1/9/08

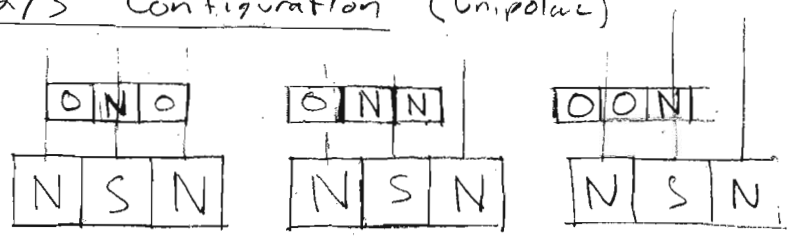
Linear Stepper Motors

How would you use a vernier type configuration to boost resolution?

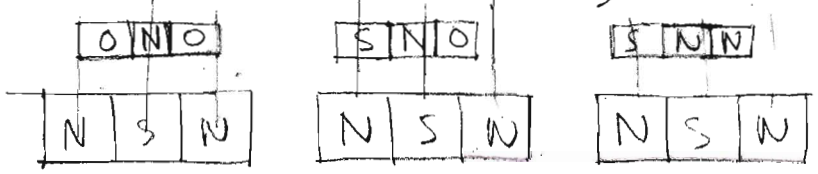
Unipolar is much simpler
- only one transister

Bipolar allows greater resolution
- four transisters per coil.

2/3 configuration (unipolar)

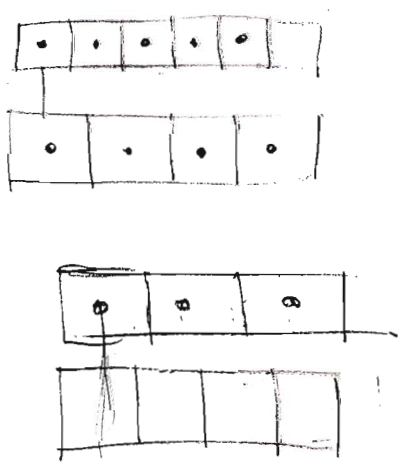


2/3 configuration (bipolar)



How to write this mathematically?

$\Sigma F = 0$ in static equilibrium.



only considering nearest neighbor.

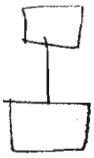
MATLAB PROGRAM

motor_state | n_1 | n_2 | n_3 | n_4 | n_5 | n_6 | n_7 | n_8 | n_9 | n_{10} | $n = -1, 0, 1$

distance_matrix

left magnet
right magnet
left magnet sign
right magnet sign

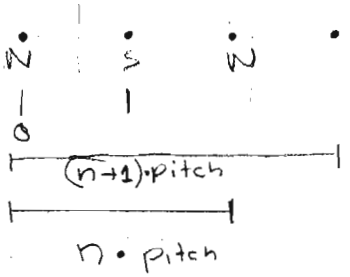
} 1 S, -1 N



$F = \frac{\lambda}{r^2}$ ⇐ simplification of attractive force between magnets

hypothesis: next stable configuration can always be found by toggling a coil.

$n = 2$



$F =$ motor_state $\cdot$ ~~sign(distance_matrix(i,1))~~ $\cdot$ distance_matrix(3,i)

motor_state: +1 N, 0 off, -1 S

~~sign(distance_matrix(i,1))~~: 1 positive, -1 negative

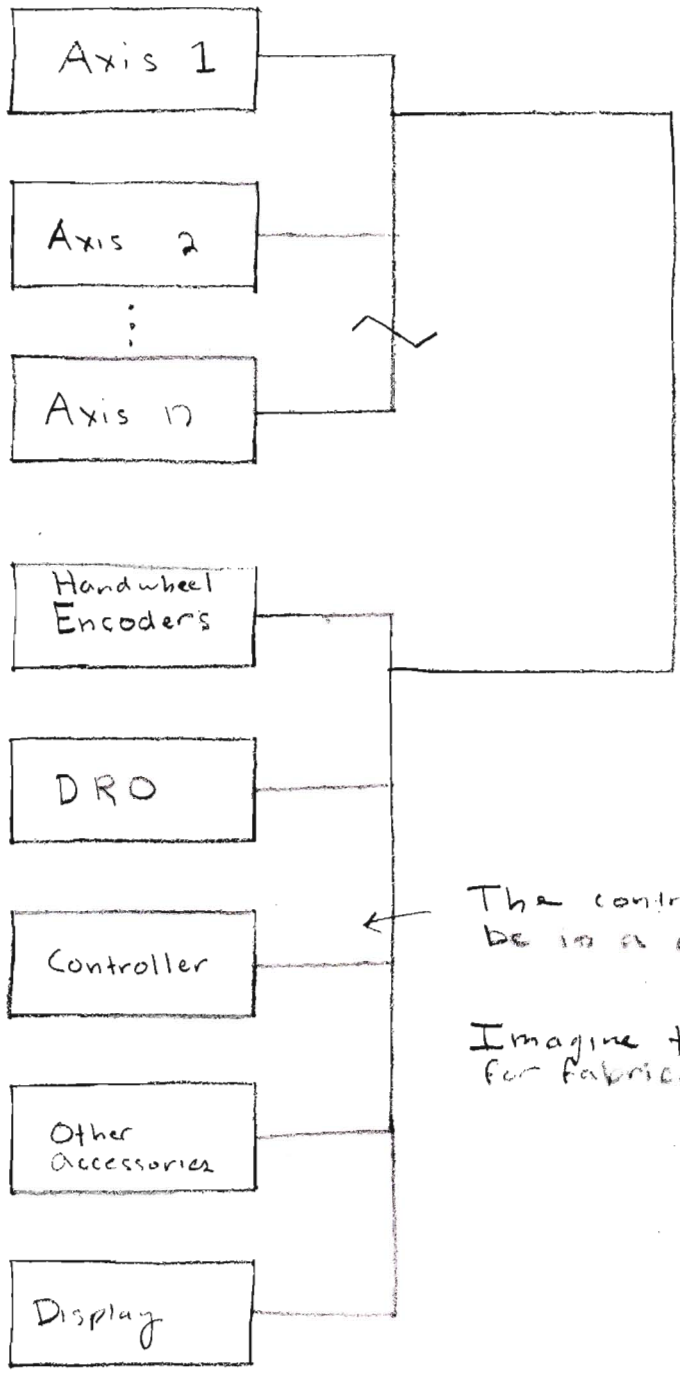
distance_matrix(3,i): -1 N, 1 S

$\cdot \frac{d}{\sqrt{d^2+g^2}} \cdot \frac{1}{d^2+g^2} = \frac{d}{(d^2+g^2)^{3/2}}$

1/16/08

Overall System Architecture:

The Goal: To create a networked machine architecture.



Hardware for now:

CNC Rotary Table	:	$395
X/Y axis	:	$450 + $130 for motors
2 motors		$130
2 handwheels	:	$ 30
		~$1000

450
235
400

Need encoders. ⇐ U.S. Digital

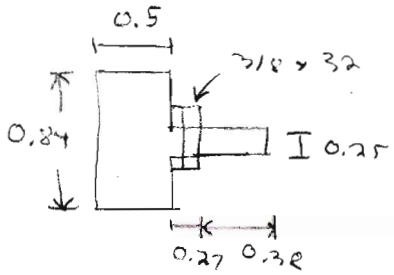
$140 per piece.

Encoders: US Digital PN: S4-200-NT

- 200 CPR, Light Static Drag

Cables: CA-3285-1FT

www.USDigital.com



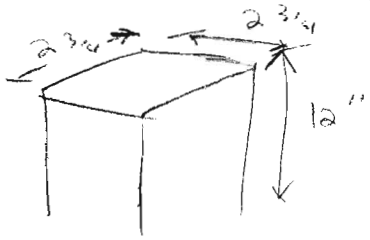
Handwheels:

Sherline PN 41050

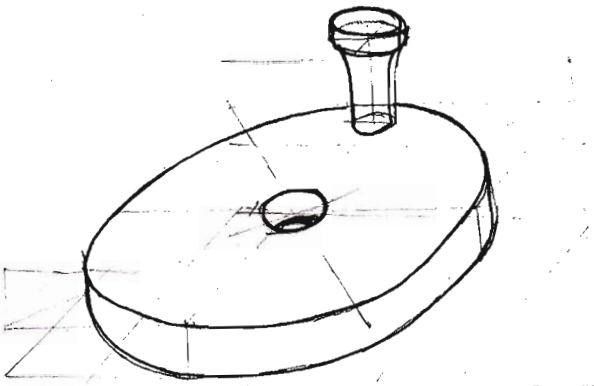
- 100 increment 1 5/8" handwheels.

Barstock

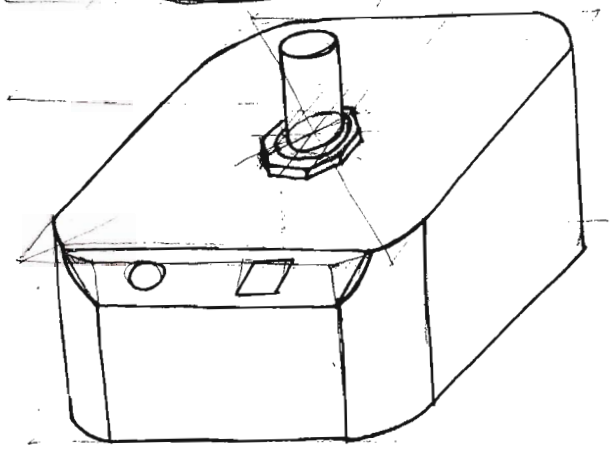
Mcm 9008K591



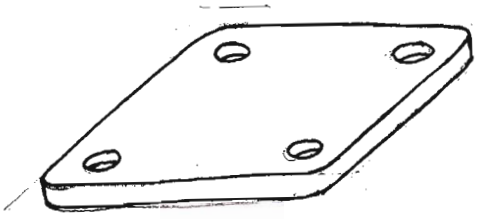
1/16/08



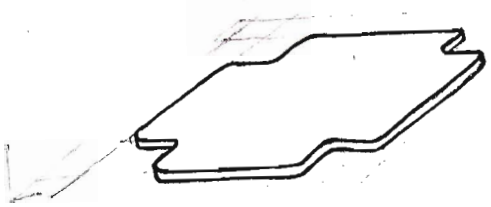
Handwheel
- Sherline



Casing and Optical Encoder
- US Digital

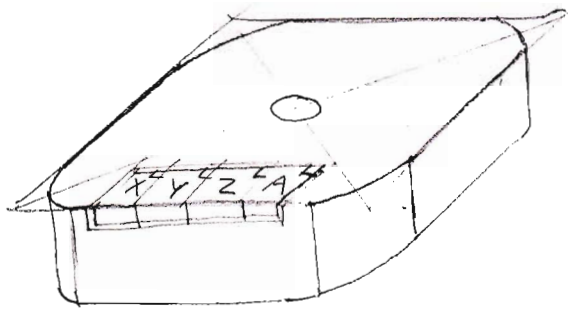


Bottom Panel



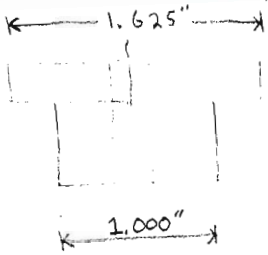
Rubber Foot Pad

Still need IØ connecter on back and internal IØ Board.

