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NuMI Pre-Target Tunnel – Magnet Stands	Mayling Wong	
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Abstract/Summary:
In the NuMI Carrier Tunnel and Pre-Target Tunnel, there are seven (7) 3Q120 magnets, two (2) 3Q60 magnets, four (4) B2 dipole magnets, and one (1) 200 turn bump magnet. The magnets rest on stand assemblies that are made up of three basic components: the cradle, the adjuster and the base. The cradle and adjuster designs are similar to the components used in the Main Injector magnet stands. The magnet stand drawings are listed in the bill of materials for the NuMI Pre-Target Tunnel Installation drawing (PPD/Mechanical Department 8875.119-ME-427811). This engineering note details the calculations of the allowable stresses and loads according to the AISC's Allowable Stress Design (ASD) and expected deflections.

Applicable Codes and References:

Manual of Steel Construction – Allowable Stress Design (ASD), American Institute of Steel Construction, Ninth Edition, 1989.

Hilti North America Product Technical Guide, Hilti Inc., 2002 Edition.

Avallone, E.A., et al, Mark's Standard Handbook for Mechanical Engineers, Tenth Edition, McGraw-Hill, 1996.

Rothbart, H.A., Mechanical Design Handbook, McGraw-Hill, 1996.

Shigley, J.E. and C.R. Mischke, Mechanical Engineering Design, Fifth Edition, McGraw-Hill, 1989.

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1 Introduction

This engineering note details the calculations of the mechanical stresses and loads on the magnet stands in the NuMI Carrier and Pre-Target Tunnels, between STA 2+50 and STA 6+50. There are seven (7) 3Q120 magnets, two (2) 3Q60 magnets, four (4) B2 dipole magnets, and one (1) 200 turn bump magnet. The B2 magnets are rotated 90° from their original orientation. The other magnets are used in a similar manner as previous applications. Table 1 lists the magnet's coordinates (which are corrected for the earth's curvature, according to the file NuMI_11Dec02_b_beam_LTCS_Z2H.xls) and pitch (from file numi_121102_b_ces0.xls).

Table 1 – Coordinates and Pitches of Magnets in NuMI Pre-Target Tunnel

Magnet	Earth Curvature Corrected coordinates			Pitch
Name	x (ft)	y (ft)	H (ft)	(deg)
HQ113 upstream iron end	100687.52145	97590.53540	667.08945	8.95017
HQ113 downstram iron end	100683.15212	97592.83849	666.31180	8.95017
HQ114 upstream iron end	100674.68171	97597.30325	664.80423	8.95017
HQ114 downstram iron end	100670.31238	97599.60630	664.02658	8.95017
HQ115 upstream iron end	100620.23359	97626.00286	655.11373	8.95017
HQ115 downstram iron end	100618.04892	97627.15441	654.72493	8.95017
HQ116 upstream iron end	100594.85719	97639.37882	650.59739	8.95017
HQ116 downstram iron end	100590.48782	97641.68187	649.81976	8.95017
HQ117 upstream iron end	100524.59993	97676.41144	638.09363	8.95017
HQ117 downstram iron end	100520.23060	97678.71453	637.31602	8.95017
H117	100511.78818	97683.16454	635.81356	8.95017
HQ118 upstream iron end	100506.15792	97686.13454	634.81134	8.95017
HQ118 downstram iron end	100501.78953	97688.43943	634.03375	8.95017
B2_UPS	100479.11415	97700.40360	629.99741	8.95017
V118_1	100478.57721	97700.68691	629.90186	8.24916
B2_DNS	100461.14421	97709.88505	627.04530	7.54815
B2_UPS	100460.66015	97710.14043	626.97281	7.54815
V118_2	100460.12128	97710.42475	626.89209	6.84742
B2_DNS	100442.63168	97719.65271	624.51856	6.14669
HQ119 upstream iron end	100440.02106	97721.03014	624.20084	6.14669
HQ119 downstram iron end	100437.82267	97722.19003	623.93330	6.14669
B2_UPS	100434.19529	97724.10393	623.49182	6.14669
V118_3	100433.65484	97724.38911	623.42608	5.44596
B2_DNS	100416.11911	97733.64133	621.53696	4.74524
B2_UPS	100415.63249	97733.89809	621.49133	4.74524
V118_4	100415.09079	97734.18389	621.44050	4.04422
B2_DNS	100397.51940	97743.45494	620.03697	3.34321
HQ120 upstream iron end	100396.37892	97744.05666	619.96171	3.34321
HQ120 downstram iron end	100391.96420	97746.38595	619.67039	3.34321
HQ121 upstream iron end	100386.30466	97749.37202	619.29692	3.34321
HQ121 downstram iron end	100381.88995	97751.70132	619.00559	3.34321

Each magnet is supported by two stands. Each stand is made up of three components: the cradle, the adjuster, and the base. The cradle and the base are unique to each magnet due to its pitch and distance above the tunnel floor. All quads and the bump magnet use the same adjuster, with the threaded supports 17-inches apart. The B2 magnets use an adjuster that is similar in design but with its threaded supports closer to each other at 9-inches. Table 2 lists the names and part numbers of each stand assembly. A complete bill of materials for the Installation Assembly (ME-427811), including drawing and manufacturing status of each item can be found at http://home.fnal.gov/~mlwong/bom.xls

