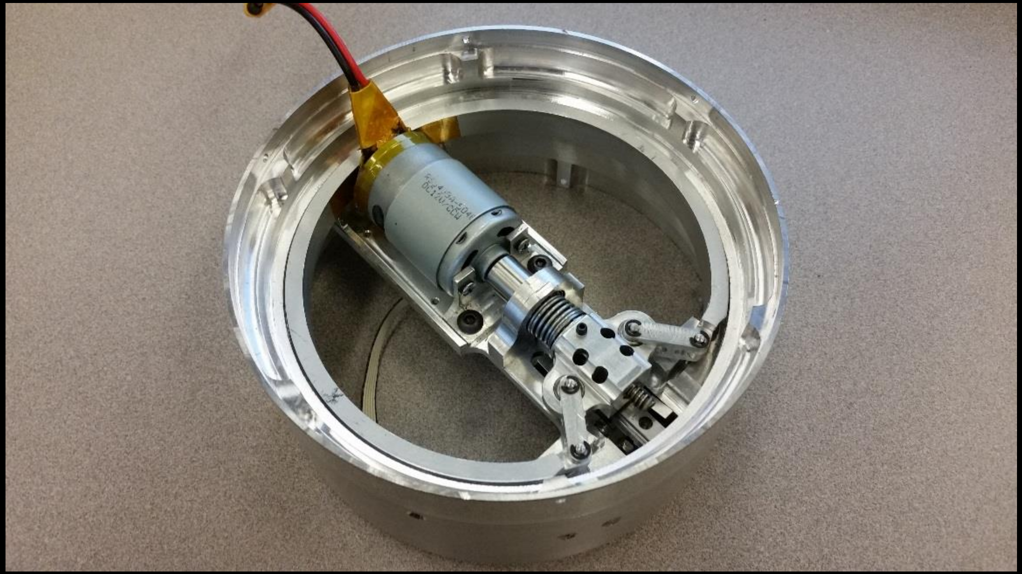
PSAS Nosecone Separation Ring



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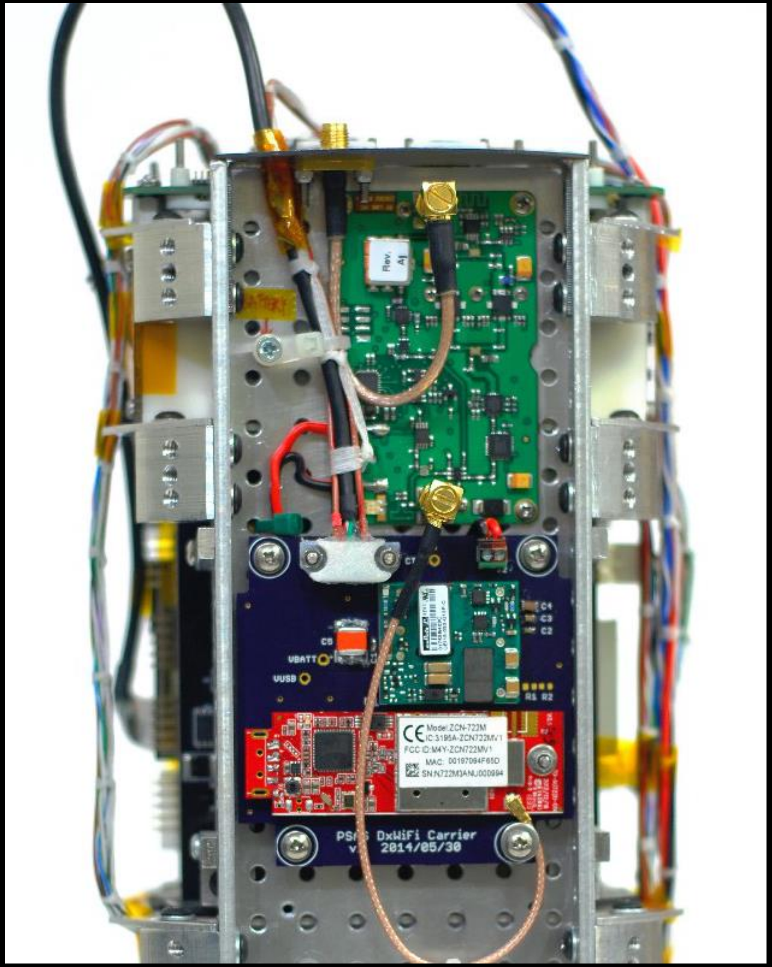
Why This Project is Awesome

- One of only two entirely electro-mechanical NSR devices in existence (to our knowledge)
- This will be the only electro-mechanical NSR device in use by an amatuer rocket club
- Significant challenges in both design and manufacturing
- Used many facets of engineering to complete the project



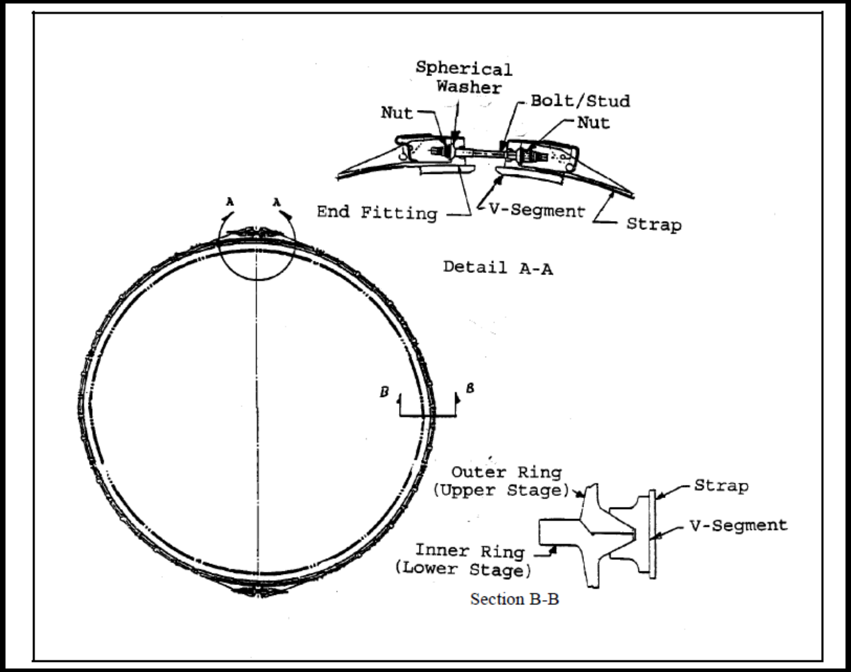
Overview

- Project Details
- Design Challenges
- Design Process
- Prototyping
- Manufacturing
- Conclusions

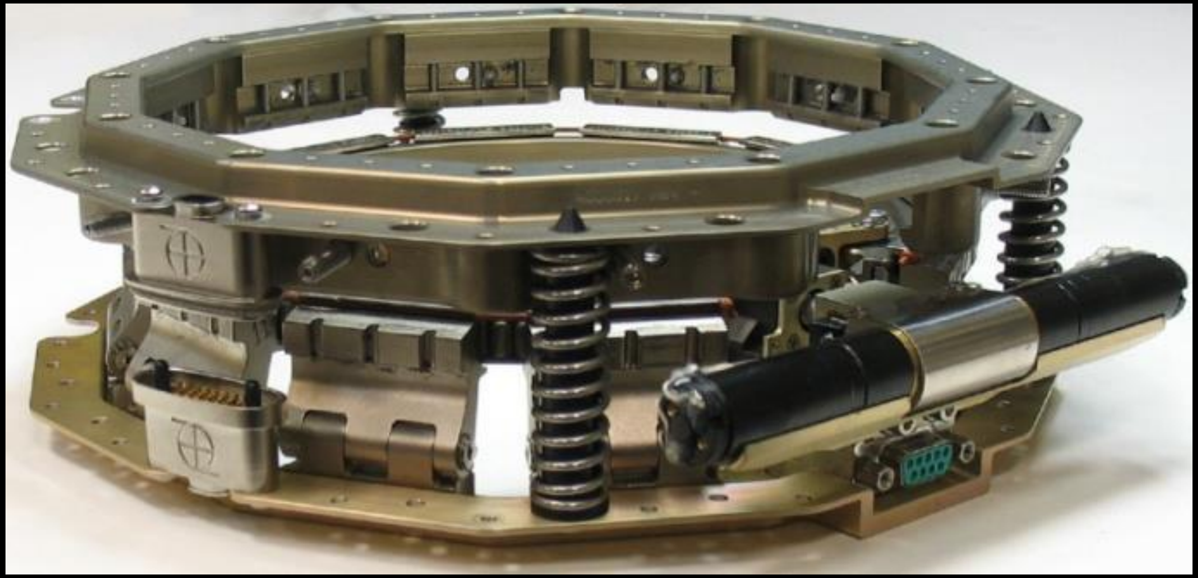
Nosecone Separation

- Deploys parachute
- Previous NSR used cyanoacrylate as an adhesive and gunpowder as the actuator
  - Very difficult to test for reliability
- Failure destroys rocket