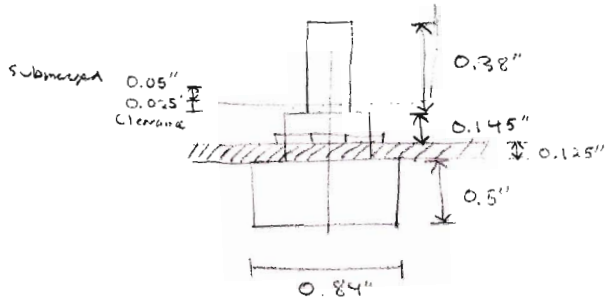

X Y Z A

1/17/08

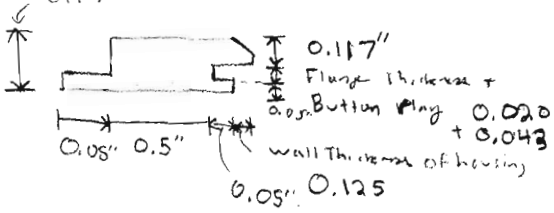
Sherline Handwheel

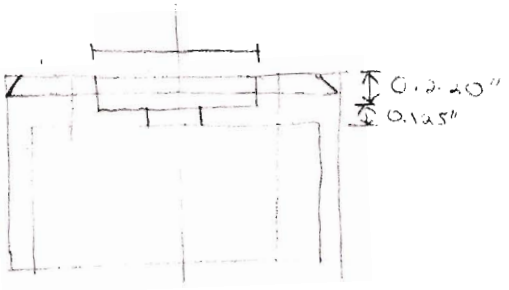


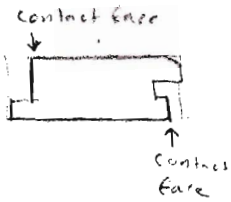
Encoder:



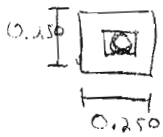
Keys

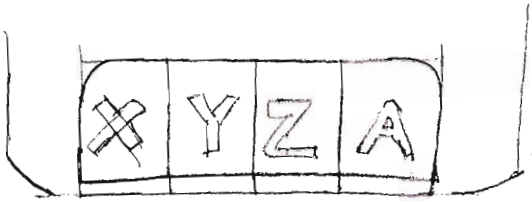


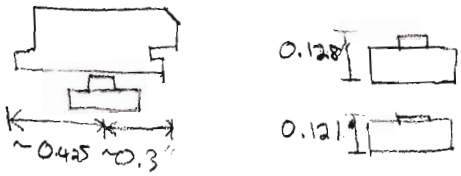




SMD Buttons

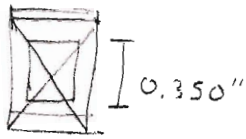




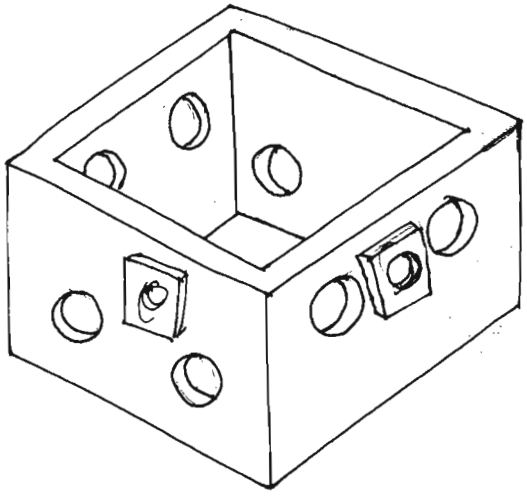


$$0.010'' \cdot \frac{0.725}{0.425} = 0.017$$

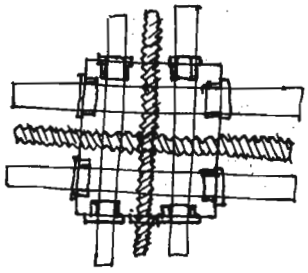
Stone

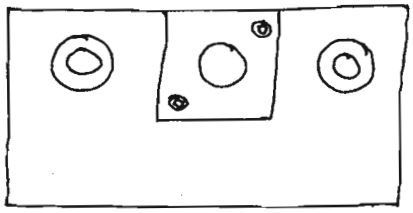


1/22/08



← Square Tube Extrusion

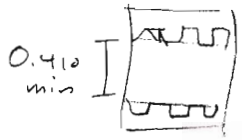




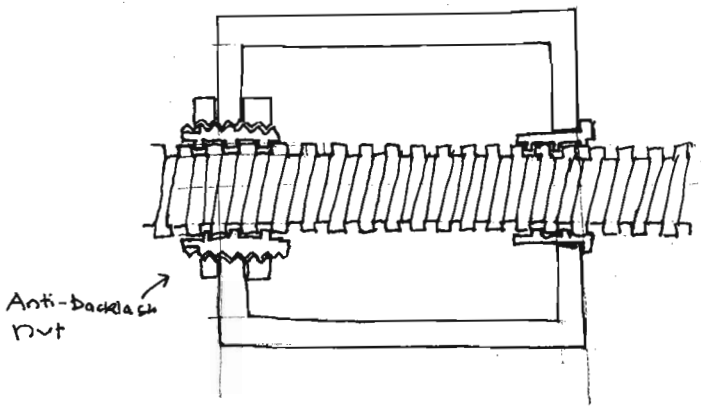
2C
(self-centering)
fit on acme
Precision rods

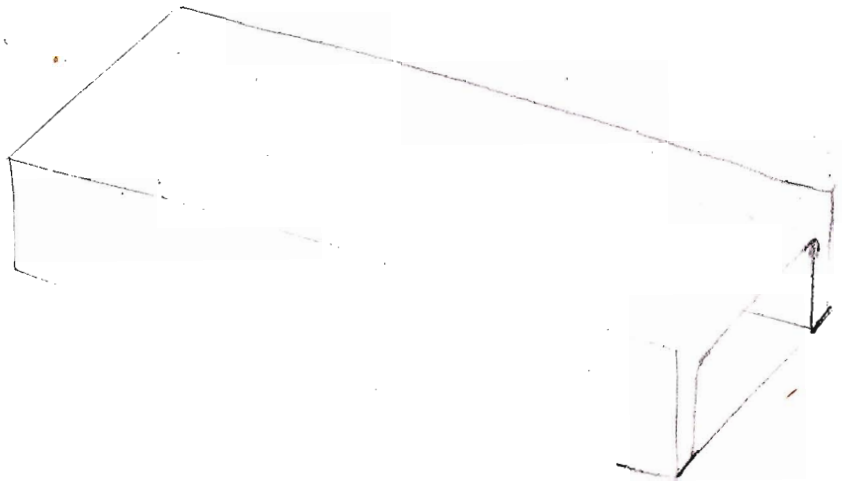
1/2-10 is 17.80/35"

P1804 ∈ M.H.



Drill Size: Letter Z
0.413

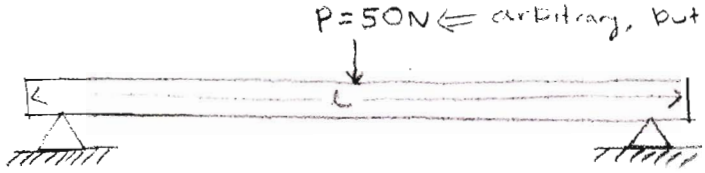

Anti-backlash
nut



1/23/08

Precision Shafts

OD	$/Ft Bar	$/Ft Pre Cut	$/Bushing
1/4"	4.44	4.47	
3/8"	5.10	5.31	
1/2"	5.41	6.23	
5/8"	6.99	7.20	



AISI 1566
UNS G15660

$E \simeq 190$ GPa ↑ efunda.com

$$\delta_{max} = \frac{PL^3}{48EI}$$ ⇐ engineersedge.com

Max allowable δ is 0.0005" per stage.

$$I_{min} = \frac{PL^3}{48E\delta_{max}} = \frac{(50)(0.305)^3}{(48)(190\times10^9)(1.27\times10^{-5})}$$

$I_{min} = 1.22\times10^{-8} = 2\left(\frac{\pi d^4}{64}\right)$ ⇐ Wikipedia "Second moment of area"

↑ two beams

$d = 0.0188\ m = 0.74"$

This is larger than I expected.

Assuming a shaft diameter of 0.5"

$$I = (2)\left(\frac{\pi}{64}\right)(0.0127)^4 = 2.55\times10^{-9}$$

$$K = \frac{P}{\delta} = \frac{48EI}{L^3} = 820.4\ kN/m$$

under a load of 50N; $\delta = 0.000061m - 0.0023$

Several factors to consider:

P < 10Lbs
Shafts are not simply supported.

$$\delta = \frac{PL^3}{192EI} \qquad I = \frac{2\pi d^4}{64} = \frac{\pi d^4}{32}$$

$$\frac{P}{\delta} = K = \frac{192EI}{L^3}$$

$$\frac{P}{\delta} = \frac{192\pi E d^4}{32L^3} = \frac{6\pi E d^4}{L^3}$$

$$d^4 = \frac{KL^3}{6\pi E} \Rightarrow d_{min} = \sqrt[4]{\frac{KL^3}{6\pi E}}$$

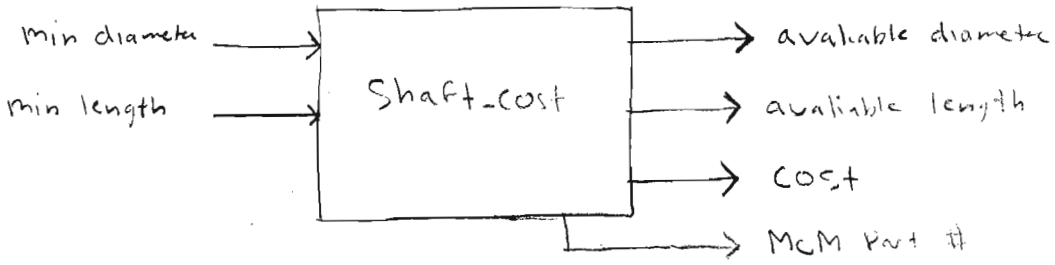
$$K = \frac{6\pi E d^4}{L^3}$$

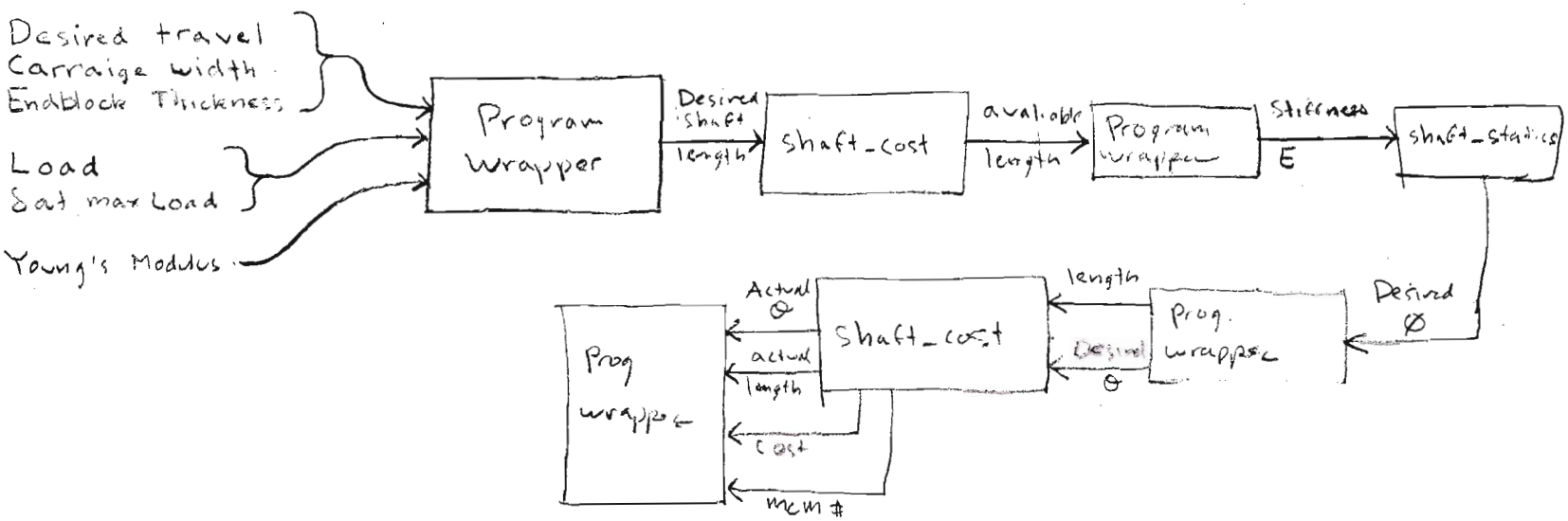
1/25/08

Project Accounting

Function [cost, diameter] = shaft_cost (min_diameter, length)

