Appendix E: Supporting Analysis

4/8/13

Buckling analysis of needles

Approximate as fixed - free column

$C = \frac{1}{4}$

$$P_{cr} = \frac{C \pi^2 E I}{\ell^2}$$

$$P_{cr} = \frac{\frac{1}{4} \pi^2 E I}{\ell^2}$$

↑ Find geometry that would buckle under a 2 lb load (minimum material)

$$I = \frac{\pi D^4}{64} \qquad P_{cr} = \frac{\frac{1}{4} \pi^3 E D^4}{64 \ell^2}$$

$$\boxed{P_{cr} = \frac{16 \pi^3 E D^4}{\ell^2}}$$

$$\ell_{max} = \sqrt{\frac{16 \pi^3 E D^4}{P_{cr}}}$$

Stainless steel $E = 180$ GPa

Titanium $E = 16 \times 10^6$ psi

Try $D = 0.06$ in, $P_{cr} = 2$ lb, $E = 180$ GPa $= 26.1 \times 10^6$ psi

$$\ell_{max} = \sqrt{\frac{16 \pi^3 (26.1 \times 10^6 \text{ lb/in}^2)(0.06 \text{ in})^4}{2 \text{ lb}}}$$

$\ell_{max} = 289.7$ in — will never buckle if $\ell \cong 2$ in

Solve for d given $\ell = 2$ in

$$D_{min} = \left(\frac{P_{cr}\,\ell^2}{16\pi^3 E}\right)^{1/4}$$

$$D_{min} = \left(\frac{(2\text{ lb})(2\text{ in})^2}{16\,\pi^3\,(26.1\times10^6\ \frac{\text{lb}}{\text{in}^2})}\right)$$

$D_{min} = 0.00498$ in $= 0.127$ mm (for solid cylinder)

4/3/13

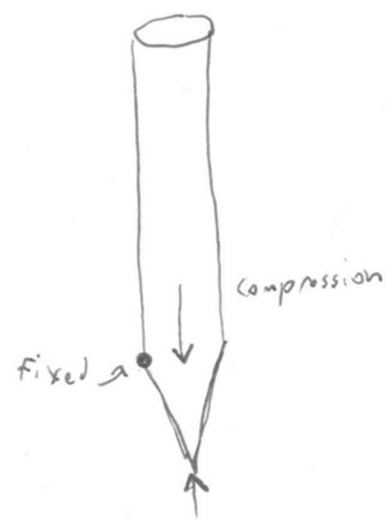
Needle Fatigue Analysis

Compression – no crack growth so no fatigue

Tension – minimal tension – need to measure this force

Bending – could have bending depending on design

Assume following geometry
(worst case ?)



Whole needle will always be in compression. Look at buckling next.

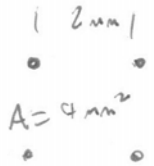
Release force calc

4/18/13

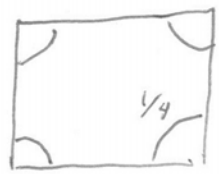
\# of needles – proportional to pit area

pit area = $\pi(1\,in)^2 = 3.1416\ in^2$

$3.1416\ in^2 = 2026.83\ mm^2$



1 needle per $4\,mm^2$



1/4 needle × 4 = 1 needle

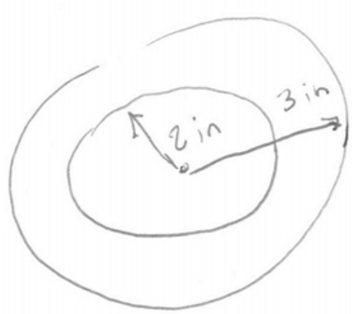
so # needles = $\dfrac{2026.83\ mm^2}{4\ mm^2} \cong 507$ needles

retraction force = 0.17 lb

total retraction force = $0.17\ lb\ (507) = 86.14\ lb$

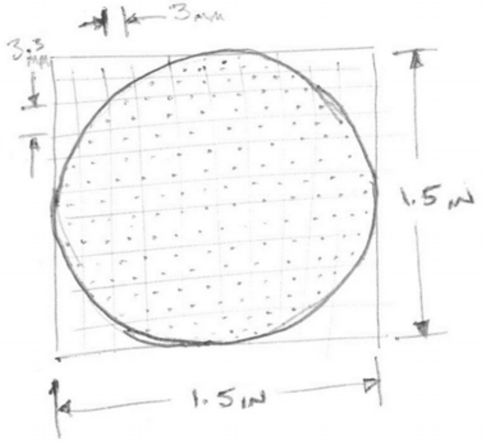
Area of ring = $\pi(1.5\,in)^2 - \pi(1\,in)^2 = 3.927\ in^2$