Table 2 – Names and Part Numbers of Magnet Stands in NuMI Pre-Target Tunnel

Magnet Name	**Stand Assembly Name**	**Stand Assembly Number**
HQ113, HQ114	HQ113, HQ114 9° DNS STAND ASSY	MD-431070
	HQ113, HQ114 9° UPS STAND ASSY	MD-431069
HQ115, HQ116	HQ115, HQ116 DNS STAND ASSY	MD-431068
	HQ115, HQ116 UPS STAND ASSY	MD-431067
HQ117	HQ117 DNS STAND ASSY	MD-431066
	HQ117 UPS STAND ASSY	MD-431065
HQ118	HQ118 DNS STAND ASSY	MD-431064
	HQ118 UPS STAND ASSY	MD-431063
HQ119	HQ119 DNS STAND ASSY	MD-431062
	HQ119 UPS STAND ASSY	MD-431061
HQ120	HQ120 DNS STAND ASSY	MD-431060
	HQ120 UPS STAND ASSY	MD-431059
HQ121	HQ121 DNS STAND ASSY	MD-431058
	HQ121 UPS STAND ASSY	MD-431057
H117	H117 200 TURN BUMP MAGNET STAND ASSY	MD-431096
B2VB1	B2VB1 STAND ASSY	MD-431075
B2VB2	B2VB2 STAND ASSY	MD-431074
B2VB3	B2VB3 STAND ASSY	MD-431073
B2VB4	B2VB4 STAND ASSY	MD-431072

Table 3 lists the weights of each type of magnet. The B2 magnet is the heaviest magnet to be installed in the pre-target tunnel.