Shaft_data




Desired travel
Carraige width
Endblock Thickness
Load
Sat max Load
Young's Modulus
Program Wrapper
Desired shaft length
shaft_cost
available length
Program wrapper
Stiffness
E
shaft_statics
Desired Ø
Prog. wrapper
length
Desired θ
shaft_cost
Actual Ø
actual length
cost
mem #
Prog wrapper

1/26/08

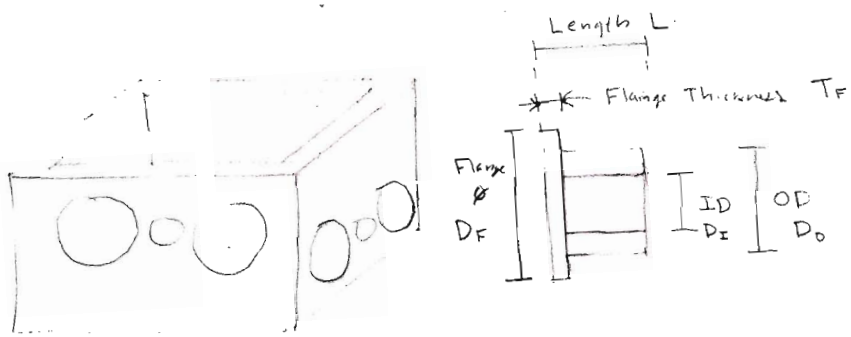
Static BOM

4" x 4" x 0.188" wall Aluminum Extruded Tubing - 12"	6546K261	$34.64
0.375 ID x 0.625 OD x 0.875 FOD x 0.5L Bushing	9440T19	$2.91
3/4"-16 Brass Threaded Rod -12"	98812A053	$40.45
3/4"-16 Expanding Locknut	94830A436	$2.76
5" wide x 0.15" Base Thickness 6061 U channel - 5'	1630T14	$69.37

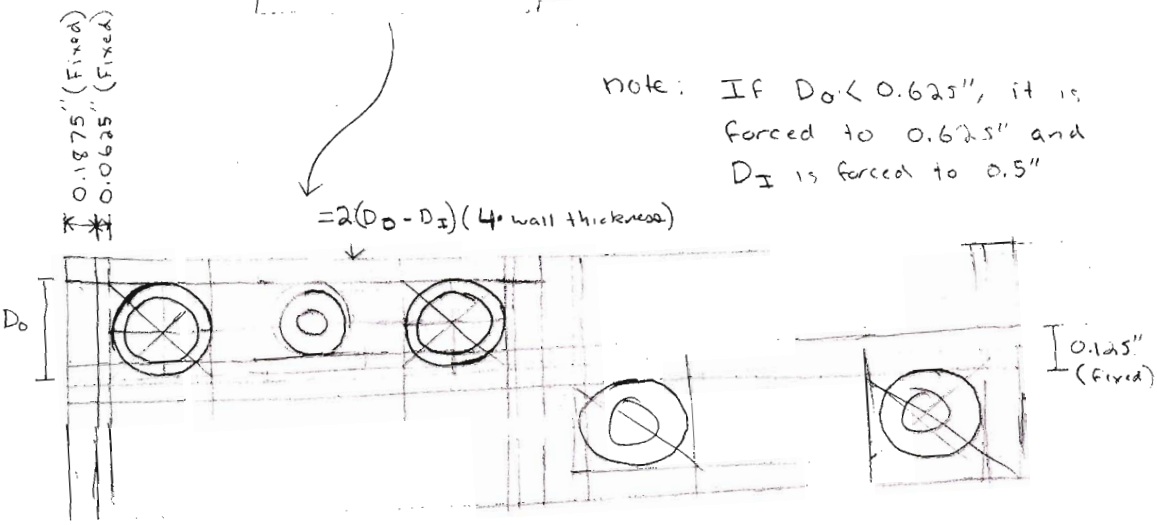
Tooling BOM

Acme Tandem Tap	1/2"-10	$69.45	25345A55

1/27/08



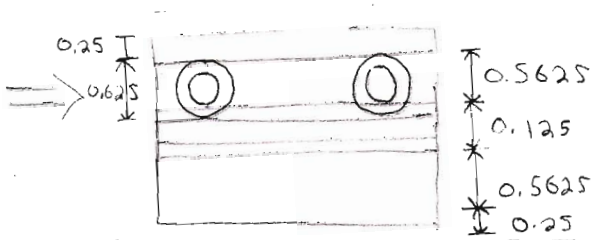
note: IF $D_O < 0.625"$, it is forced to $0.625"$ and D_I is forced to $0.5"$



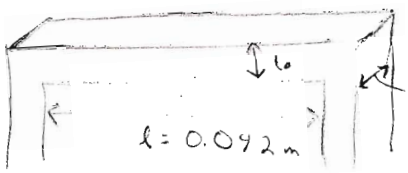
For Example

$D_I = 0.5$

$D_O = 0.625$



a quick bending problem: (assuming simply supported because this yields the highest stress/deflection)



$t_o = 0.00476\ m$

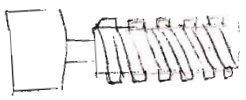
$I = \frac{bh^3}{12} = 2\times10^{-10}$

$E = 70\ GPa$

$\delta = \frac{PL^3}{48EI} = 0.00045\ m$

$= 0.455\ mm$

$= 0.017"$



$T = 136\ oz\cdot in$

$\approx 1\ N\cdot m$

$\tau \cdot \theta = F \cdot d$

$1 = F \cdot 0.00254$

$F = 393\ N$

$\sigma_{max} = \frac{Pl}{4Z}$

$Z = \frac{I}{\frac{t}{2}}$

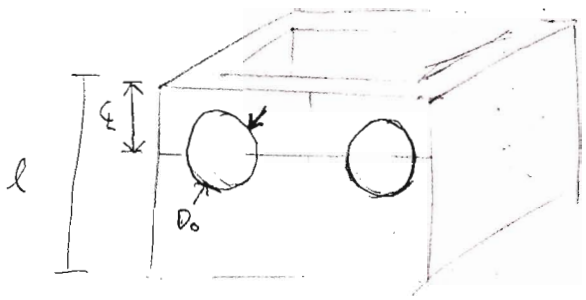
$Z = 8.403\times10^{-8}$ (thickness)

$= 1.075\times10^{-7}$

$\sigma_{max} =$

$= 10.75\ MPa \ll 275\ MPa$

Carriage.sldprt



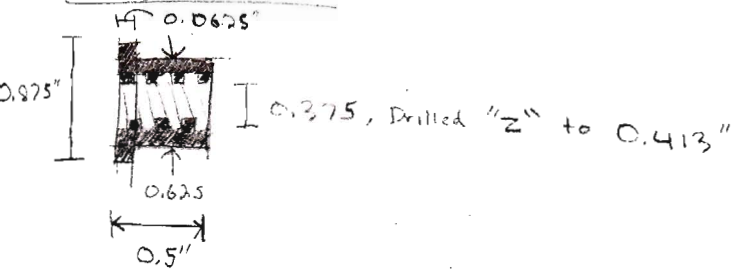
Locknut OD

$\ell = 2\ell + D_I + 0.125 \Rightarrow \ell = 2\ell + 1.25" + 0.125"$

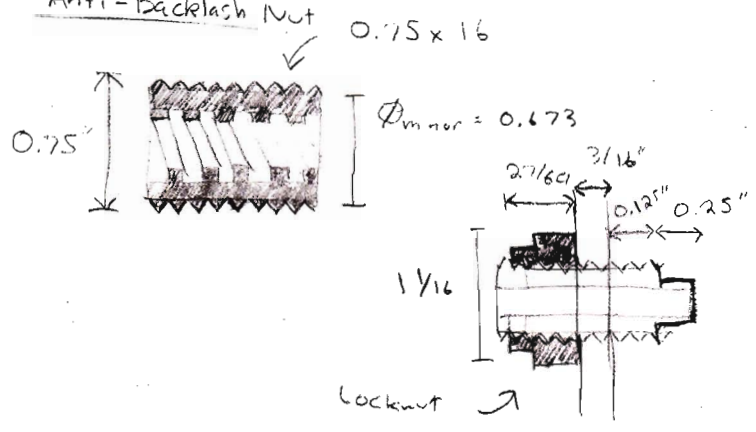
$\ell = 2 \cdot \left[\underbrace{2(D_o - D_I)}_{\text{wall} \to \text{bushing gap}} + D_I + \left(\frac{D_o - D_I}{2}\right) \right] + 0.125$

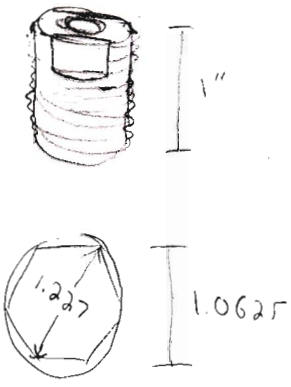
$\ell = \frac{D_o}{2} + 2(D_o - D_I)$ $\quad \ell > 0.625$ min. required for the anti-backlash nut.

ACME NUT

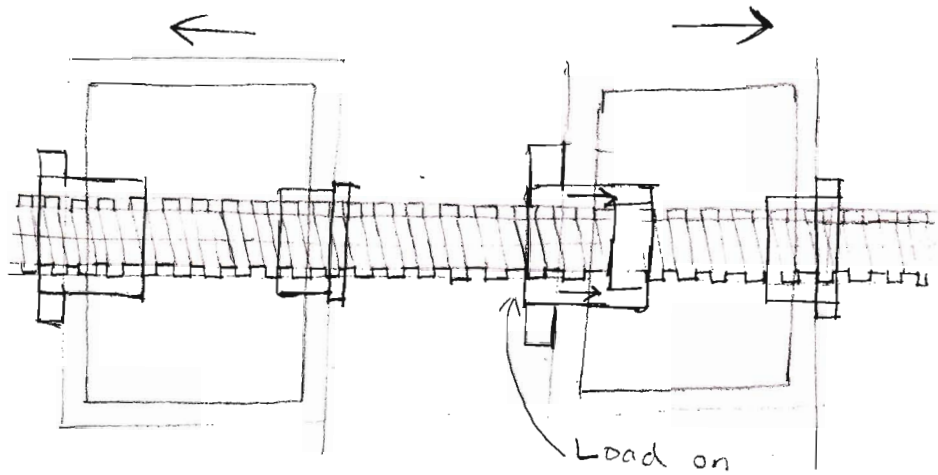


Anti-Backlash Nut





1/30/08



Load on backlash NUT.