Inspiration



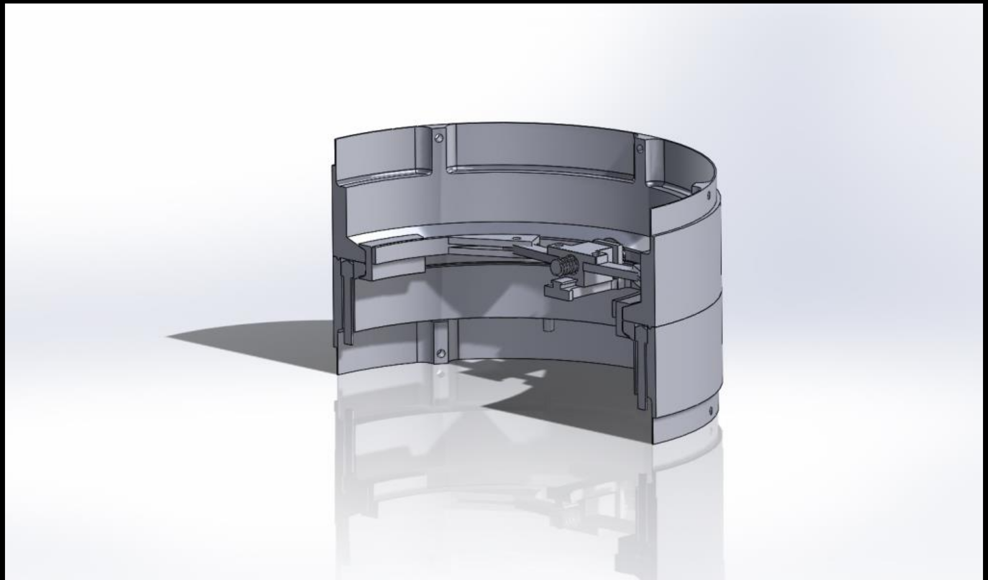
Marmon Clamp



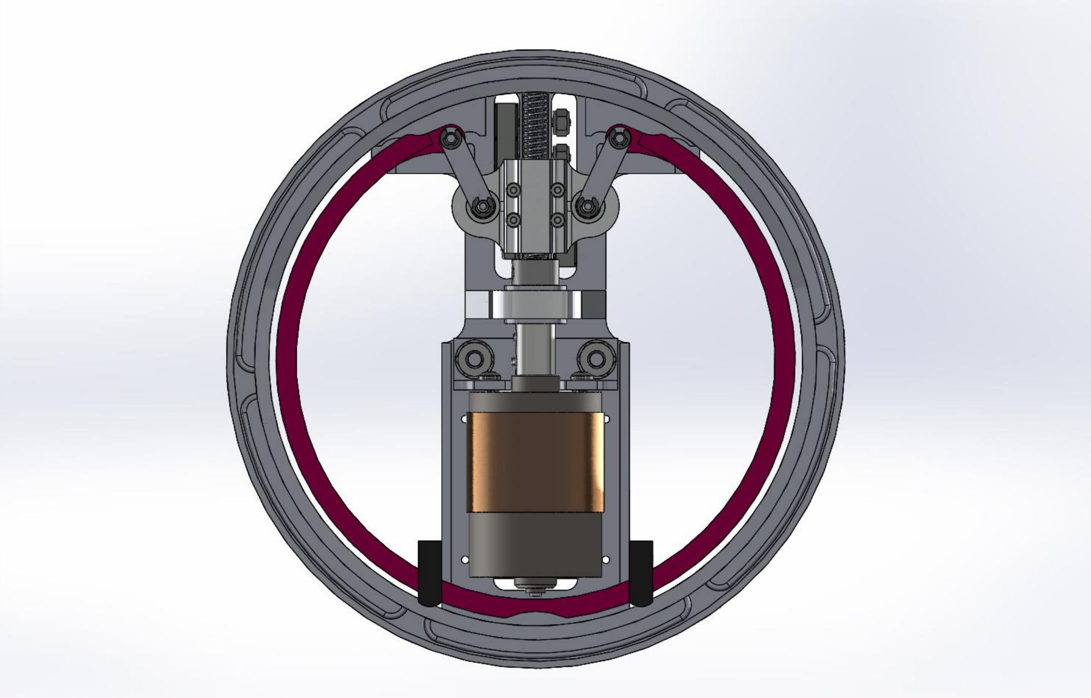
Lightband™

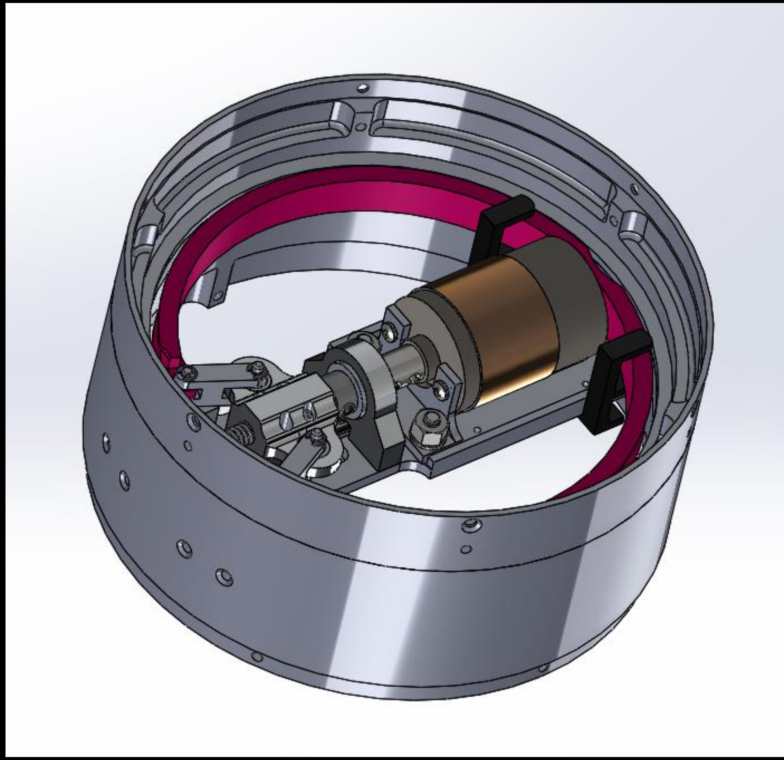
Design Challenges

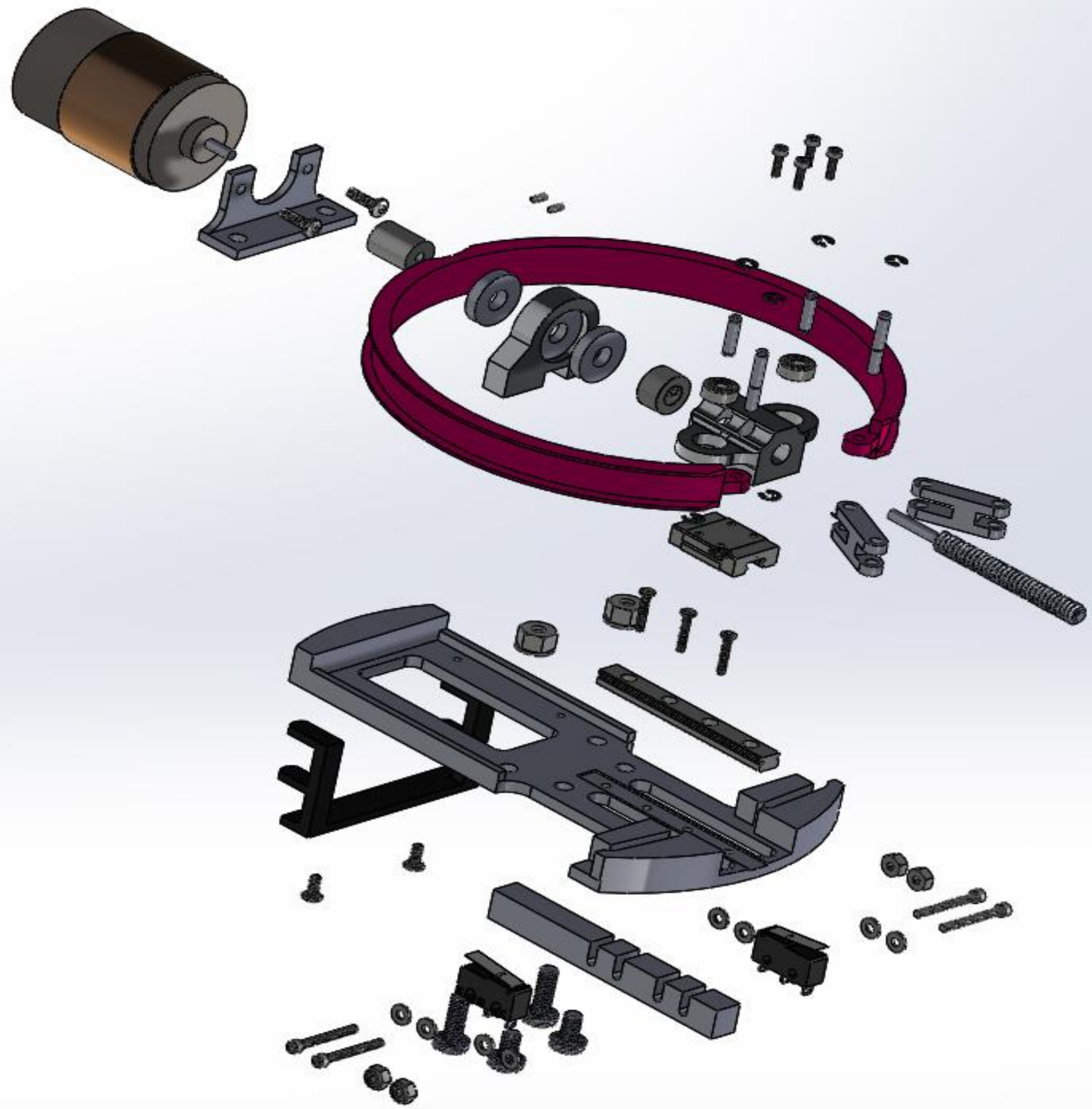
- Entire device must fit inside 6” diameter ring
- Needs to be fast and be robust
- Torque / power requirements
- Isolation of components to avoid axial loading of motor