pressure = $\dfrac{F}{A} = \dfrac{86.14\ lb}{3.927\ in^2} = 21.94\ psi$



Release Ring

PRESSURE SENSOR FINAL NOTES 03/09



NEEDLE ARRAY

2MM x 2MM CIRCULAR ARRAY

SENSOR MATRIX

3MM x 3.3MM RECTANGULAR ARRY

1. FIND # OF SENSOR ELEMENTS IN CONTACT W/ NEEDLE ARRAY
   - WE CAN IGNORE SENSOR ELEMENTS WHICH ARE NOT PART OF PRESSURE SENSING SYSTEM
   - NOTE: TO BEGIN, USE RECTANGULAR OUTER AREA TO SIMPLIFY MULTIPLEXING LOGIC. WILL REFINE LATER IF NEEDED

$$A_{TOT} = (1.5\,in)(1.5\,in)\left(\frac{25.4\,mm}{in}\right)^2$$

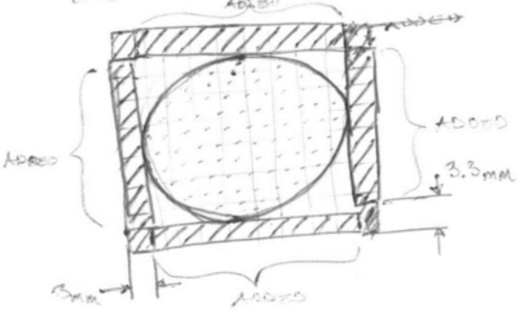
$$A_{TOT} = 1452\ mm^2$$

USE $3\,mm \times 3.3\,mm = 10\,mm^2$ / SENSOR RESOLUTION TO FIND # OF SENSORE ELEMENTS IN THIS AREA

$$\frac{A_{TOT}}{A_{SENS}} = \frac{1452\ mm^2}{10\ mm^2/sensor} = \#\ \text{SENSOR ELEMENTS}$$

\# SENSOR ELEMENTS = 145

2. NOW, REDO 1, BUT INCLUDE AN EXTRA STRIP OF SENSOR ELEMENTS ALONG THE TOP, BOTTOM, & SIDES, IN CASE NEEDLES AT THE EDGE LIE BETWEEN SENSOR ELEMENTS



~~$A = (1.5_{in})(1.5_{in})\left(25.4\frac{mm}{in}\right)^2 + 2(3mm)(1.5_{in})\left(\frac{25.4mm}{in}\right) + 2(3.3mm)(1.5$~~

$$A_{TOT} = (1.5_{in})(1.5_{in})\left(25.4\frac{mm}{in}\right)^2 + 2(3mm)(1.5_{in})\left(25.4\frac{mm}{in}\right) + 2(3.3mm)(1.5_{in})\left(25.4\frac{mm}{in}\right) + 4(3mm)(3.3mm)$$

$$A_{TOT} = 1452\,mm^2 + 2(1.5_{in})\left(25.4\frac{mm}{in}\right)[3mm + 3.3mm] + 39.6\,mm^2$$

$A_{TOT} = 1995\,mm^2$

AGAIN, USING $3mm \times 3.3mm = 10\,mm^2/\text{sensor}$ RESOLUTION

$$\frac{A_{TOT}}{A_{SEN}} = \frac{1995\,mm^2}{10\,mm^2/\text{sensor}} = \#\text{ SENSOR ELEMENTS}$$

SENSOR ELEMENTS = 200

~~ROUNDING UP TO A NICE EVEN #~~

~~# SENSOR ELEMENTS = 200~~

NOTE ON ADC:

~~PARA~~

THROUGHPUT:

PARALLEL: 500 KSps

SERIAL: 350 KSps

FULL RANGE READING THROUGHPUT:

PARALLEL

$\frac{500k}{200} =$ 2.5 kHz = 400 μS

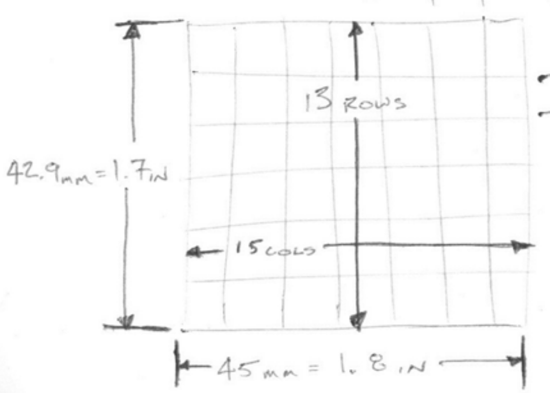
SERIAL

$\frac{350k}{200} =$ 1.75 kHz = 571 μs

3. NOW, IF THE SENSORS WERE EVENLY DISTRIBUTED, CALCULATING THE DIMENSIONS IN TERMS OF # OF SENSORS WOULD BE A BREEZE. HOWEVER, THEY ARE NOT. USING WEIGHTED VALUES OF 3mm SIDE & 3.3mm SIDE.

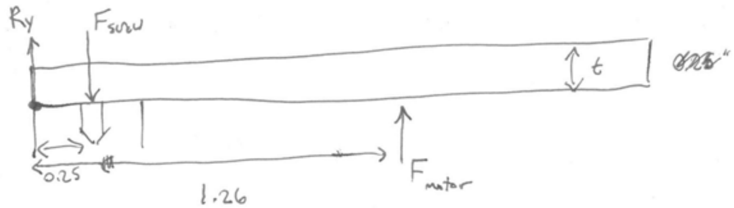
$$\left(\frac{3.3}{3.3+3}\right)\sqrt{200} = 15.55$$

$$\left(\frac{3}{3.3+3}\right)\sqrt{200} = 12.86$$



USE 13 x 15 ARRAY FOR 195 TOTAL SENSOR ELEMENTS

Bracket Bending



$F_{motor} = 21$ lb

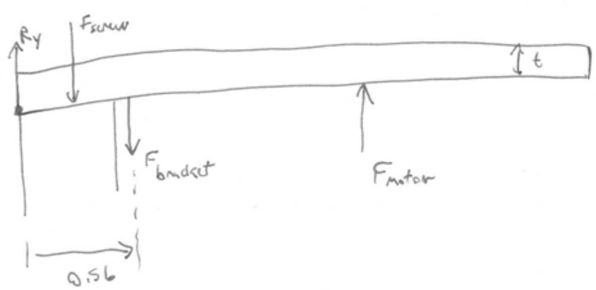
$\Sigma M_R = -F_{screw}(0.25) + F_{motor}(1.26) = 0$

$F_{screw} = F_{motor}\left(\frac{1.26}{0.25}\right)$

$F_{screw} = 105.84$ lb

per screw = 52.92 lb

with reinforcement



$\Sigma M_R = -F_{screw}(0.25) - F_{bracket}(0.56) + F_{motor}(1.26) = 0$

Assume $R_y = 0$ (worst case)

$\Sigma F_y = -F_{screw} - F_{bracket} + F_{motor} = 0$

2 eq, 2 unknowns

$$F_{screw} = \frac{F_{motor}(1.26) - F_b(0.56)}{0.25}$$

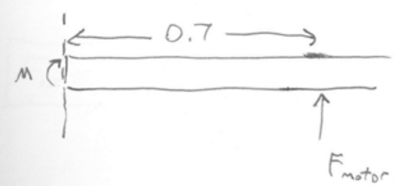
$$-\left(\frac{F_{motor}(1.26) - F_b(0.56)}{0.25}\right) - F_b + F_{motor} = 0$$

$F_b = 68.42$	bracket
$F_s = -47.4$	screw

⇒ Do fastener analysis

34.21 lb shear on each bolt
-23.7 lb of compression on each bolt

Beam analysis



$M = 0.7\,(21\ lb)$

$M = 14.7\ lb\text{-}in$

$\sigma = \frac{Mc}{I}$ $\quad I = \frac{bh^3}{12}$ $\quad b = 4.5\ in$, $h = 0.25\ in$