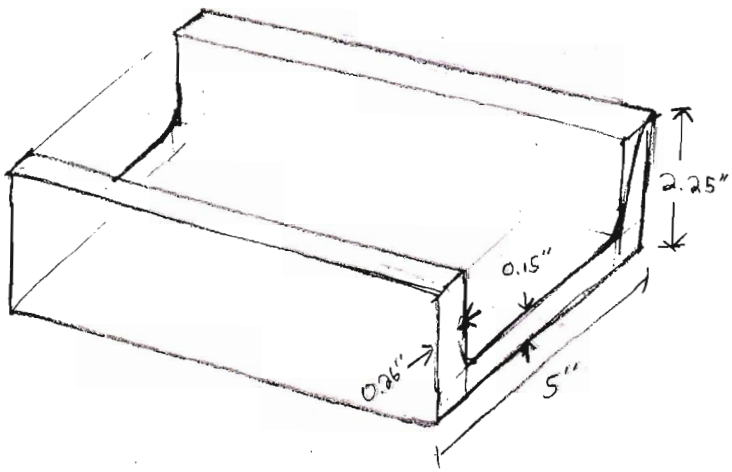
Locknut prevents rotation.

Load on static nut.

Flange bears force.

Press Fit prevents rotation.

BONUS?: Preloading the screw gives much greater stiffness to the box structure - Stiffness of screw + box?



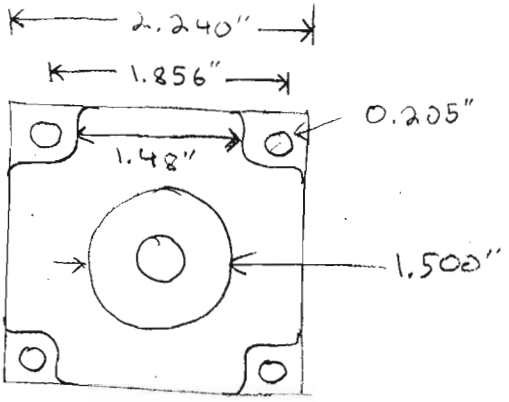
ASTM B308

≈2.21 Lbs/Ft

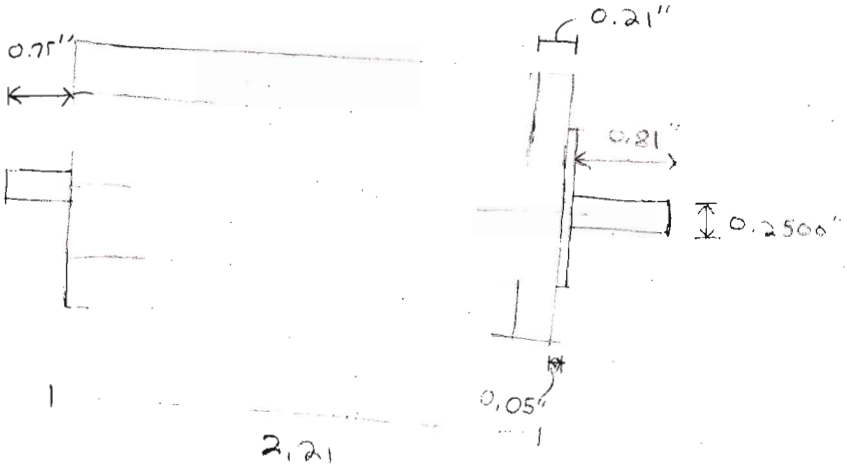
r = 0.38"

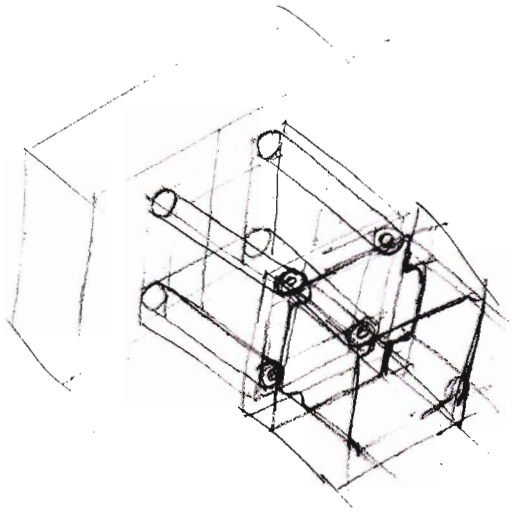
NEMA 23

Based on
Pacific Scientific
T-Series NEMA 23

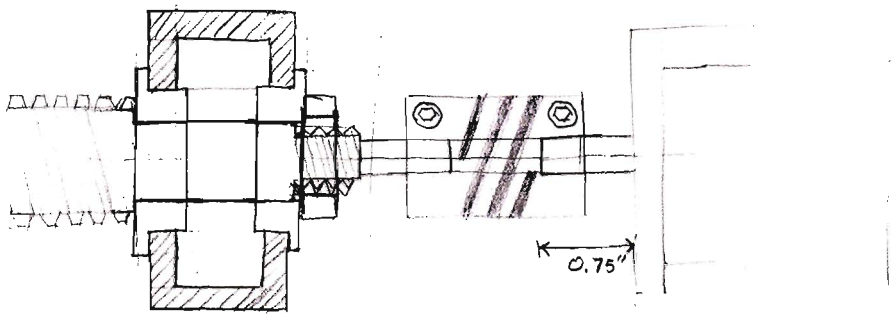


$1.856 \cdot \sqrt{2} = 2.625$





2/1/08

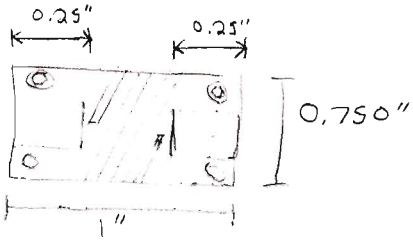


Coupler Selection:

136 oz-in = 8.5 Lbs-in

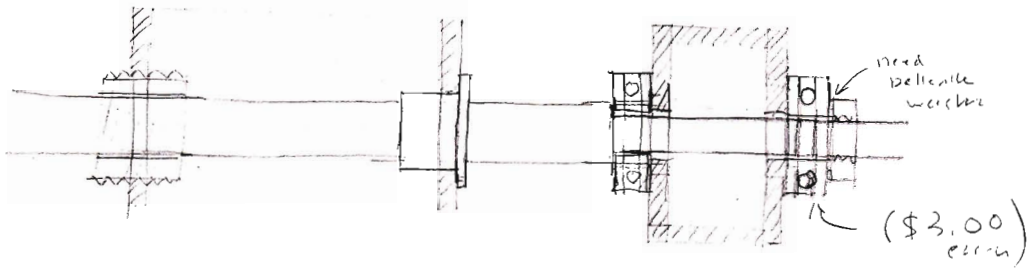
McM#

6208K13

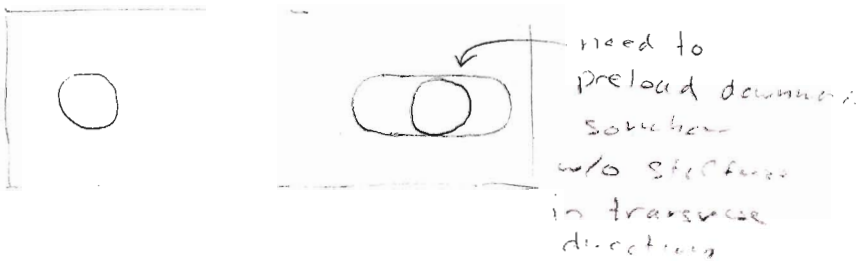


2/17/08

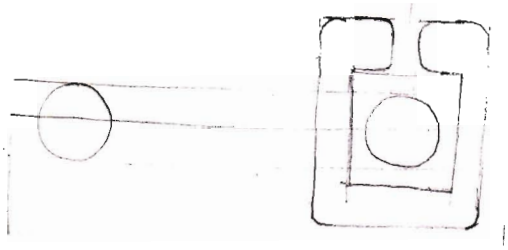
Thrust Bearing option

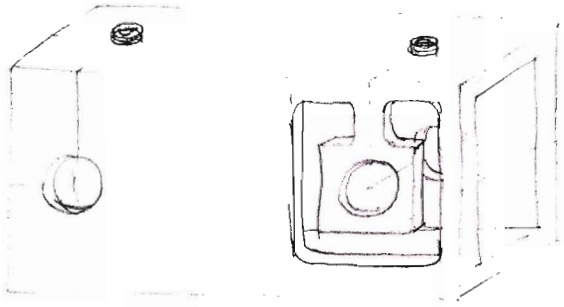


— Angular contact bearings are incredibly expensive from McMaster (~$40 each). Much cheaper from Misumi (~$5.00) but then not single-vendor.



— need to reposition machine tool in both cases (in Y direction) because slot can't have both top and bottom contact with the precision shaft.

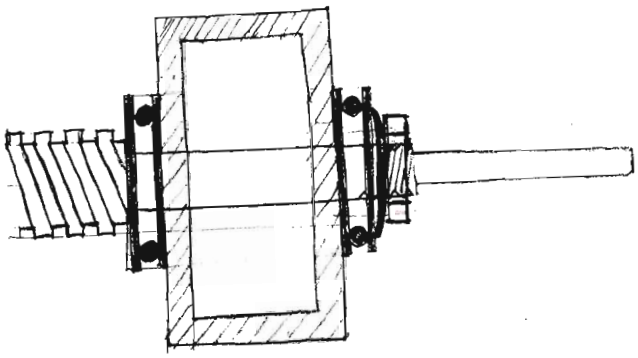




Downsides: This makes the shafts simply supported, which reduces stiffness by a factor of 4.

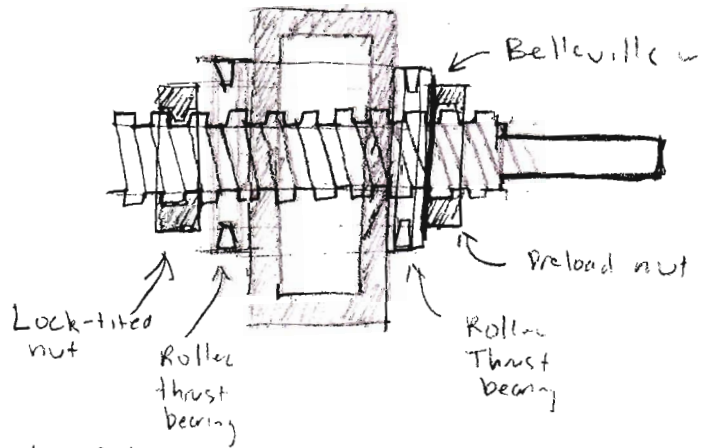
Could carriage have flexure instead?

2/20/08



Some discoveries today:

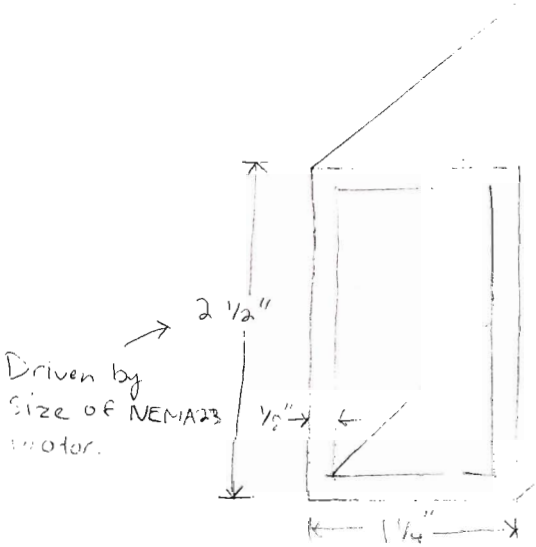
- Standard thrust bearings (cl cheapo) handle dynamic loads ~ 35 Lbs. This is not enough to handle both the preload and the maximum motor thrust force of ~ 80 Lbs.
- Needle-roller bearings handle dynamic thrust loads of 1920 Lbs ⇒ plenty! BUT, they are only available in a min. shaft Ø of 1/2".