Final Solid Model



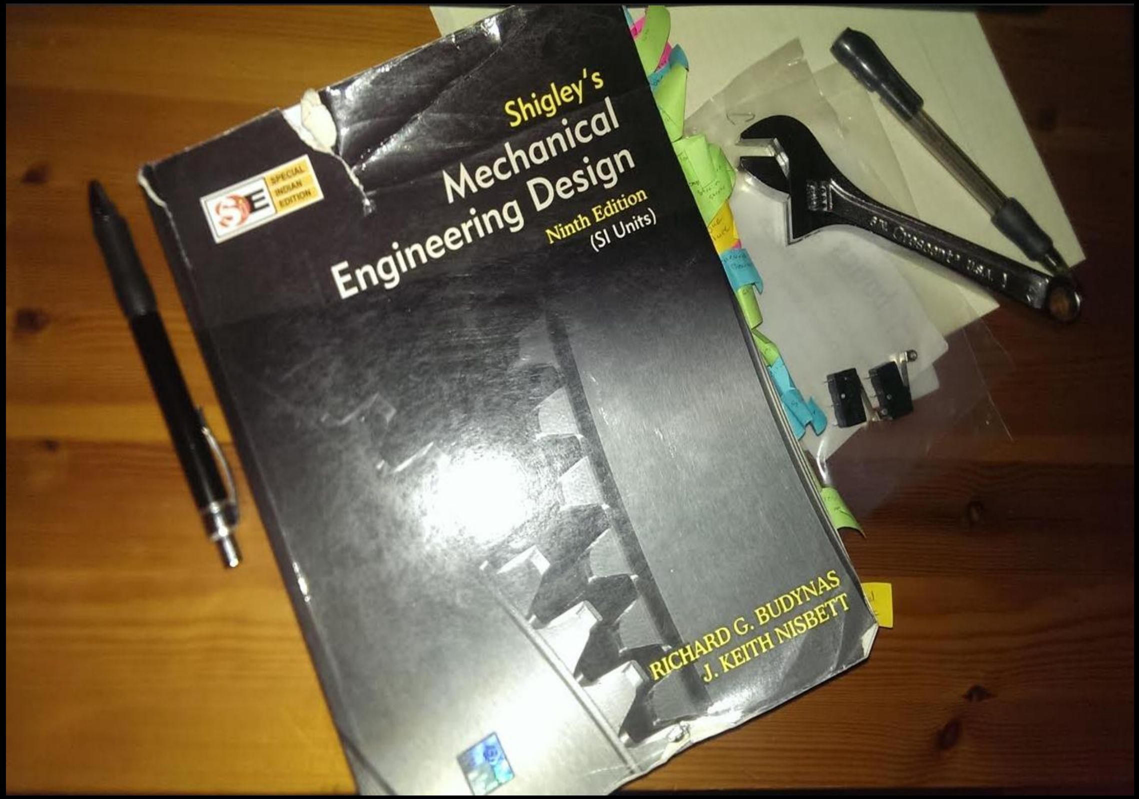




Analysis Based Design

- Used mathematical modeling, FEA, and real world testing on components under high stress loads

- Used statics, dynamics, vibrations, machine design and CAD software to assist in the design of moving parts and support structure



V-Band Optimization

Castiglianos analysis for beam deflection on flex band:

$M \equiv F \cdot R \cdot \cos(\theta)$

$F_\theta \equiv F \cdot \cos(\theta)$

$F_r \equiv F \cdot \sin(\theta)$

$\frac{dF_\theta}{dF} \equiv \cos(\theta)$

$\frac{dF_r}{dF} \equiv \sin(\theta)$

$\frac{dM}{dF} \equiv R \cdot \cos(\theta)$

$\frac{dM \cdot F_\theta}{dF} \equiv 2 \cdot F \cdot R \cdot \cos^2(\theta)$

$U_1 \equiv \int \frac{M^2}{2 \cdot A \cdot e \cdot E} \, d\theta \qquad U_2 \equiv \int \frac{F_\theta^2 \cdot R_{curve}}{2 \cdot A \cdot E} \, d\theta$

$U_3 \equiv \int \frac{M \cdot F_\theta}{A \cdot E} \, d\theta \qquad U_4 \equiv \int \frac{C_c \cdot F_r^2 \cdot R_{curve}}{2 \cdot A \cdot G_{al}} \, d\theta$

because R/h >10, eccentricity does not apply. U1 is then estimated as: $U_1 \equiv \int \frac{M^2 \cdot R_{curve}}{2 \cdot E \cdot I_{y.channel}} \, d\theta$

Assumptions:

$C_c := 1.0$ C is a cross-sectional correction factor, see Table 4-1 in Shigley's, page 163

Material is aluminum T-6061:

$E := 68.9 \cdot GPa \qquad G_{al} := 26 \cdot GPa$

Model properties:

$R_{curve} := \frac{5.21}{2} in = 2.605 \cdot in \qquad A_{beam} := 2 \cdot .15in \cdot .19in = 3.677 \times 10^{-5} m^2$

$e_c \equiv R - r_n$ eccentricity term (not used, see above)

$Disp := 0.4346in$

$Arc_{curve} := 180deg - 25.37deg = 154.63 \cdot deg$ going from 25.37 deg to 180 deg

$I_{y.channel} \equiv \frac{1}{3} \left[2 \cdot s_{channel} \cdot b_{channel}^3 + l_{channel} \cdot t_{channel}^3 + \frac{g_{channel}}{2} \cdot \left(b_{channel}^4 - t_{channel}^4 \right) \right] - A_{channel} \cdot \left(b_{channel} - y_{channel} \right)^2$

$a_{channel} := 0.14in \qquad y_{channel} := 0.2554in \qquad h_{channel} := 0.44in$

$t_{channel} := 0.15in \qquad l_{channel} := 0.37in$

$b_{channel} := 0.29in \qquad n_{channel} := .0704in$

$s_{channel} := 0.0433in \qquad d_{channel} := 2 \cdot y_{channel} = 0.511 \cdot in$

$A_{channel} := d_{channel} \cdot t_{channel} + a_{channel} \cdot \left(s_{channel} + n_{channel} \right) = 0.093 \cdot in^2$

$g_{channel} := \frac{h_{channel} - l_{channel}}{2 \left(b_{channel} - t_{channel} \right)} = 0.25$

$I_{y.channel} := \frac{1}{3} \left[2 \cdot s_{channel} \cdot b_{channel}^3 + l_{channel} \cdot t_{channel}^3 + \frac{g_{channel}}{2} \cdot \left(b_{channel}^4 - t_{channel}^4 \right) \right] - A_{channel} \cdot \left(b_{channel} - y_{channel} \right)^2 = 1.283 \times 10^{-3} \cdot in^4$

setup of problem:

$U_{tot} \equiv U_1 + U_2 + U_3 + U_4$ from castiglianos: $Disp \equiv \frac{\delta U_{tot}}{\delta F}$

$$Disp \equiv \int \frac{M \cdot R_{curve}}{2 \cdot E \cdot I_{y.channel}} \cdot \left(\frac{dM}{dF} \right) d\theta + \int \frac{F_\theta \cdot R}{A \cdot E} \cdot \left(\frac{dF_\theta}{dF} \right) d\theta + \int \frac{1}{A \cdot E} \cdot \left(\frac{d(M \cdot F_\theta)}{dF} \right) d\theta + \int \frac{C_c \cdot F_r \cdot R}{A \cdot G} \cdot \left(\frac{d \cdot F_r}{dF} \right) d\theta$$

$$Disp \equiv \frac{F \cdot R^3}{2 \cdot E \cdot I_{y.channel}} \cdot \int_{25.37}^{180} \cos^2(\theta) \, d\theta + \frac{F \cdot R}{A \cdot E} \cdot \int_{25.37}^{180} \cos^2(\theta) \, d\theta + \frac{2 \cdot F \cdot R}{A \cdot E} \cdot \int_{25.37}^{180} \cos^2(\theta) \, d\theta + \frac{F \cdot R}{A \cdot G} \cdot \int_{25.37}^{180} \sin^2(\theta) \, d\theta$$

Solving integral term: $\int_{25.37}^{180} \cos^2(\theta) \, d\theta = 77.4405 \qquad \int_{25.37}^{180} \sin^2(\theta) \, d\theta = 77.1895$

$$Disp \equiv 77.4405 \cdot \left(\frac{F \cdot R^3}{2 \cdot E \cdot I_{y.channel}} + \frac{F \cdot R}{A \cdot E} + \frac{2 \cdot F \cdot R}{A \cdot E} \right) + 77.1895 \cdot \left(\frac{F \cdot R}{A \cdot G} \right)$$