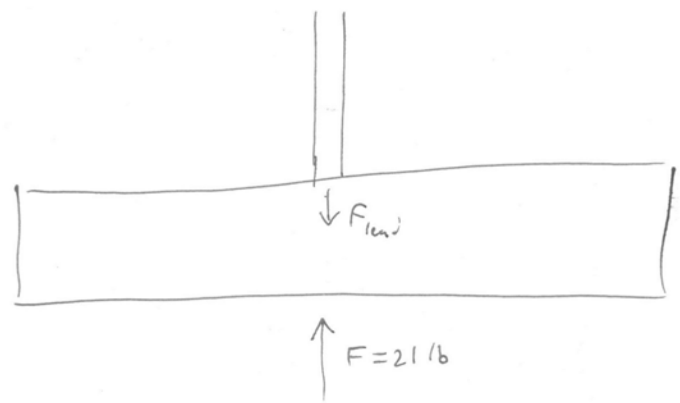
$I = 0.005859$

$\sigma = \frac{14.7\,(0.125)}{0.005859}$ $\quad \boxed{\sigma = 313.6\ psi}$

$\sigma < 1\,ksi$, $> 10^9$ cycles ✓

Lead screw buckling ~~Carriage Bearing~~

Force



$\boxed{F_{lead} = 21\ lb}$ $\quad K = 0.5$ (both ends fixed)

$F_{cr} = \frac{\pi^2 E I}{(K\ell)^2} = \frac{\pi^2 (26.1\times10^6\ psi)(9.707\times10^{-4}\ in^4)}{(0.5\cdot 12\ in)^2}$

$\ell = 12\ in$

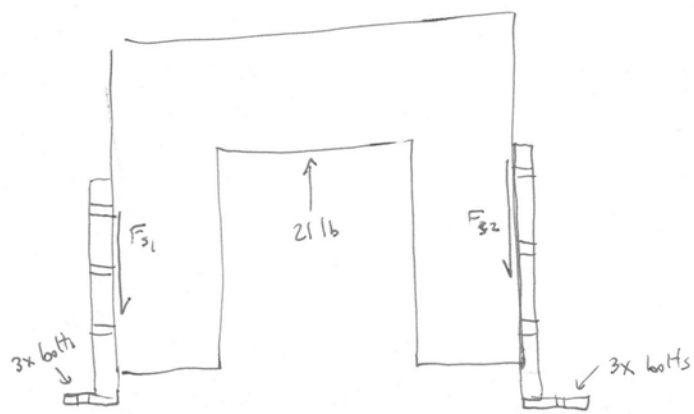
$I = \frac{\pi d^4}{64} = \frac{\pi (0.375)^4}{64} = 9.707\times10^{-4}\ in^4$

$\boxed{F_{cr} = 6945.8\ lb}$

$21\ lb << 6945.8\ lb$ ✓

No buckling

Stand Shear Analysis



$F_{s_1} = F_{s_2} = \frac{21\text{ lb}}{2} = 10.5$ lb per stand

= **3.5 lb shear on each bolt** ✓

= **3.5 lb of tension on each bottom bolt** ✓

Fatigue Analysis on Bolts

Assume: 1. lowest strength steel bolts available (SAE 1)
2. All load is carried by bolts

This is a very conservative estimate of the strength of the bolts. In reality the load is shared between the bolt and the material, and we are using stainless steel.

$S_p = 33$ kpsi

$A_t = 0.0175\ in^2$ (#10 bolt)

Recommended preload $= 0.90 F_p = 0.9 S_p A_p$

$F_i = 519.75$ lb

$K_f = 2.8$ (cut threads)

$F_{b_{min}} = F_i$

$F_{b_{max}} = F_i + CF_b$ Assume $C = 1$

Stand tensioned bolts $F_b = 3.5$ lb

$$\sigma_a = \frac{3.5}{2(A_t)} = 100 \text{ psi}$$

$$\sigma_m = \frac{519.75 + 3.5/2}{A_t} = 29.8 \text{ ksi}$$

$$\sigma_i = \frac{519.75}{A_t} = 29.7 \text{ ksi}$$

~~– transformation σ~~

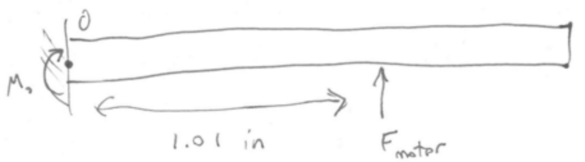
~~[illegible]~~ Shear strength of stainless steel = 0.5 S_y