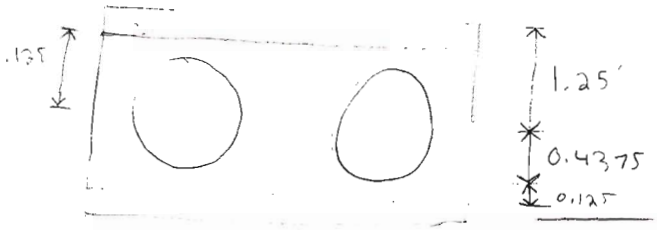
Cost of bearing set

2 x $2.53	= $5.03	Cage Assemblies
4 x $0.85	= $3.40	washers
2 x $2.22	= $4.44	nuts
1 x $0.44	= $0.44	Belleville washer

2/21/08



12"	88935K541	$9.89
36"	88935K593	$23.65
6'	88935K74	$41.50



$C_{max} = \frac{1.125}{2} + 2(0.25)$

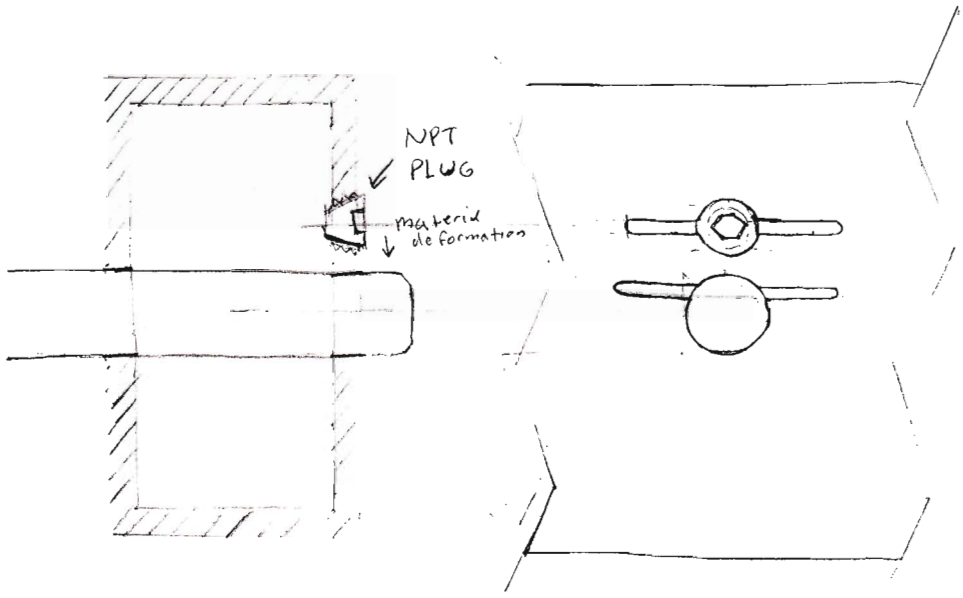
$= 1.0625$

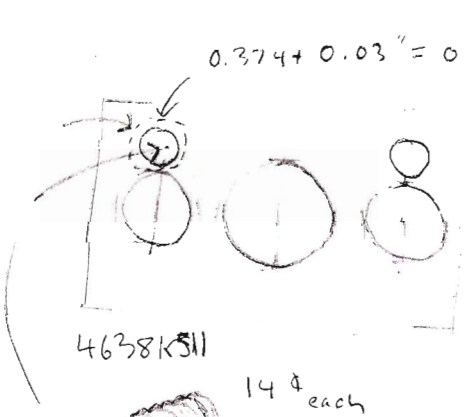
Clears

2" min for block w/o motor

2/23/08

Preloading the guide shaft seats



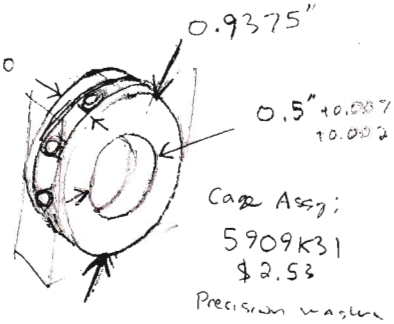


46381K511

14 ¢ each

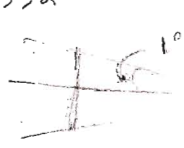
Hex Socket Plug

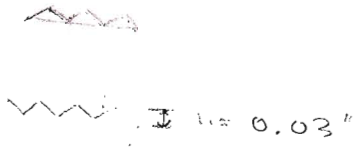
Letter Q drill
0.332"



Cage Assy;
5909K31
$2.53

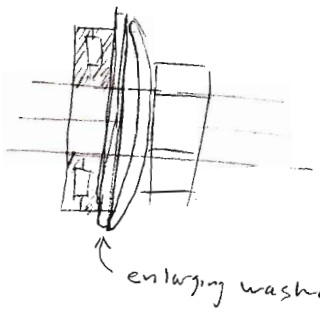
Precision washer
5909K44
$0.85



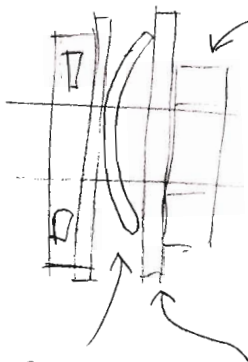


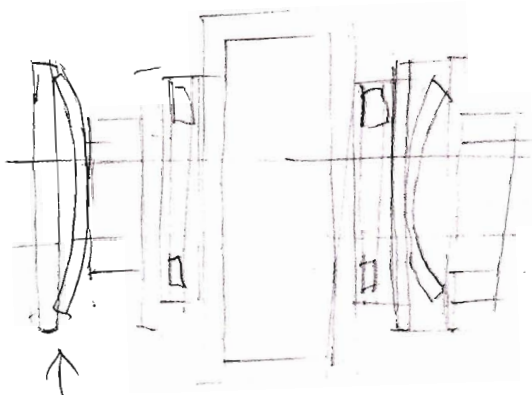
Belleville Disc Spring Selection / Preload Selection:

- Spec the smallest load available: 260 Lbs max



or

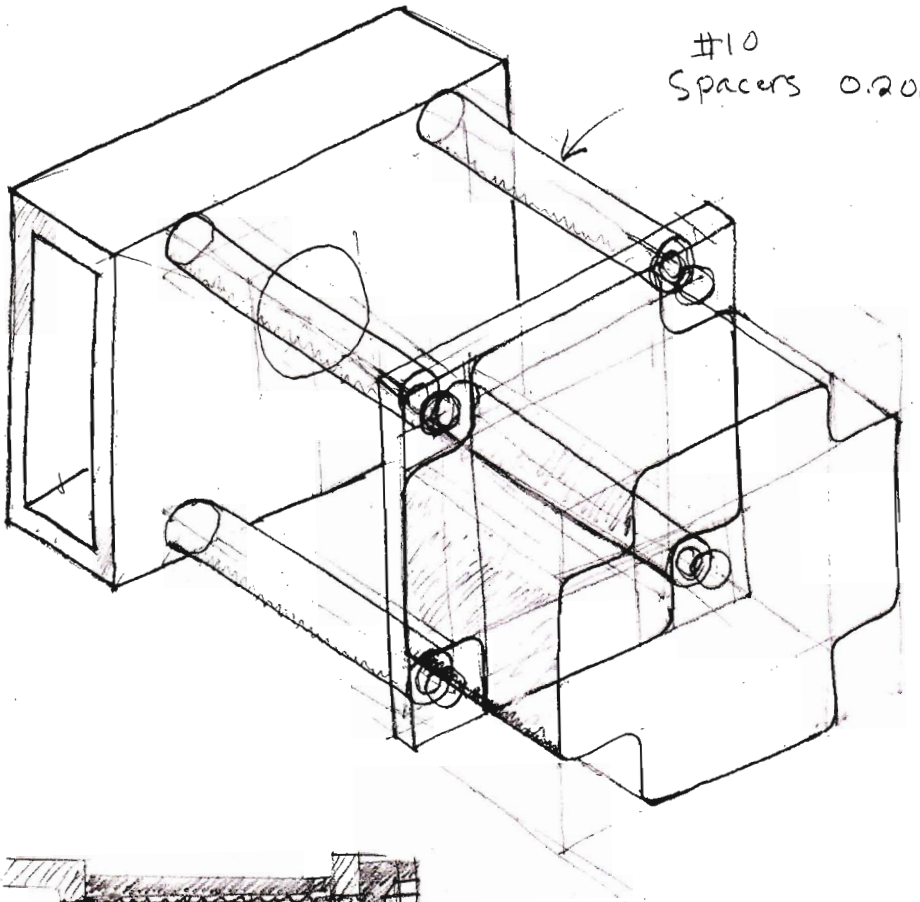


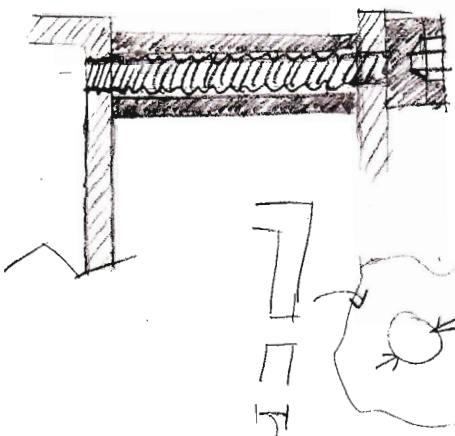


Safety feature to reduce impact if overrun occurs

3/1/08

Motor Mount





$\varnothing = 0.190$ (major $\varnothing$ of 10-32)
$0.004826m$

Surface area of thread root:

$0.004826m \cdot \pi \cdot 0.003175$
$= 4.813 \times 10^{-5} m$

D = 0.125"
0.003175m

$\sigma = \frac{F}{A}$

Safety factor of 4

$\sigma_Y \cdot \frac{1}{4} = \frac{F_{max}}{A}$

$\sigma_{Y\text{-aluminum}} = 270 \times 10^6$ Pa

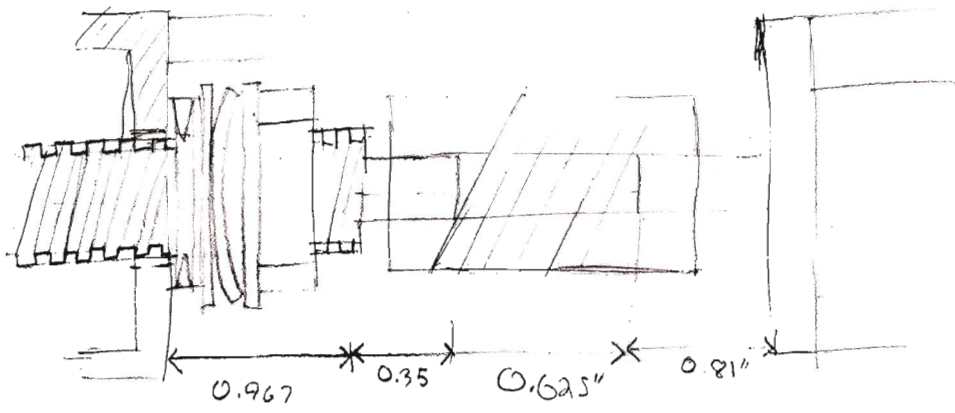
$F_{max} = 0.25\, \sigma_y \cdot 4.813 \times 10^{-5}$

$F_{max} = 3248$ N ∈ max bolt tension

bolt minor $\varnothing = 0.00383m$

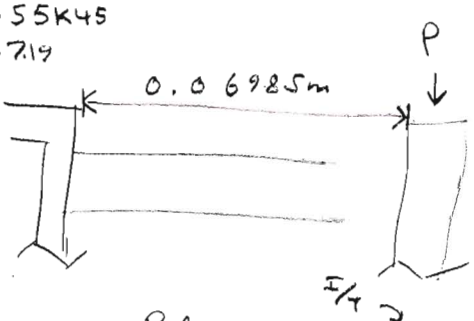
$\sigma_{bolt} = \frac{3248}{\pi \cdot \frac{\varnothing^2}{4}} = 281$ MPa $< \sigma_y$ for steel

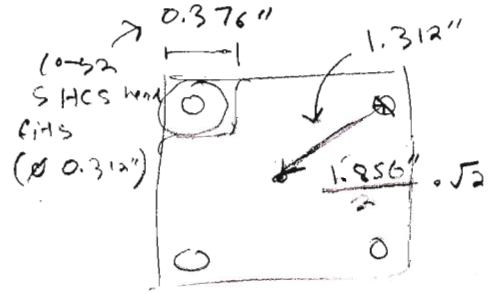
1200mPa



Total ^{min} Length of Spacer: 2.75" = 0.06985m

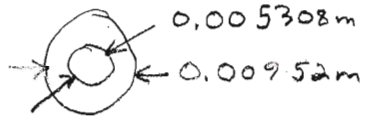
McM 89955K45
6' - $ 27.19
0.325" OD
0.209" ID





$\sigma_{max} = \frac{Pl}{2Z}$ where $Z = \frac{\pi(D^4 - d^4)}{32D}$ $r = 0.0333m$

$\tau = 1$ Nm distributed over 4 spacers.