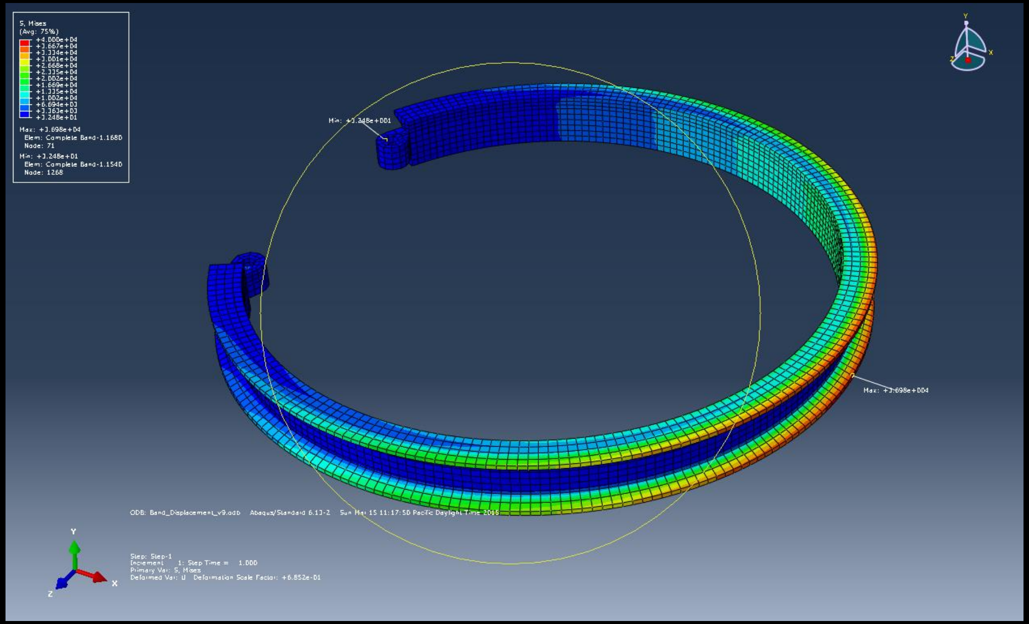
factor out F from all terms, solve for F with known displacement:

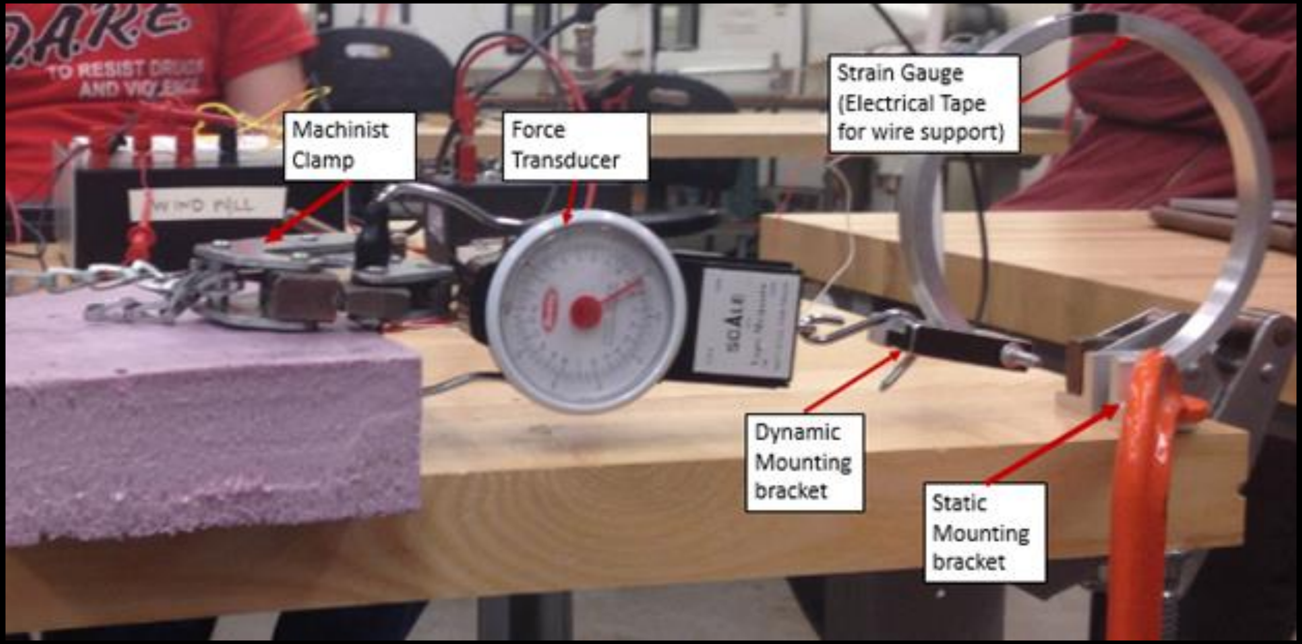
$$Disp \equiv F \cdot \left[77.4405 \cdot \left(\frac{R^3}{2 \cdot E \cdot I_{y.channel}} + \frac{R}{A \cdot E} + \frac{2 \cdot R}{A \cdot E} \right) + 77.1895 \cdot \left(\frac{R}{A \cdot G} \right) \right]$$

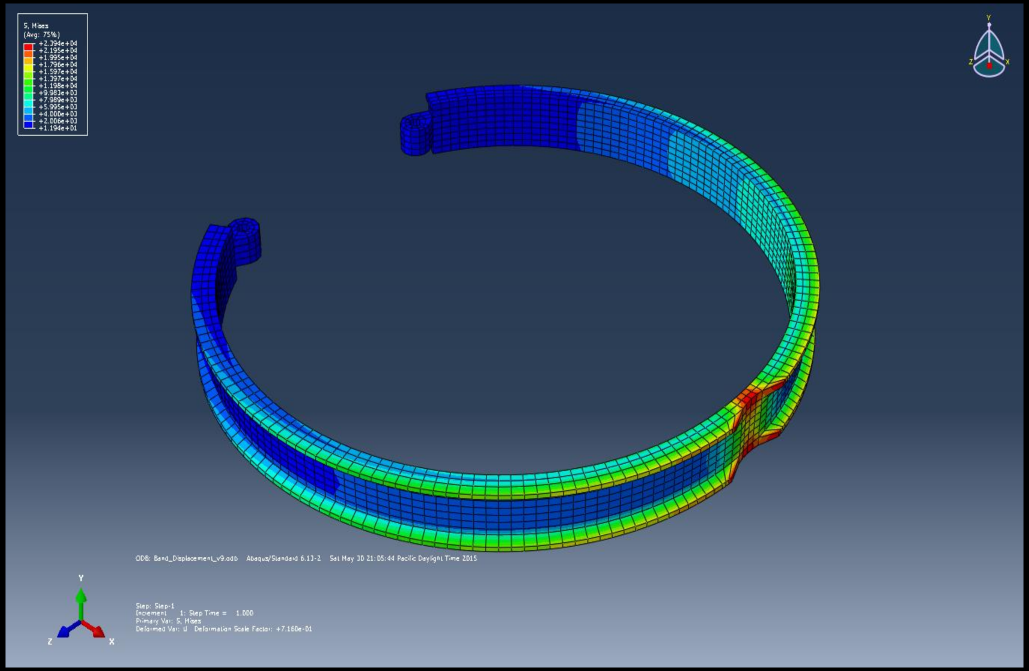
$$F_1 := \frac{Disp}{\left[77.4405 \cdot \left(\frac{R_{curve}^3}{2 \cdot E \cdot I_{y.channel}} + \frac{R_{curve}}{A_{beam} \cdot E} + \frac{2 \cdot R_{curve}}{A_{beam} \cdot E} \right) + 77.1895 \cdot \left(\frac{C_c \cdot R_{curve}}{A_{beam} \cdot G_{al}} \right) \right]} = 34.908 \cdot N$$

$\boxed{F_1 = 7.848 \cdot lbf}$

Model	Stress (psi)	Force Predicted (lbf)
Mathematical (Original)	NA	18.2
Mathematical (Optimized)	NA	15.68
Experimental	37520	20.6
FEA (Original)	36980	20.6
FEA (optimized)	22866	16

