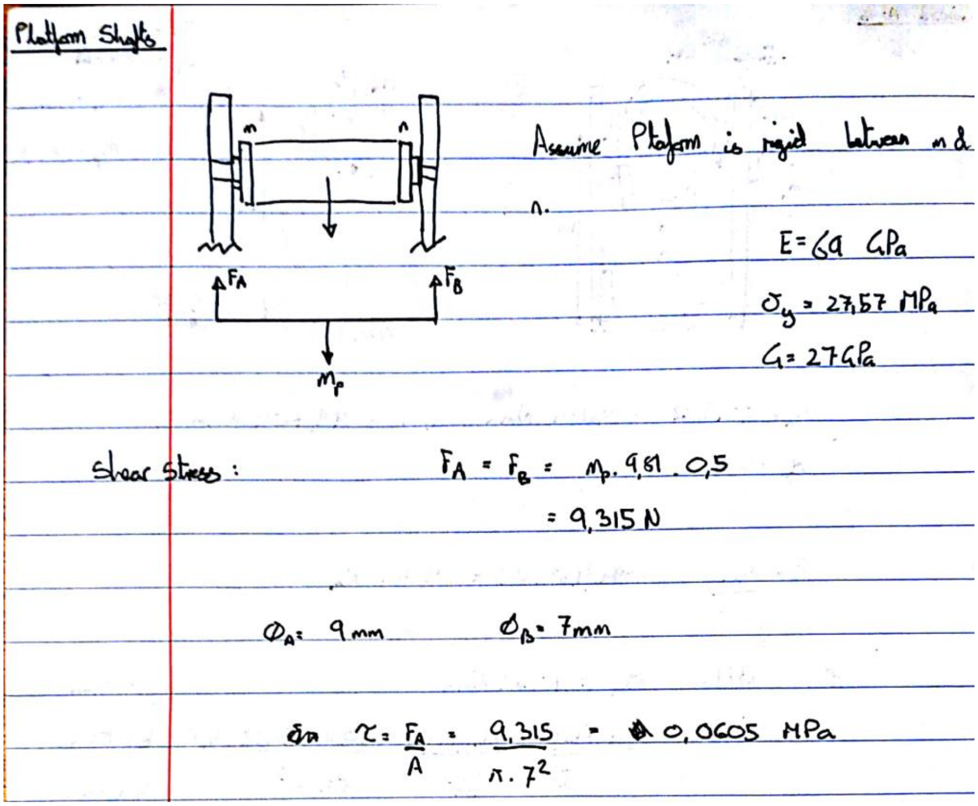
δ: $\delta_x = \frac{ML^2}{2EI} = \frac{M_m L^2 . 6}{Ebh^3} = \frac{99{,}709 . 165^2 . 6}{69000 . 80 . 6^3}$

$\delta_x = 13{,}66$ μm

From the calculations above, a shear stress in the arms was evident at 0.208 MPa, far lower than the yield stress of the 1060 Aluminum from which they were made, and a deflection in the x direction of 13.7 µm was evident.

Side 1 and Side 2 Turned

These parts supported the telescope platform and were required to bear the shear load of the telescope weight.



Platform Shafts

Assume Platform is rigid between m & n.

$E = 69$ GPa

$\sigma_y = 27.57$ MPa

$G = 27$ GPa

Shear stress: $F_A = F_B = m_p \cdot 9.81 \cdot 0.5$

$= 9.315$ N

$\varnothing_A = 9$ mm $\qquad \varnothing_B = 7$ mm

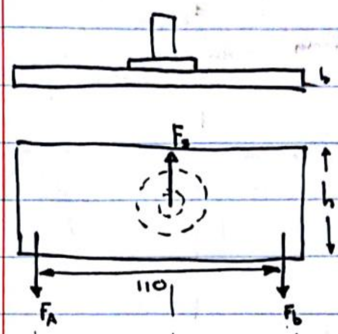
$\tau = \frac{F_A}{A} = \frac{9.315}{\pi \cdot 7^2} = 0.0605$ MPa

Shear stress in the parts at 0.061 MPa can be seen to be far lower than the limit for 1060 Aluminum from which they were made.

Platform Sides

The platform sides support the load of the telescope and transfer it to the Side 1 and 2 turned parts. In the calculations below they can be seen to be under low stress and deflect by very small amounts when manufactured from 5 mm Aluminum.

Platform Sides



$E = 69\ GPa$

$\sigma_y = 27{,}57\ MPa$

$b = \cancel{3t}\ 5\ mm$

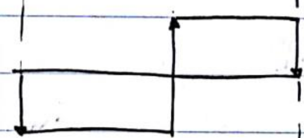
$h = \cancel{5}\ 34\ mm$

$F_A = F_B = F_s / 2$

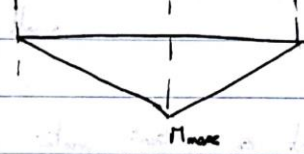
$= 4{,}658\ N$

$F_s = 9{,}315\ N$

$\delta\tau$:



BM↓:



$M_{max} = F_A .\ 55 = 9{,}315.55$

$= 512{,}325\ Nmm$

σ_b:

$$\sigma_b = \frac{M_{max} . y}{I} = \cancel{\frac{M_{max} . h . 12\ 6}{8 b h^3}}$$

$$\cancel{= \frac{512{,}325 . 6}{34^3} = 0{,}078\ MPa}$$

$$= \frac{M_{max} . 6}{b\ h^2} = \frac{512{,}325 . 6}{5 .\ 34^2} = 0{,}532\ MPa$$

δ:

$$\delta_{max} = \frac{PL^3}{48EI} = \frac{F_s .\ 110^3 . \cancel{4}}{4 . \cancel{48} . 69000 . 5 . 34^3} = 0{,}229\ \mu m$$