$Z = 7.65 \times 10^{-8}$

Shear load per spacer → $0.25 Nm = P \cdot 0.0333m$

$P = 7.5$ N

$\sigma_{max} = \frac{(7.5)(0.06985)}{2 \times 7.65 \times 10^{-8}}$

$\sigma_{max} = 3.42$ MPa

$\sigma_{preload} = \frac{3248N}{\pi\left(\frac{D^2 - d^2}{4}\right)} = 66$ MPa

Making sure that the spacers can stay in contact with the motor and support block under max motor torque.

Preload is plenty... by a factor of 20.

Lets assume a safety factor of 3 on the preload stress...

Page 2 →

3/1/08

Motor Mount - Page 2

Bolt Tension

$$\frac{F_{bolt}}{\pi\left(\frac{D^2 - d^2}{4}\right)} = 10\ MPa \quad (3 \times \sigma_{max})$$

$$F_{bolt} = 490\ N$$

Assume that $\eta = 0.10$

10-32 bolt → lead = $\frac{1}{32}$" = 0.000793 m / turn

$$0.1 \cdot \tau \cdot \theta = 490 \cdot 0.000793$$

$$\tau = 0.618\ N \cdot m = \boxed{0.45\ ft\text{-}lbs}$$ bolt tensioning torque

Deflection

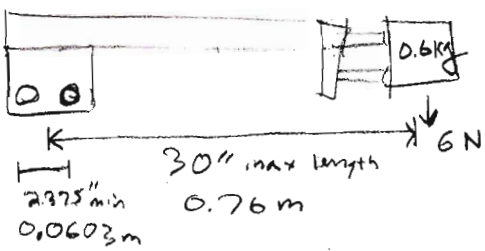
$$\delta_{max} = \frac{Pl^3}{12EI} = \frac{7.5 \cdot 0.06985^3}{(12)(200\times10^9)\left(\pi\right)\left(\frac{D^4 - d^4}{64}\right)} = 2.92 \times 10^{-6} m$$

Convert to radians → $\frac{2.92 \times 10^{-6}}{0.0333} = 8.768 \times 10^{-5}$ rad

In rotations of leadscrew: 0.1"/rev = 0.00254m

$$8.768\times10^{-5} \cdot \frac{0.00254}{2\pi} = \text{insignificant.}$$

Torque exerted by cantilevered motor on lower stage:



$T = 4.56\ N \cdot m$

$F_R \simeq \left(\frac{0.76}{0.06}\right) \cdot 6N = 76N$

Stage stiffness of 4×10^6 N/m

$1.9\times10^{-5} m = 0.000019m$

$= 0.019mm$

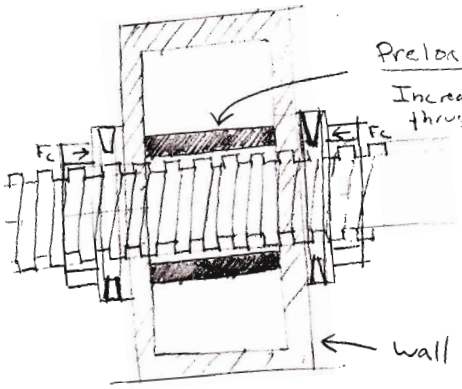
This is worst case and tolerable ⇒ $= 0.00074in$

Fasteners:

10-32 shcs 3" min

McM # 91251A360
$6.91 pk 25

3/3/08



Preload spacer

Increase the rigidity of the tube under thrust loads.

0.5" ID 0.0127m
0.625" OD 0.0159m
SAE 841 Bronze

← Wall thickness tolerance:
+/- 0.010"

↑
Spec. bushings with a length of 1 1/8"
- face down to proper size.

Area: $\frac{\pi(0.0159^2 - 0.0127^2)}{4}$

$= 7.19 \times 10^{-5} m^2$

$\sigma_y = 75$ MPa ∈ SAE 841
$\sigma_y = 125$ MPa ∈ SAE 660 $E = 100$ GPa

SAE 841

5/8" OD	6391K217	$1.29
3/4" OD	6391K226	$2.56

SAE 841 Bronze ∈ oil impregnated for now → SAE 660

5/8" OD	6381K112	$2.15
3/4" OD	6321K523	$3.08

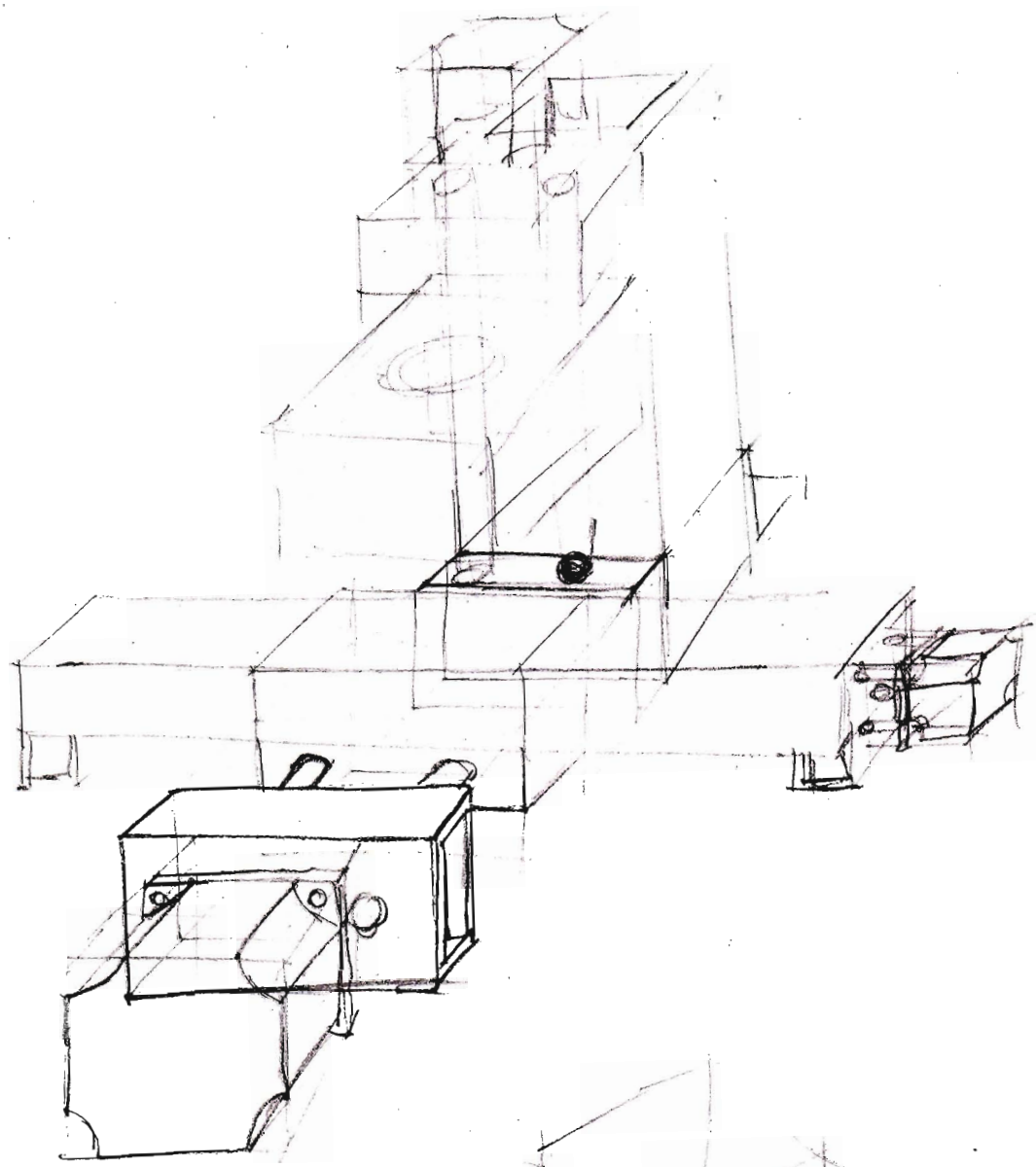
$\sigma_y = 75 \times 10^6$

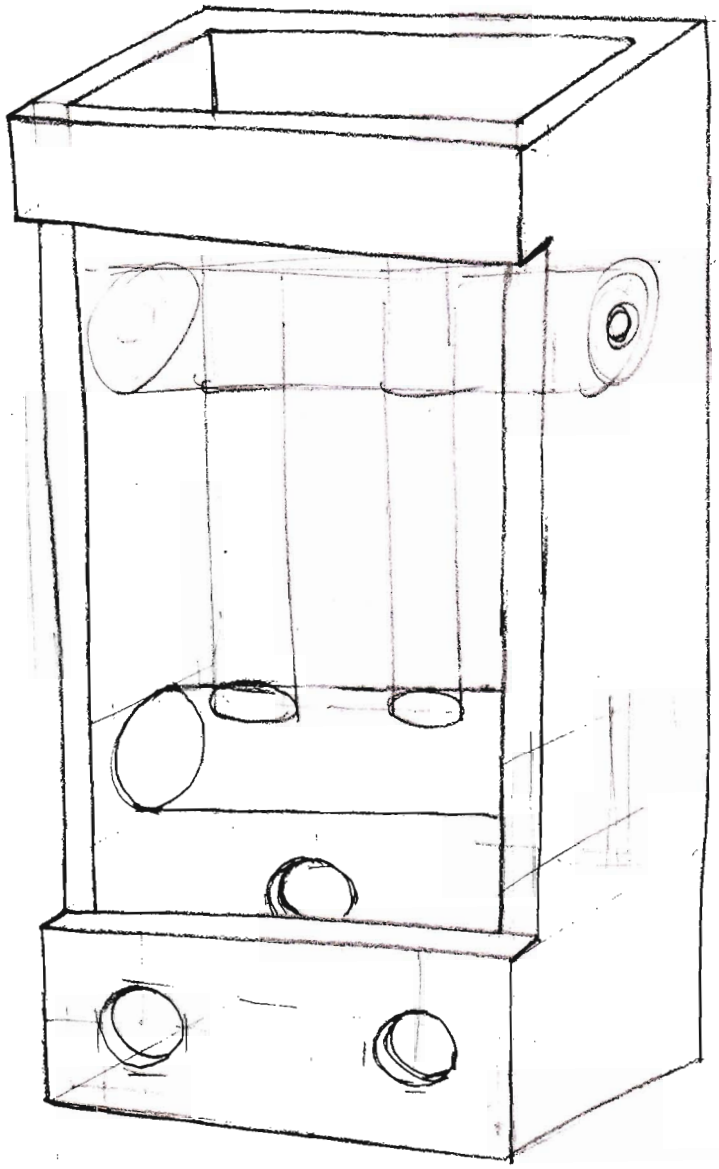
$\sigma_y = \frac{F}{A}$

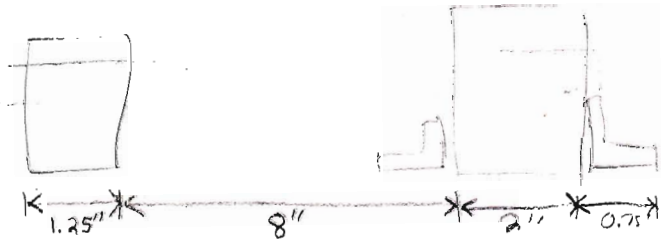
$F_{max} = \sigma_y \cdot A = 5392$ N ⇒ 539 kg ≈ 1000 lbs