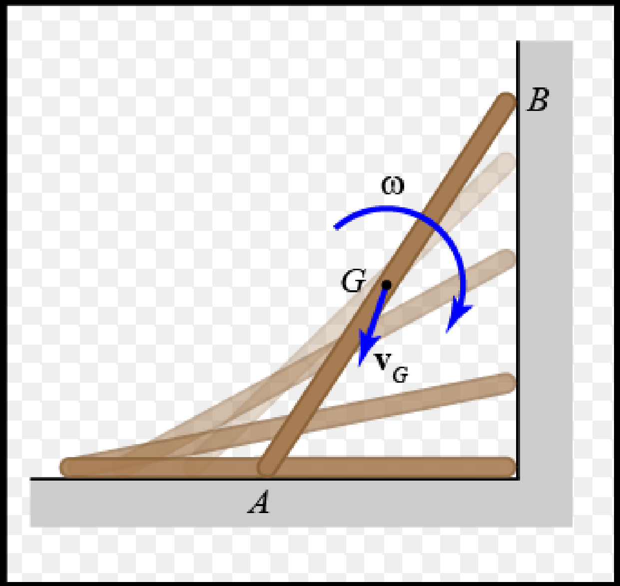


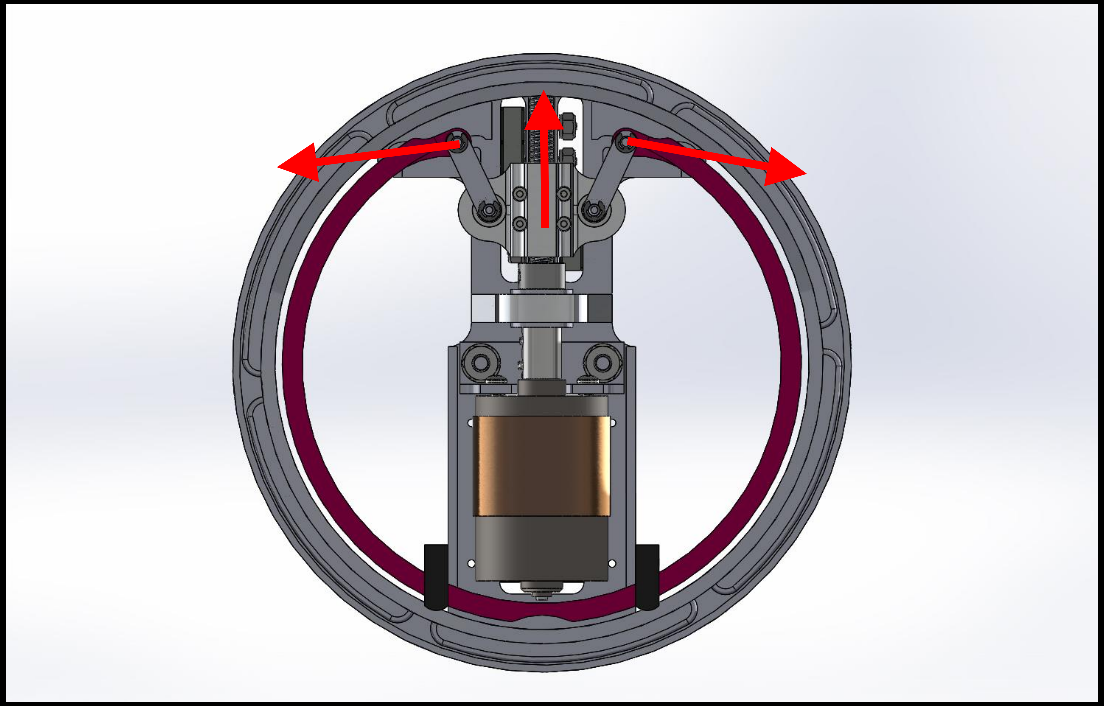


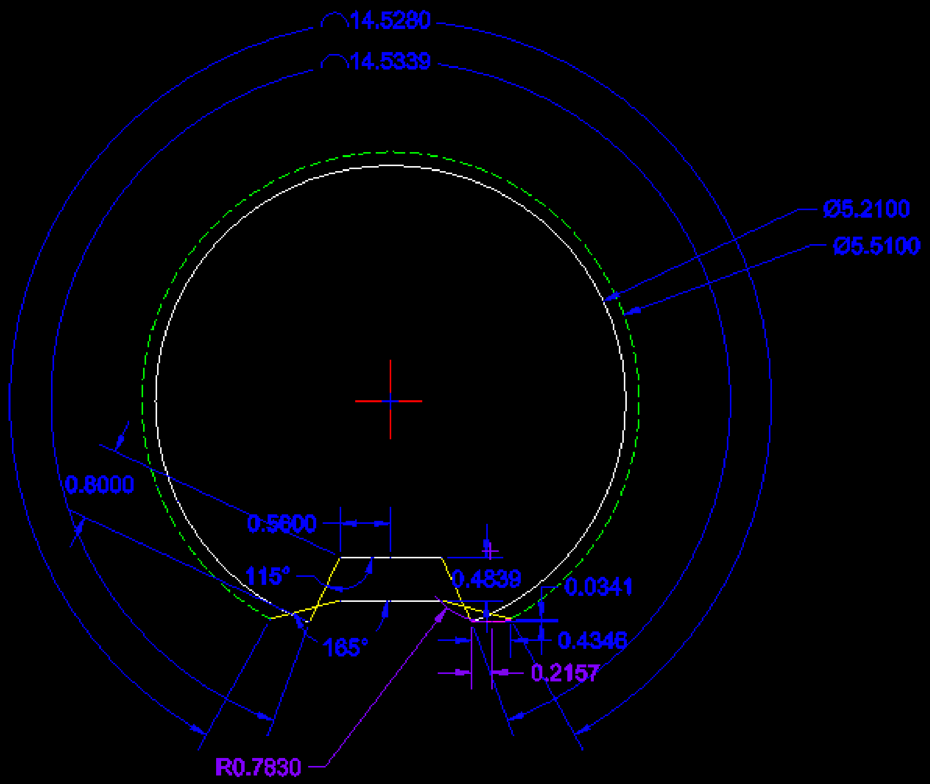


Design of Dynamic Action

- Model using rigid body kinematics
- End point (A) is equal to distance traveled by (B) plus rotation about part center





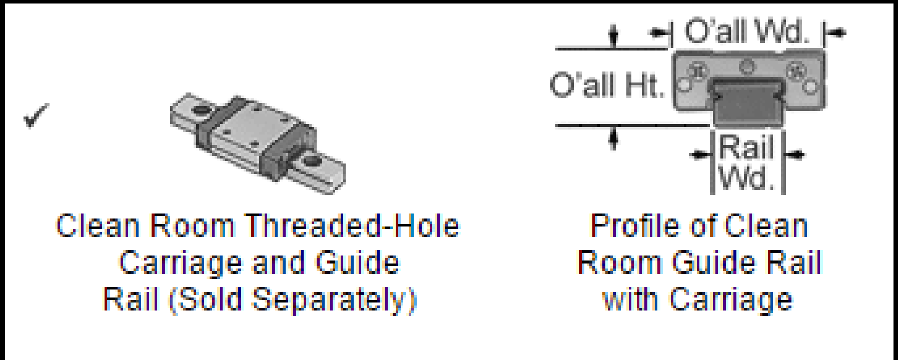


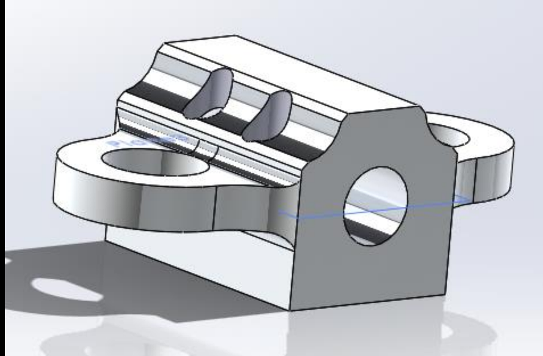
Linear Motion Constraint

- Linear motion obtained using Guide rail and Power screw

- Guide rail assembly eliminates most friction and has 300 lbf dynamic load capacity

- Used needle thrust bearings to prevent transmission of axial load to motor





Power Requirements

$$F_{band} = 0:10$$

$$\theta_{arm} = 25.37°:75°$$

$$F_{arm} = \frac{F_{band}}{\sin(\theta_{arm})}$$

$$F_{sled} = \frac{F_{arm}}{\cos(\theta_{arm})}$$

Power screw linear force translation:

Governing equation:

$$T_R = \frac{F_{axial} \cdot d_m}{2} \cdot \left[\frac{(P + \pi \cdot f_{thr} \cdot d_m \cdot \sec(\alpha))}{[\pi \cdot (d_m - f_{thr}) \cdot P \cdot \sec(\alpha)]} \right] + \frac{F_{axial} \cdot f_c \cdot d_c}{2}$$

Axial force is taken as F1 from castiglianos beam deflection analysis multiplied by the cosine of the angle the force is being applied at (0 force is translated to the threads when the relative motion arms are perpendicular to the power screw, worst case scenario is that both arms are at 45 degree angle to the sled):

$$F_{axial} := 2 \cdot 68.573 \text{lbf} \cdot \sin(45\text{deg}) = 96.977 \cdot \text{lbf}$$

Power screw properties:

Assuming a 3/8x16 ball thread powerscrew for linear torque transmission:

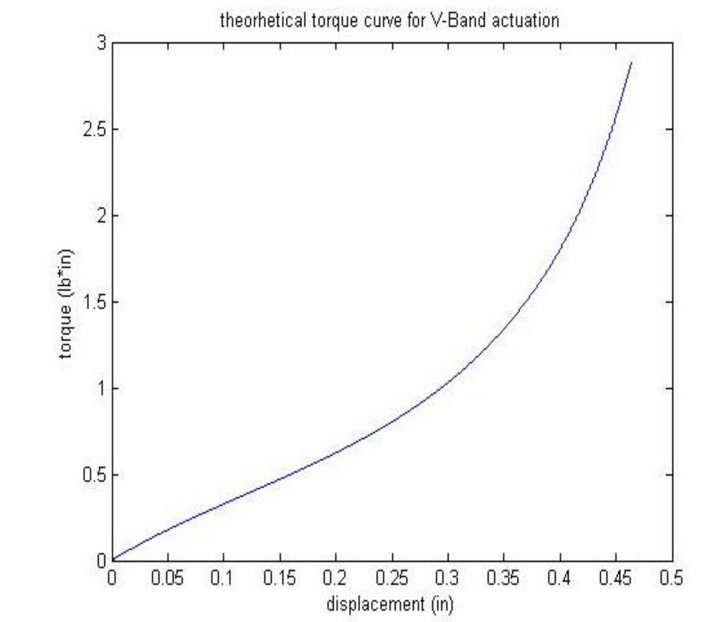
$$dm = d - \frac{P}{2} \qquad P := \frac{1}{16} \text{in} \qquad \alpha := 30\text{deg}$$

$$d_m := \frac{1}{4} \text{in} - \frac{P}{2} = 0.219 \cdot \text{in}$$

$f_{thr} := 0.2$ conservative assumption for unlubricated threads

Assume no collar is present (power screw interfaces directly with drive-nut attached to sled:

$$T_R := \frac{F_{axial} \cdot d_m}{2} \cdot \left[\frac{(P + \pi \cdot f_{thr} \cdot d_m \cdot \sec(\alpha))}{((\pi \cdot d_m - f_{thr} \cdot P \cdot \sec(\alpha)))} \right] = 3.487 \cdot \text{lbf} \cdot \text{in}$$