= 35(0.5) = 17.5 Ksi

$$n_f = \frac{17.5}{1.95} = \boxed{8.97}$$ ✓

<u>All bolts are strong enough</u>

Beam bending Analysis on Motor plate



$$\Sigma M_O = -M_o + F_{motor}(1.01) = 0$$

$$M_o = (21 \text{ lb})(1.01) = 21.21 \text{ lb-in} = M_{max}$$

$$\sigma = \frac{Mc}{I} \qquad I =$$

Bolt analysis cont

$$n_f = \frac{S_e (S_{ut} - \sigma_i)}{\sigma_a (S_{ut} + S_e)}$$

~~$S_e = K_a K_b \dots S_e'$~~

~~$S_e' = 0.5 S_{ut} = 30$ ksi~~

~~$K_a = a S_{ut}^{b} = 2.7(60)^{-.265} = 0.912$~~

~~$K_b = 1$ (axial)~~

~~$K_c = 0.85$ (axial)~~

$S_e = 18.6$ kpsi for SAE 5

~~$S_e = (0.912)(0.85)(30) = 23.3$ ksi~~

$$n_f = \frac{(\cancel{23.3}\ 18.6)(60 - 29.7)}{0.1(60 + \cancel{23.3}\ 18.6)} = \boxed{71.7} \checkmark$$

static yield

$$n_p = \frac{33}{29.7 + 0.1} = \boxed{1.1} \checkmark$$

Failure in static loading first

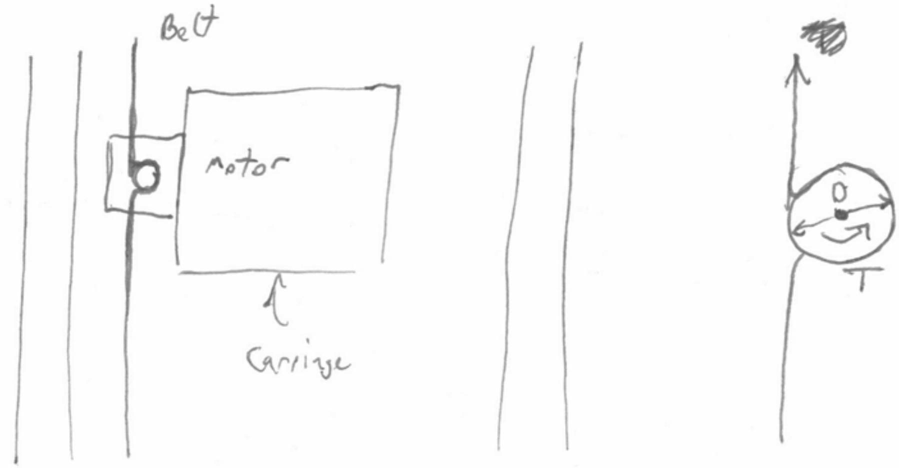
For shear Assume static failure

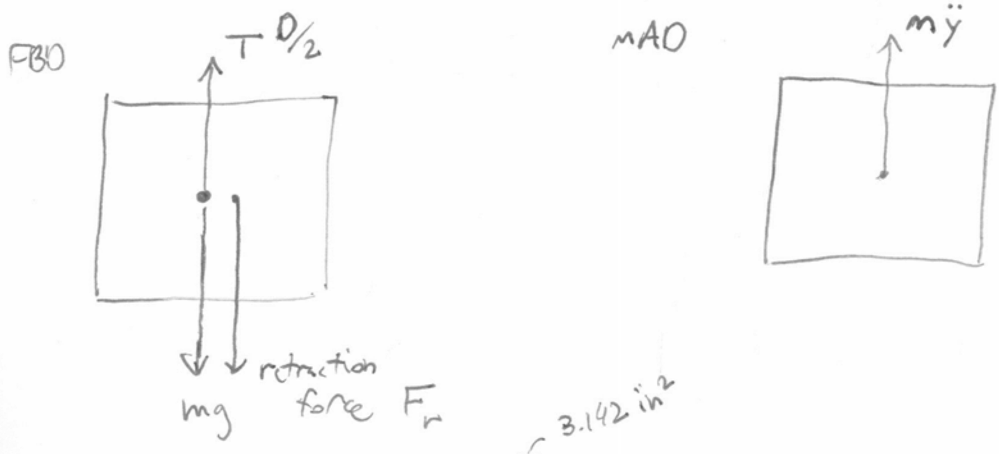
$$F = 34.21 \text{ lb/bolt}$$

$$\tau = \frac{F}{A_c} = \frac{34.21}{0.0175} = 1954.9 \text{ psi}$$

Motor selection

4/23/13





$3.142\ in^2$ (area of array)