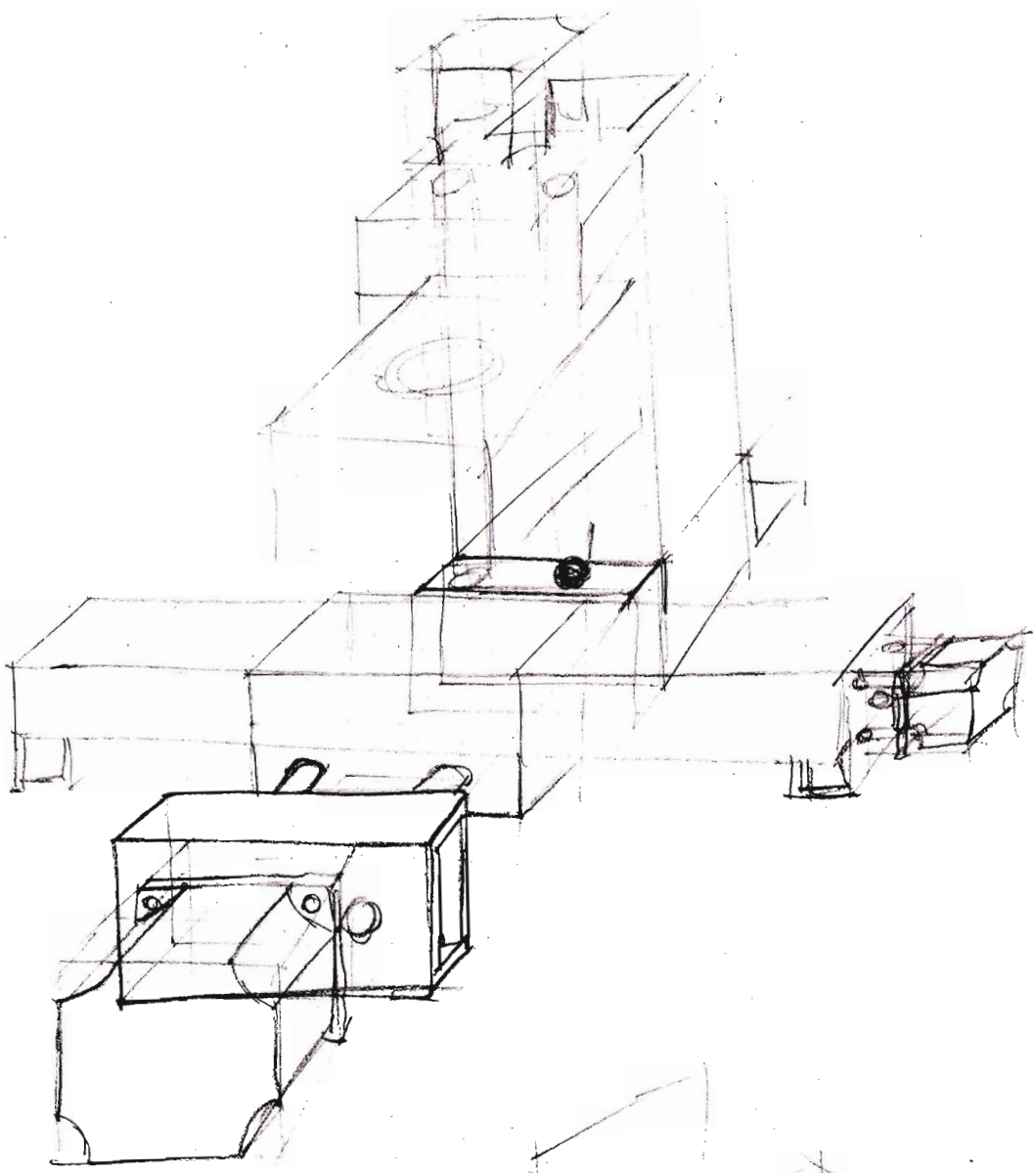
Problem: leadscrew clearance holes in support blocks are 0.625 Ø, which means that the bushings have no support.

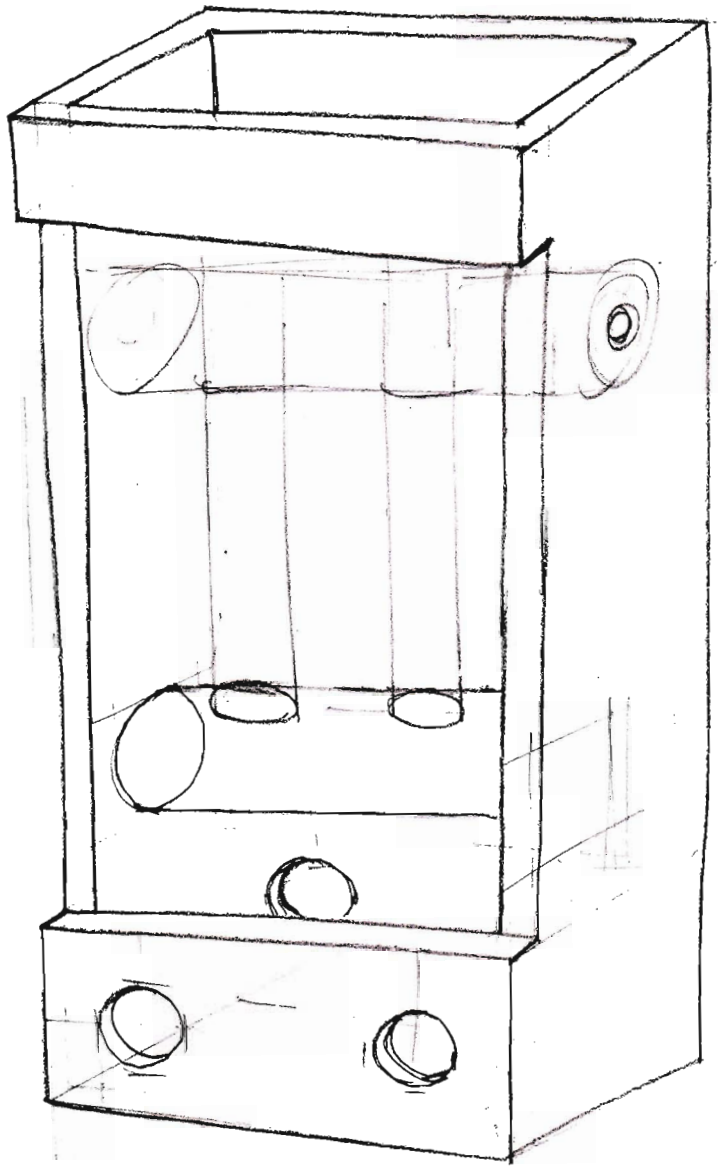
0.750 OD
0.625 ID
SAE 660
1.125" long
} 6381K534 $2.34

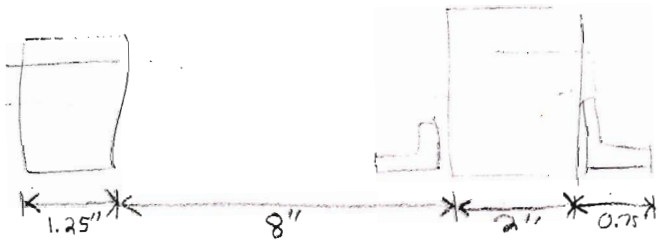





1.25"
8"
2"
0.75"



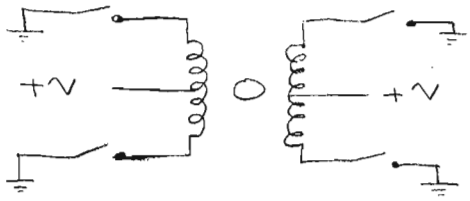



1.25"
8"
2"
0.75

118
456

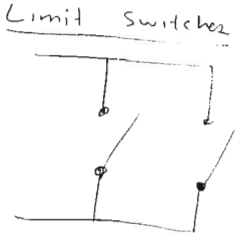
3/8/08

Stepper Motor Driver



ATMega 88

I/C data line to pin 12 ⇒ ICP1 (input capture pin)