Screw will be self locking as pi*fthr*dm=~1.5>>1/16

Motor Selection

	A	B	C	D	E	F	G	H	I
1	Requirements:								
2			mNm	lbf in					
3		T (ideal)	565	5				1.97	
4		T (min)	249	2.2					
5									
6		RPM (min)	2000						
7		RPM (ideal)	5000+						
8									
9	Comparison:								
10									
11	Vendor	Part #	Dia (mm)	Length (mm)	length (in)	Voltage (V)	Power (W)	Stall Torque (mN*m)	no load speed
12	Maxon	226785	29	44.7	1.788	12	22	299	9350
13	Maxon	448595	30	68	2.72	12	15	342	2870
14	Micromo	3257G012CR	32	57	2.28	12		547	5400
15	Micromo	2657W012CR	26	57	2.28	12		286	6400
16	Micromo	2657W012CXR	26	57	2.28	12		3.7	5800
17	Pittman	14205 (DC054B-5	52	113	4.524248	12	58	1485	2990
18	Pittman	9237 (DC040B-6)	40	85	3.406648	12	34	461	5210
19	Pittman	9236 (DC040B-5)	40	78	3.101848	12	26	387	4730
20	Pittman	14201 (DC054B-1	52	75	3.000248	12	24	431	4140
21	AndyMark	9015	25	57	2.27584	12	180	428	16000
22	BanBots	RS-550	39	57	2.27584	12	180	498	19300

stall torque = 498 mNm

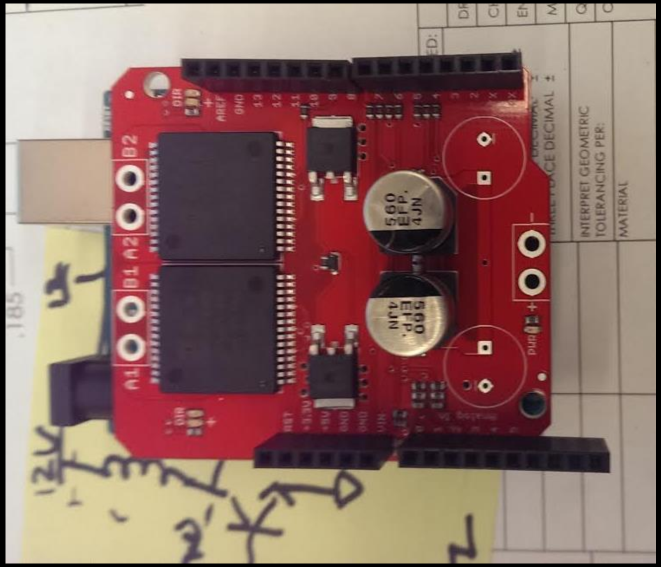
speed = 19300 RPM
(no load)

dimensions fit nominal for direct drive

total cost: $7.00 (does not include shipping and handling)

Electrical System (Prototype)

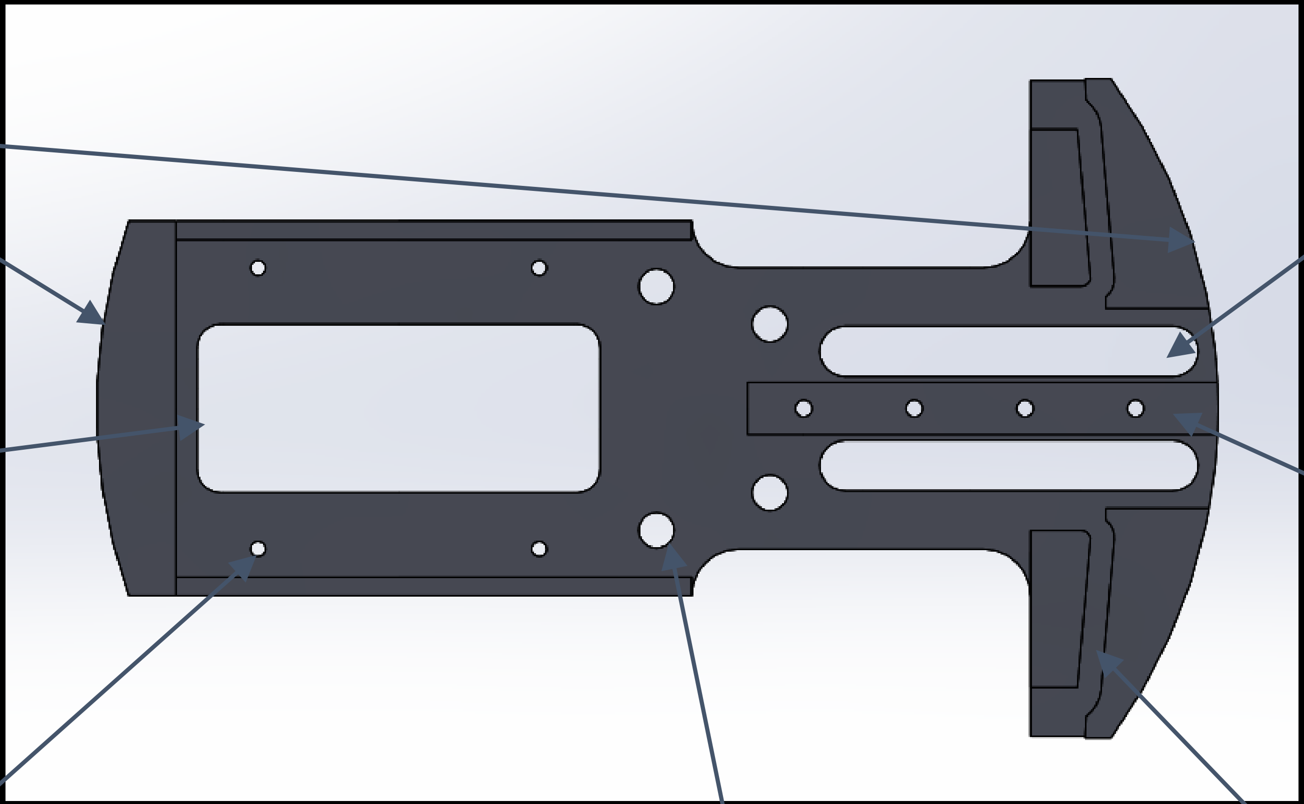
- Motor controlled by PWM via Arduino equipped with 30A H Bridge
- Push button operation (for display)
- Limit switches used as fore and aft stops
- Proprietary control board which interfaces with flight computer will be developed for use in LV3 vehicle



How to Incorporate it All?