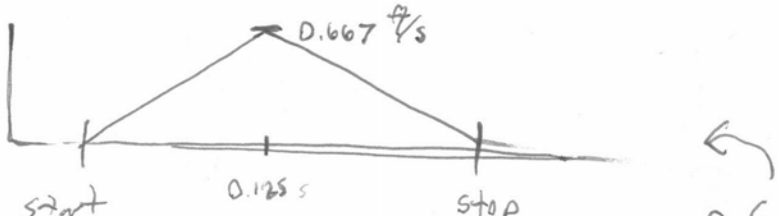
insertion force = (6 psi)(Area of array) = 19 lb

retraction force = (2 psi)(Area of array) = 6.3 lb

$$\Sigma F_y = T\,{}^{2}/_{D} - mg - F_r = m\ddot{y}$$

$$m = \frac{10\ lb}{32.2\ ft/s^2} = 0.311\ slugs$$

$$\ddot{y} = \frac{\Delta V}{\Delta t} \Rightarrow \bar{V} = \frac{1\ in}{0.25\ s} = 4\ in/s = 0.333\ ft/s$$



0.5 sec cycle time

$$\ddot{y} = \frac{0.667\ ft/s}{0.125\ s} = 5.336\ ft/s^2$$

Linear Actuator

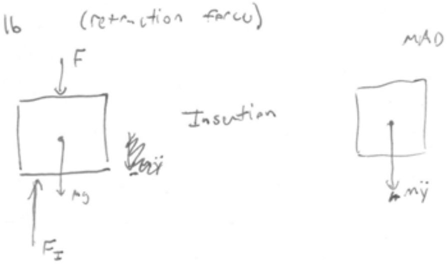
Research & calcs

4/30/13

$F = m\ddot{y} + F_r + mg$

$F = (0.311\ slug)(5.336\ ft/s^2) + (6.3\ lb) + (0.311\ slug)(32.2\ ft/s^2)$

$F = 17.97\ lb$ (retraction force)



$-F + F_I - mg = m\ddot{y}$

$F = F_I - mg - m\ddot{y}$

$F = (19\ lb) - (0.311)(32.2\ ft/s^2) - (0.311)(-5.336\ ft/s^2)$

$F = 10.64\ lb$

Not accounting for weight

$F = 20.66\ lb$

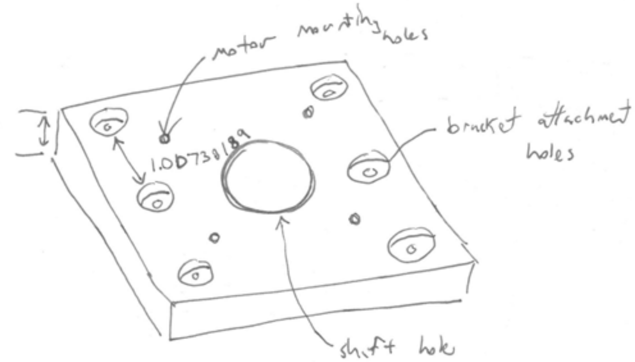
Actuator is rated for 9.5 in/s @ 22 lb load, 9 in/s no load

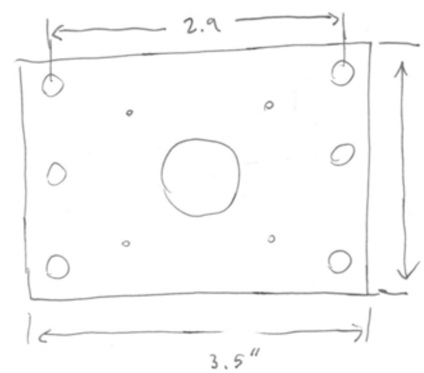
Use 2 actuators $119 each

See binder for details

Motor Bracket

5/1/13





STEPPING MOTOR ANALYSIS 05/13/13

	NEEDLE CARRIAGE	RETAINING RING
MOTORS:	3.0 A, 1.5 mH	~~3.0~~ 1.0 A, 1.5 mH
	200 STEPS/REV	200 STEPS/REV
ACTUATORS:	$2.50 \cdot 10^{-3}$ IN/STEP	~~$6.125 \cdot 10^{-5}$~~ $6.25 \cdot 10^{-4}$ IN/STEP