either seperate or in tandem

Mosfets in lab are enhancement mode

PC0...PC3 ⇐ Stepper out pins

0b
↑
binary

.DEVICE ATmega 88

ST/STS/STD

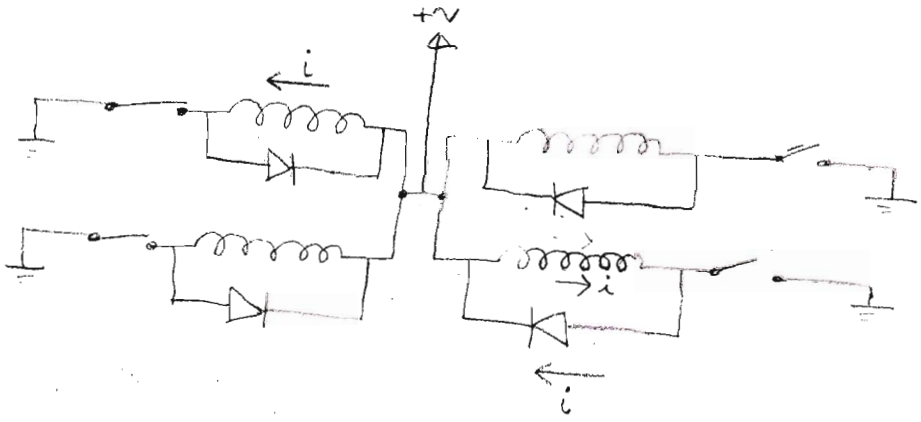
LD/LDS/LDD

STS k, Rd ⇐ Store directly to data space

Use instead of OUT for certain ~~register~~ registers

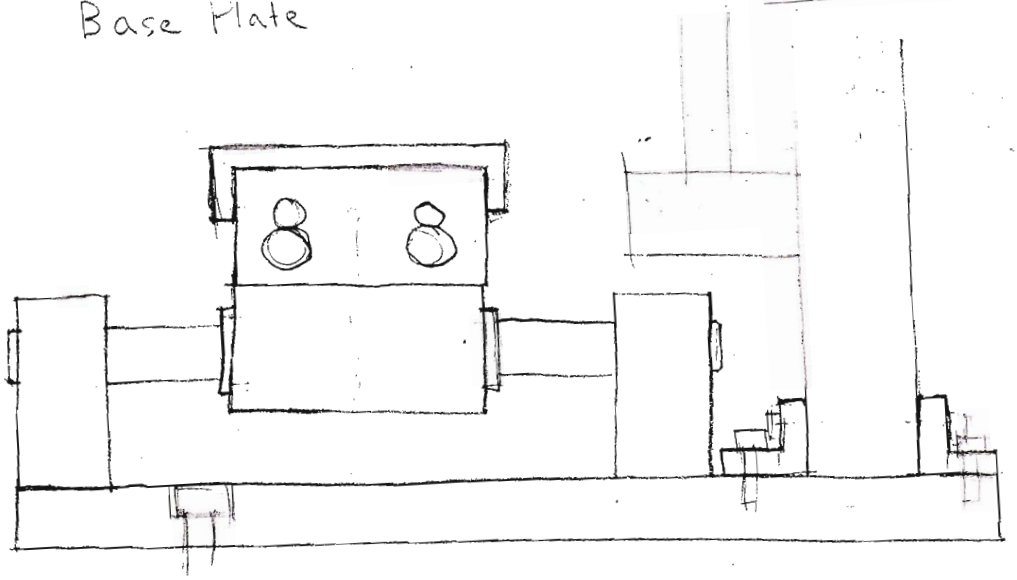
3/17/08 IØSMC Rev2

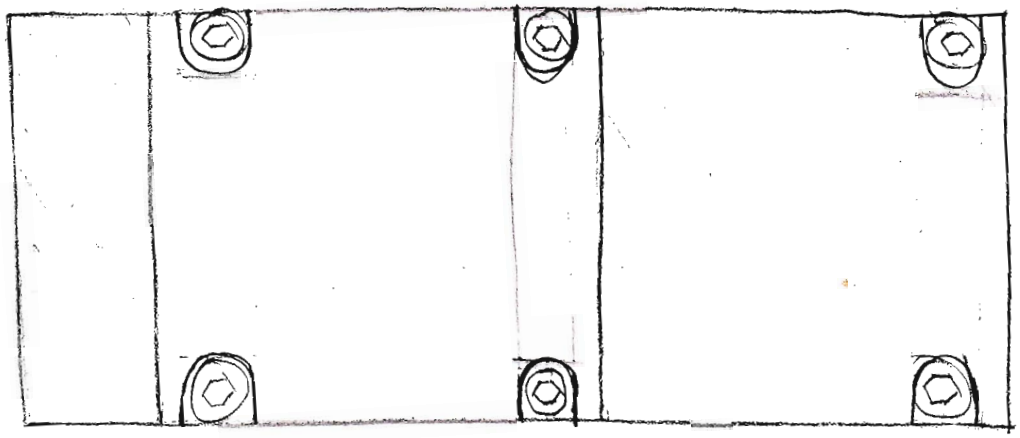
According to "Practical Electronics for Inventors" should add a protection (flyback) diode across each coil:



3/16/08

Base Plate





Does mounting bolt location affect accuracy of Plate?

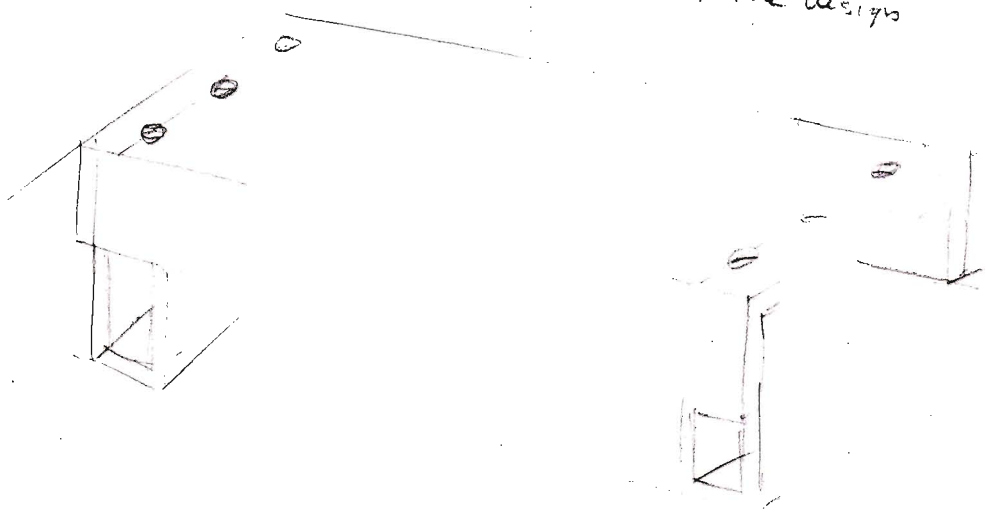
Might even want rubber or spring washers to prevent machine from being excessively clamped to the table

1/4" SHCS or LHCS as standard fasteners

Mounting the Table

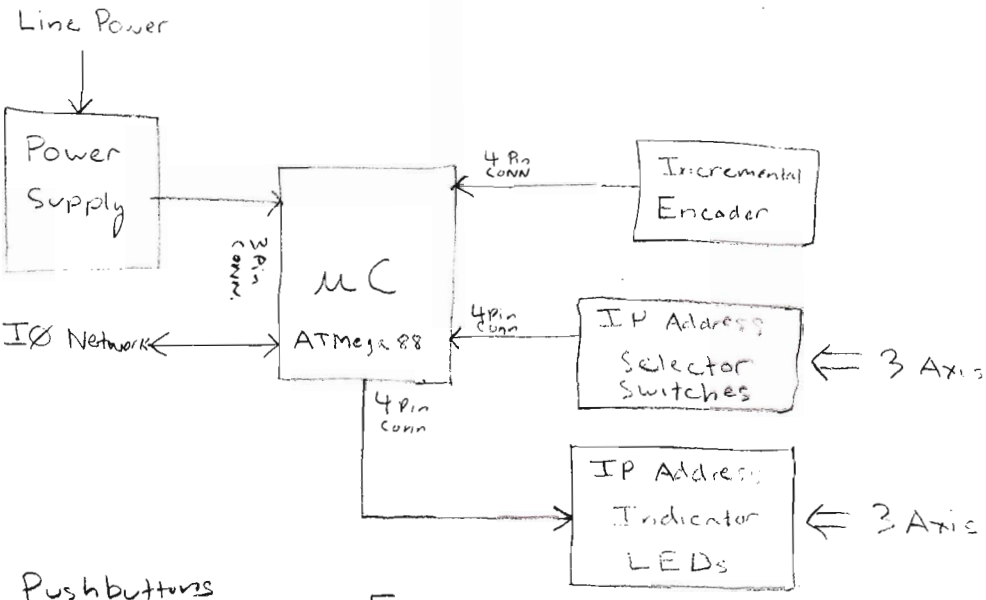
can't use FHCS

→ Self-centering screws will over constrain the design

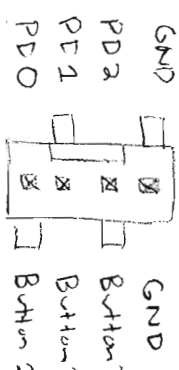


3/17/08

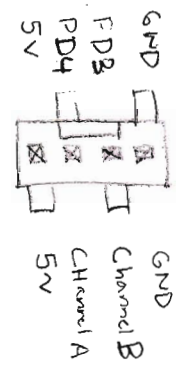
IØ Jog Dial



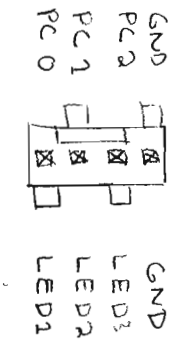
Pushbuttons



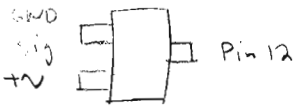
Encoder



LEDs



Power



2/17/08

Design Table Parameters

Property	Dimension Name
X Shaft Ø	D1@ Shaftprofile
X Shaft L	D1@ Shaftbody
Y Shaft Ø	D1@ Shaftprofile
Y Shaft L	D1@ Shaftbody
Y Bushing ID	D2@ flange_profile
X Bushing OD	D1@ body-profile
X Bushing L	D1@ body
X Bushing Flange Ø	D1@ flange-profile
X Bushing Flange Thickness	D1@ flange
Y Bushing	Same as X Bushing
Carriage Length	D1@ CarriageBody
X Axis Centerline	D3@ X-axisBushingLocations
X Axis Shaft Seperation	D1@ " "
X Axis Bushing Seat Ø	D2@ " "
Y Axis Centerline	D3@ Y-axisBushingLocations
Y Axis Shaft Seperation	D2@ " "
Y Axis Bushing Seat Ø	D1@ " "
Carriage Width	D1@ CarriageExtrusion
Carriage Wall Thickness	D2@ CarriageExtrusion
X Axis Motor Block seat sep.	D2@ bushinglocations
X Axis Motor Block seat Ø	D1@ bushinglocations
Table Length	D1@ Body
X Axis Leadscrew Overall Length	D1@ Body, D3@ thread
X Axis Leadscrew Reduced Ø length	D1@ bearingreduction
Y Axis Leadscrew " "	" "

2/17/08

Excel Commands

Match

MATCH (value, Range, mode)

Example of Range is A4:A10

mode:

- -1 Closest match that is larger than value
  - Range must be in descending order
- 0 Exact match
- 1 Closest match that is smaller than value
  - Range must be in ascending order

INDEX (array, row_num, column_num)

Concatenate

CONCATENATE (text1, text2, ...) also &

ROUND (number, num_digits)

2/17/08

Steps and conventions for merging parts with Design Table

In the solidworks part:

1. name all linked features and sketches appropriately
2. name all linked dimensions appropriately ← Can't name features, so won't do this for the sake of consistency
3. make sure that configuration is called "Default"

In the Excel workbook "Machine Design.xls":

1. Create a new worksheet entitled the part name
2. Fill in only dimensions which are present in the model

- Need all excel files to be open in order for auto-update to function.

3/3/08

Bug List

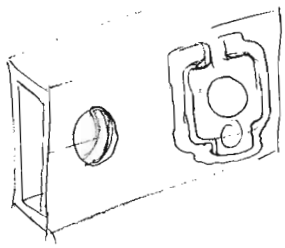
1. Desired stiffness is greater than actual stiffness.
   - Partly due to discrete shaft lengths available.
   - Methods of computing stiffness are different
     - different working length.

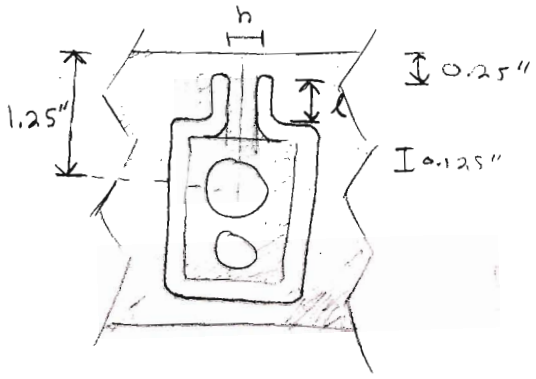
   FOUND 3/3/08

4/6/08

Problem: Guide shafts are overconstrained.

add flexure to support block.





Desired stiffness:

10 N / 0.005"

≈ 75 kN/m

$$\delta = \frac{P\ell^3}{3EI} \qquad \frac{P}{\delta} = \frac{3EI}{\ell^3} \qquad I = \frac{bh^3}{12}$$

$\ell = 0.56"$ max

$b = 0.125"$

$$\frac{P}{\delta} = \frac{(3E)\left(\frac{bh^3}{12}\right)}{\ell^3} = 75\times10^3$$

$h^3 = 3.87\times10^{-9}$ ← $h^3 = \dfrac{(75\times10^3)(0.0142)^3(12)}{(3)(70\times10^9)(0.00317)}$

$h = 0.00157\ m$

$= 0.062"$

Euler Buckling load:

$$F = \frac{\pi EI}{(2\ell)^2} = \frac{(\pi)(70\times10^9)(0.00317)(0.00157)^3}{12\,(2\cdot0.0142)^2}$$

$\boxed{F = 278\ N}$

Yield Strength

$$P = \sigma A = (200\times10^6)(0.00317)(0.00157) \approx \boxed{1000\ N}$$