-.002/-.004
Diameter Control
Surface and
Mounting Holes

Limit Switch
Channels



Motor Profile Relief
Cut

Precision Rail
Slot and
Fixturing Holes

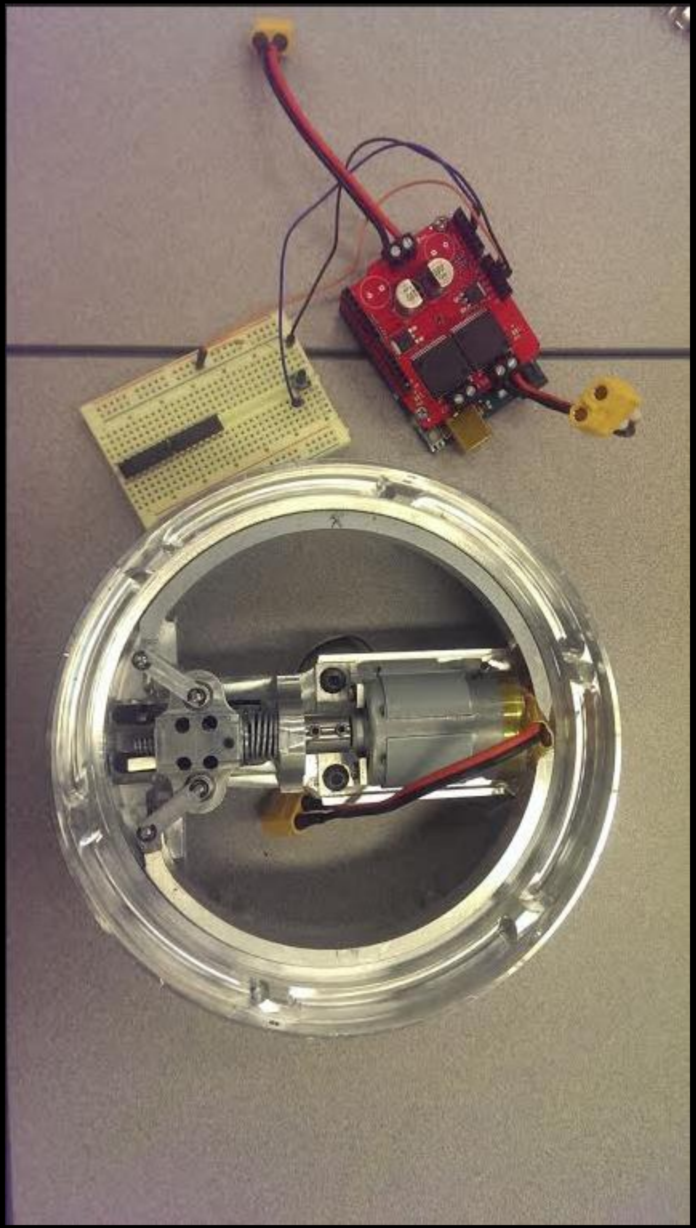
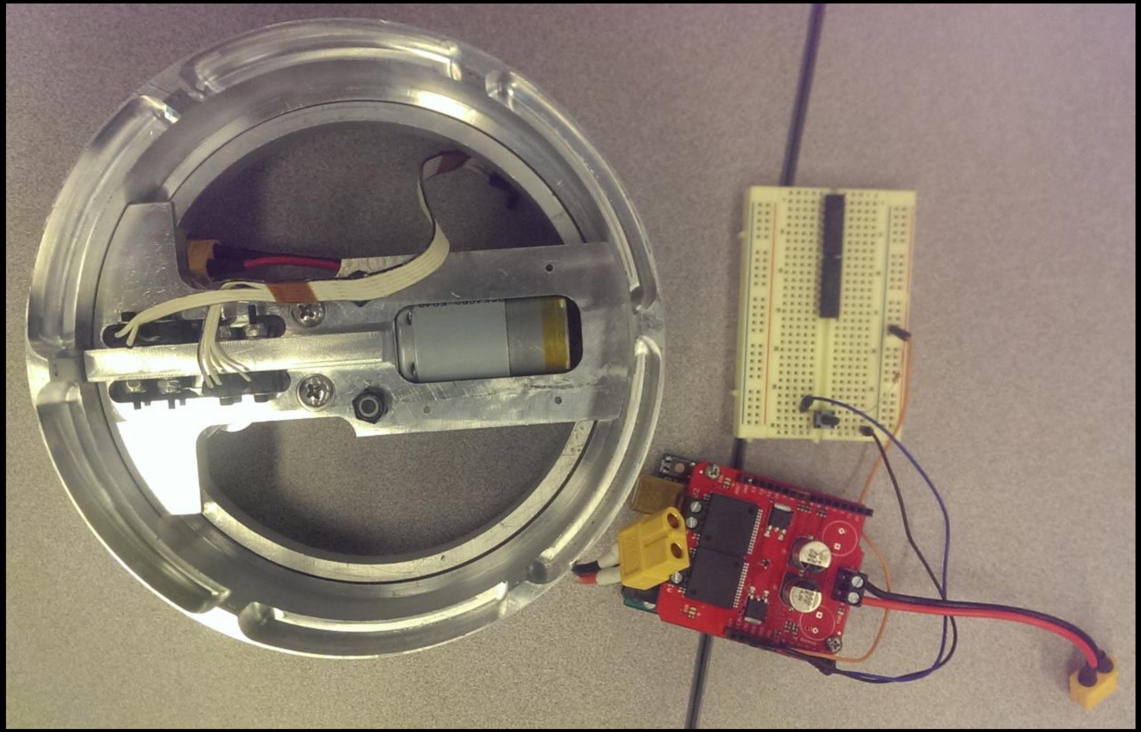
1.5” x 1.5” 4-40 UNF-2B
Mounting Hole Pattern
for Electronics

Motor and Journal
Mounting Holes

Relative Motion
Guide Channels

Design Notes

- All parts designed with ½" or ¼" tooling in mind (where allowable) for manufacturing
- Linear actuator parts must hold tight tolerancing to eliminate unwanted play. This was achieved by sizing and cutting the parts to each other
- Custom couplers and pins were manufactured due to space constraints