DESIGN FOR: 6 IPS OPERATION

1. POWER SUPPLY

FROM GECKO.COM (STEPPING MOTOR DRIVER MANUFACTURER)

$V_s = 32\sqrt{L}$, WHERE L = MOTOR INDUCTANCE [mH]

$V_s = 32\sqrt{1.5}$

$$\boxed{V_s = 40V}$$

MUST BE ABLE TO POWER BOTH MOTORS SIMULTANEOUSLY

$I = 3.0A + 1.0A$

$$\boxed{I = 4.0A}$$

2. STEP PULSE FREQUENCY

AT 6 IPS

$$f = \frac{6\left[\frac{IN}{S}\right]}{(\text{STEP SIZE})\left[{}^{IN}/_{STEP}\right]} \propto \left[\frac{STEP}{S}\right]$$

NEEDLE CARRIAGE

$$f = \frac{6\left[\frac{IN}{S}\right]}{2.5 \cdot 10^{-3}\left[{}^{IN}/_{STEP}\right]}$$

$$\boxed{f = 2.4\ KHz}$$

RETAINING RING

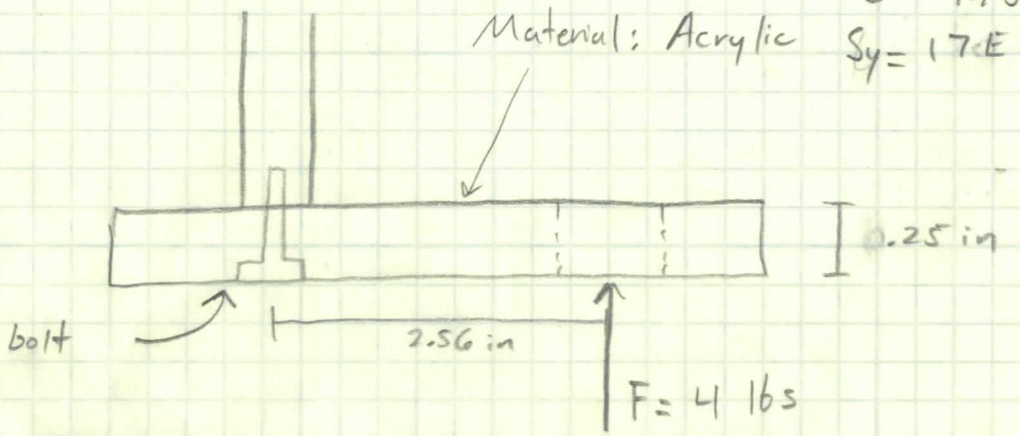
$$f = \frac{6\left[\frac{IN}{S}\right]}{6.25 \cdot 10^{-4}\left[{}^{IN}/_{STEP}\right]}$$

$$\boxed{f = 9.6\ KHz}$$

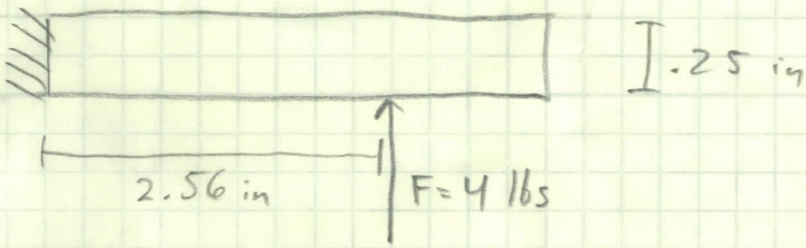
Rick Hayes

Retaining Ring Analysis

$E = 475E3$ psi
$S_y = 17E3$ psi



Beam Analysis



$$\sigma = \frac{Mc}{I} = \frac{(10.24)\left(\frac{.25}{2}\right)}{.0065}$$

$$I = \frac{1}{12} bh^3 = \frac{1}{12}(5)(.25)^3 = .0065$$

$\sigma = 197$ psi

$\sigma < S_y$

197 psi < 17,000 psi ✓

Well below the Endurance Limit ∴ no Fatigue calc necessary