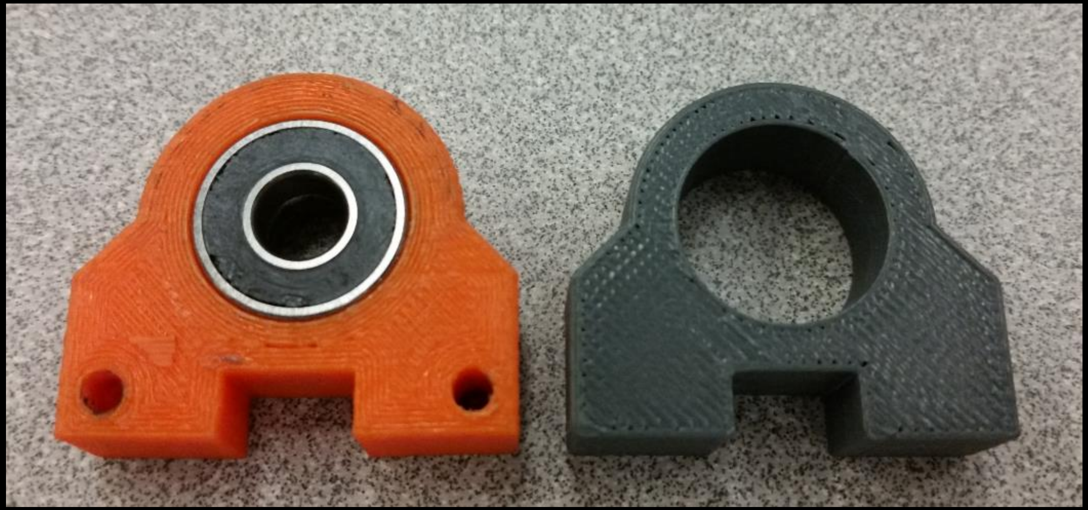



Prototyping: Coupling Rings

V-Band

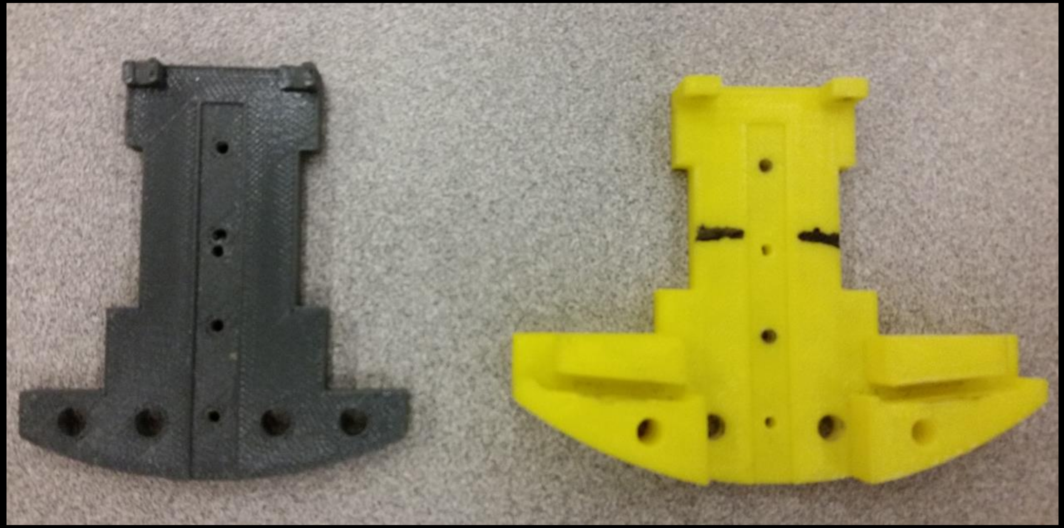
Carriage

Journal



Arms

Baseplate

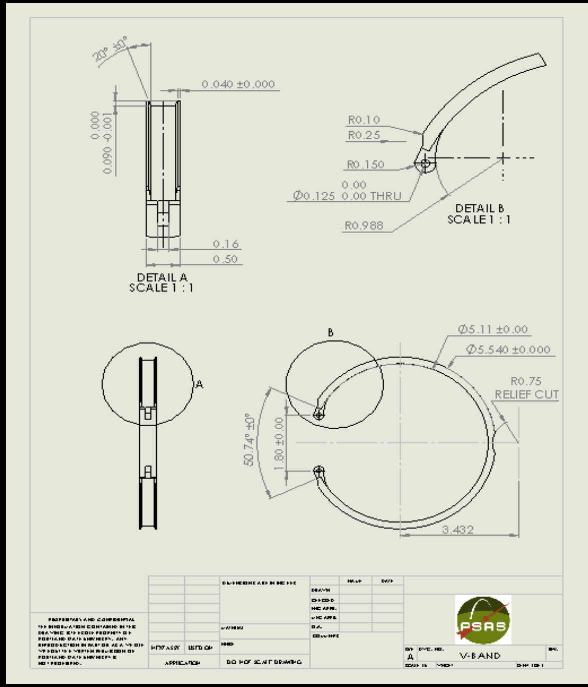



MACHINERY'S
HANDBOOK
SEVENTEENTH EDITION

Machining

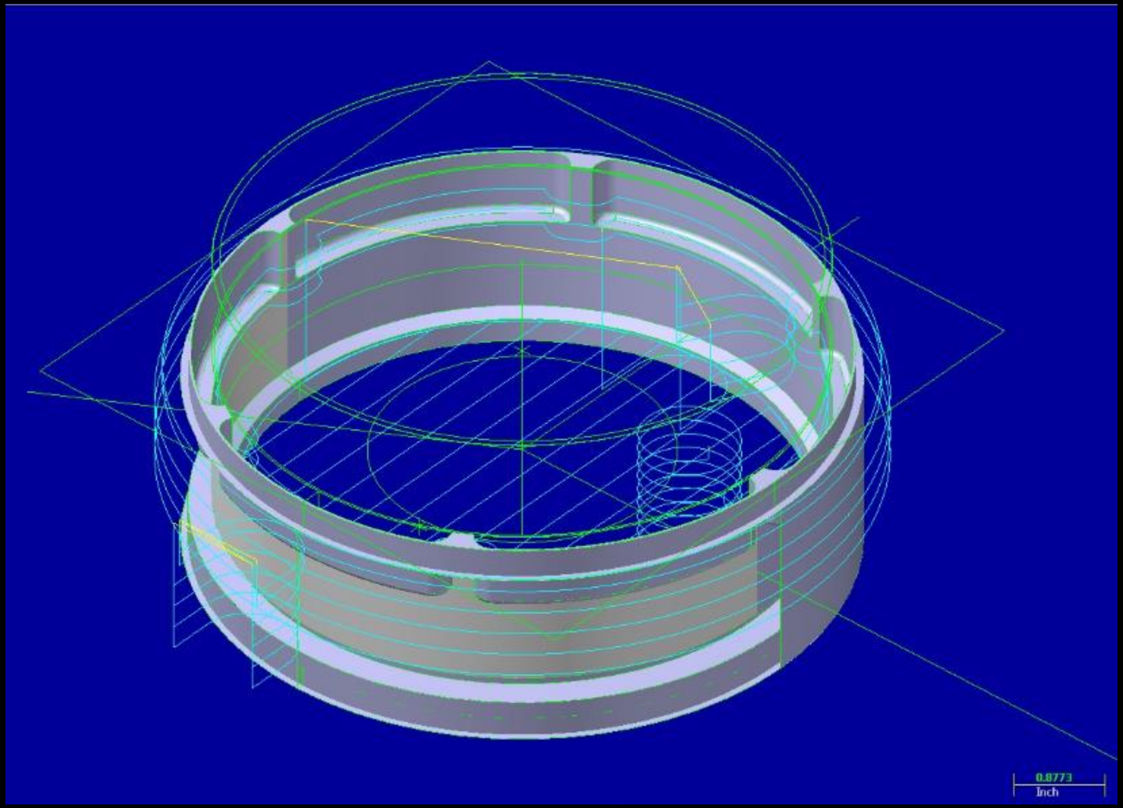
V-Band

- Used 2-axis CNC to mill OD
- Turned 20deg clamping surface with custom tool
- Cut ID and relief cut with 2-axis



Large and Small Rings

- Used Mastercam to define tool paths and write G code.
- Machined using 3-axis Haas mill
- Started with solid aluminum stock



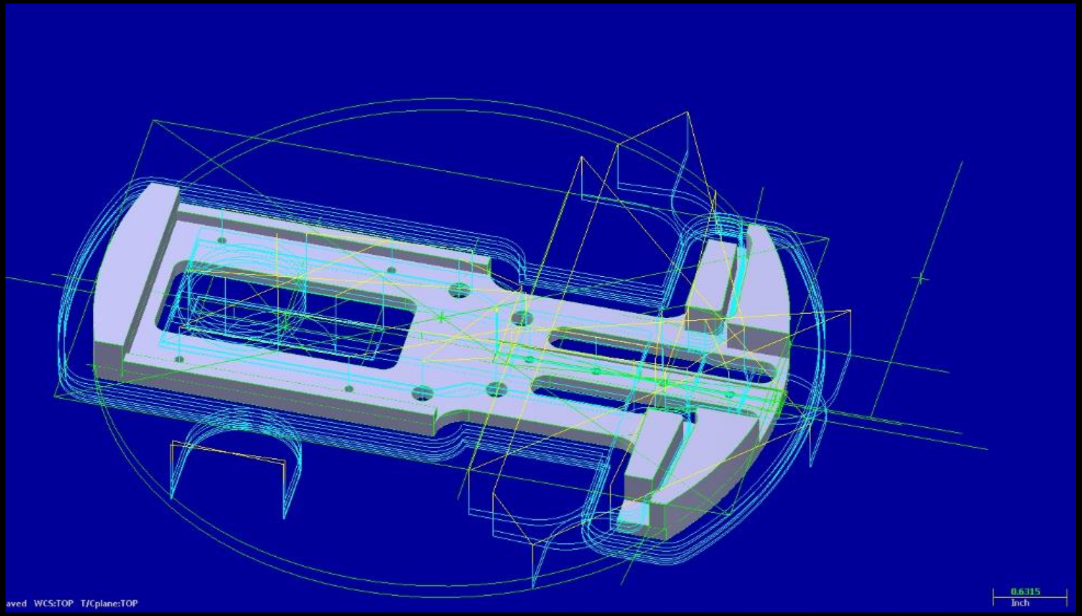
Challenges:

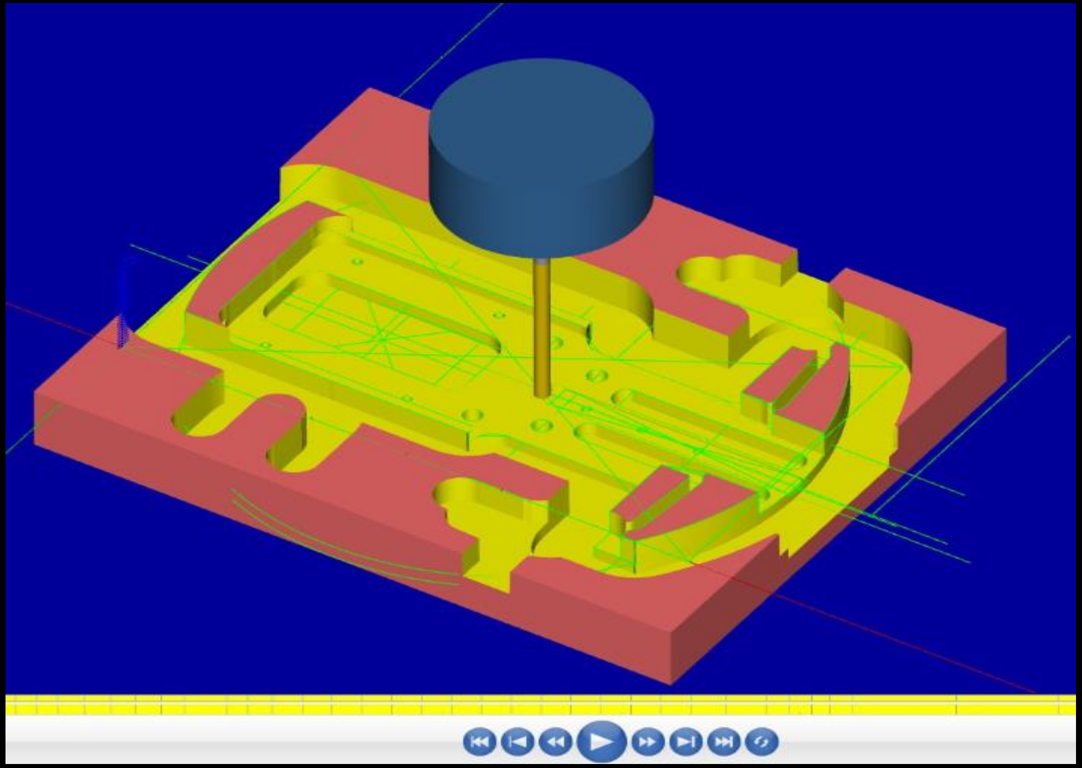
- 20 deg chamfer to interface with v-band
- 1/8” fillet
- Flipped part over several times


KURT

Baseplate

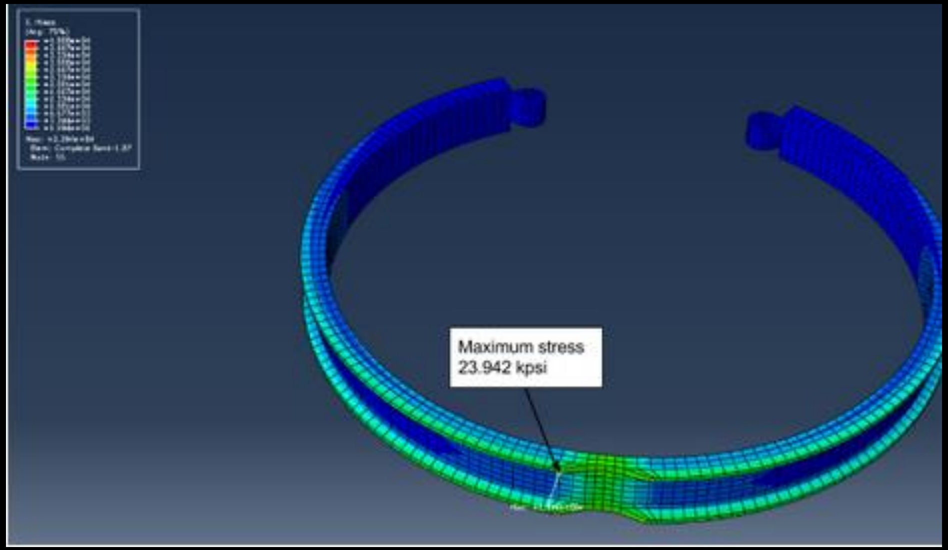
- Machined on 3-axis Haas from 1/2” plate
- Significant challenges in fixturing
- 1/8” and .110” channels





Fatigue

- PDS Requirement: Mechanical failure of less than 1%.
- 24 kpsi is maximum stress on the system.
- Fatigue analysis shows V-band could be actuated 44,000 times.



Conclusions

- Mechanical components are rocket ready
- Design can withstand a 250 lbf tensile load required by PDS
- Actuates in ~63 milliseconds
- Needs more robust electrical system

Questions?