Table 3 – Weight of Magnets in NuMI Pre-Target Tunnel

Magnet Type	Weight (lb)
3Q120	8050
3Q60	4025
B2	26000
200 Turn Bump Magnet	2100

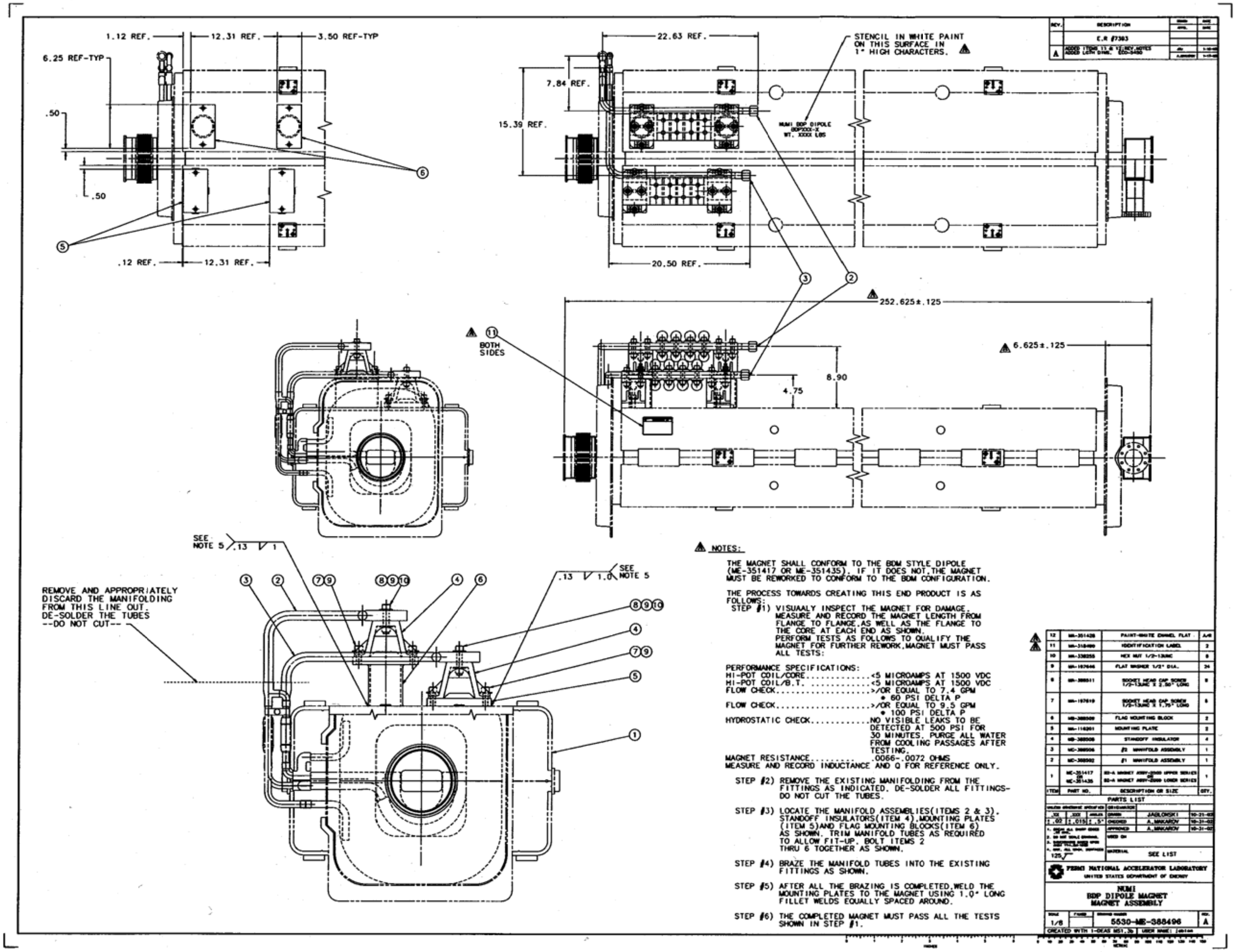


Figure 1 – B2 Magnet, Drawing 5530-ME-388496

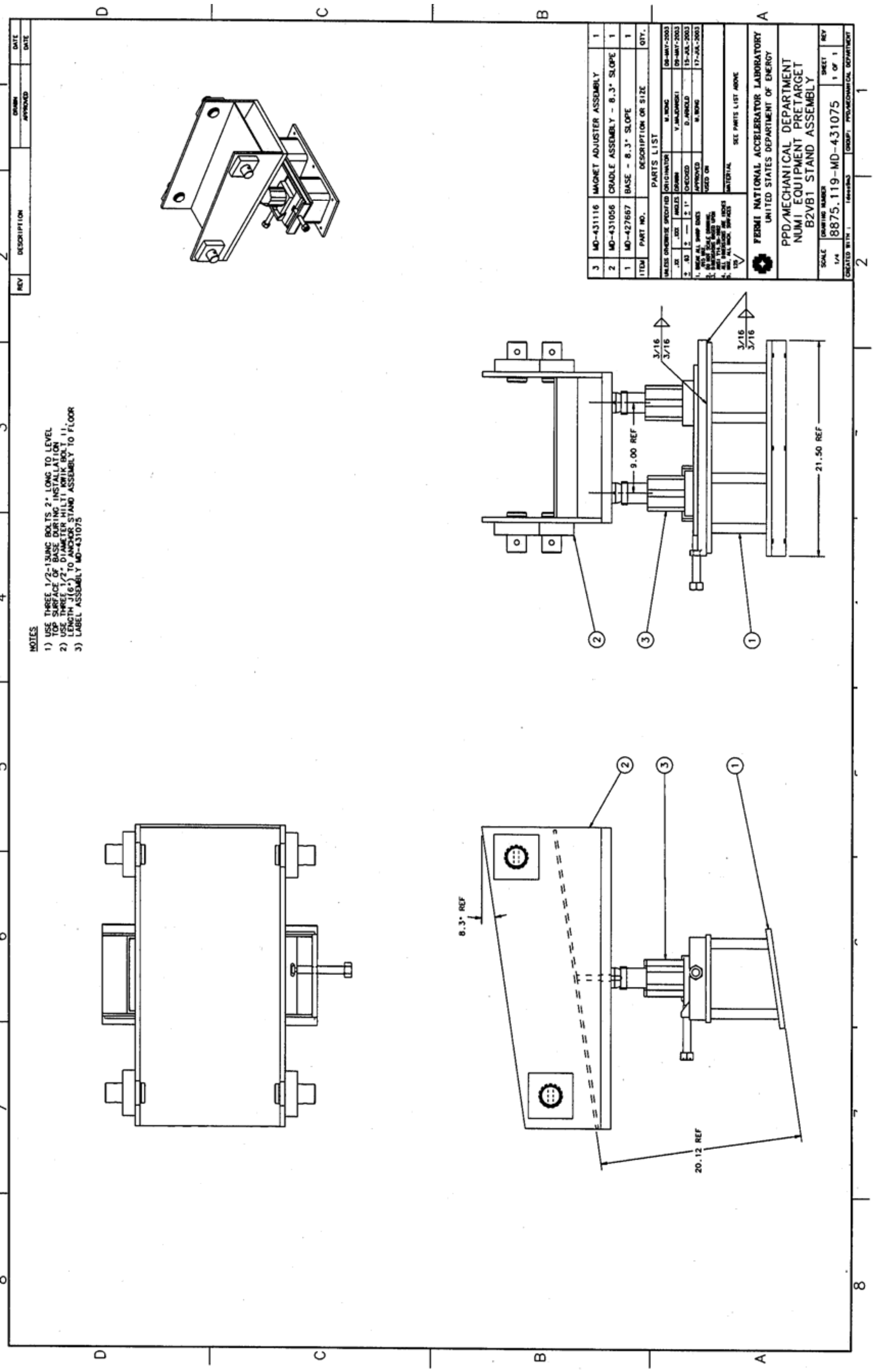


Figure 2 – Typical B2 Magnet Stand (Drawing 8875.119-MD-431075)

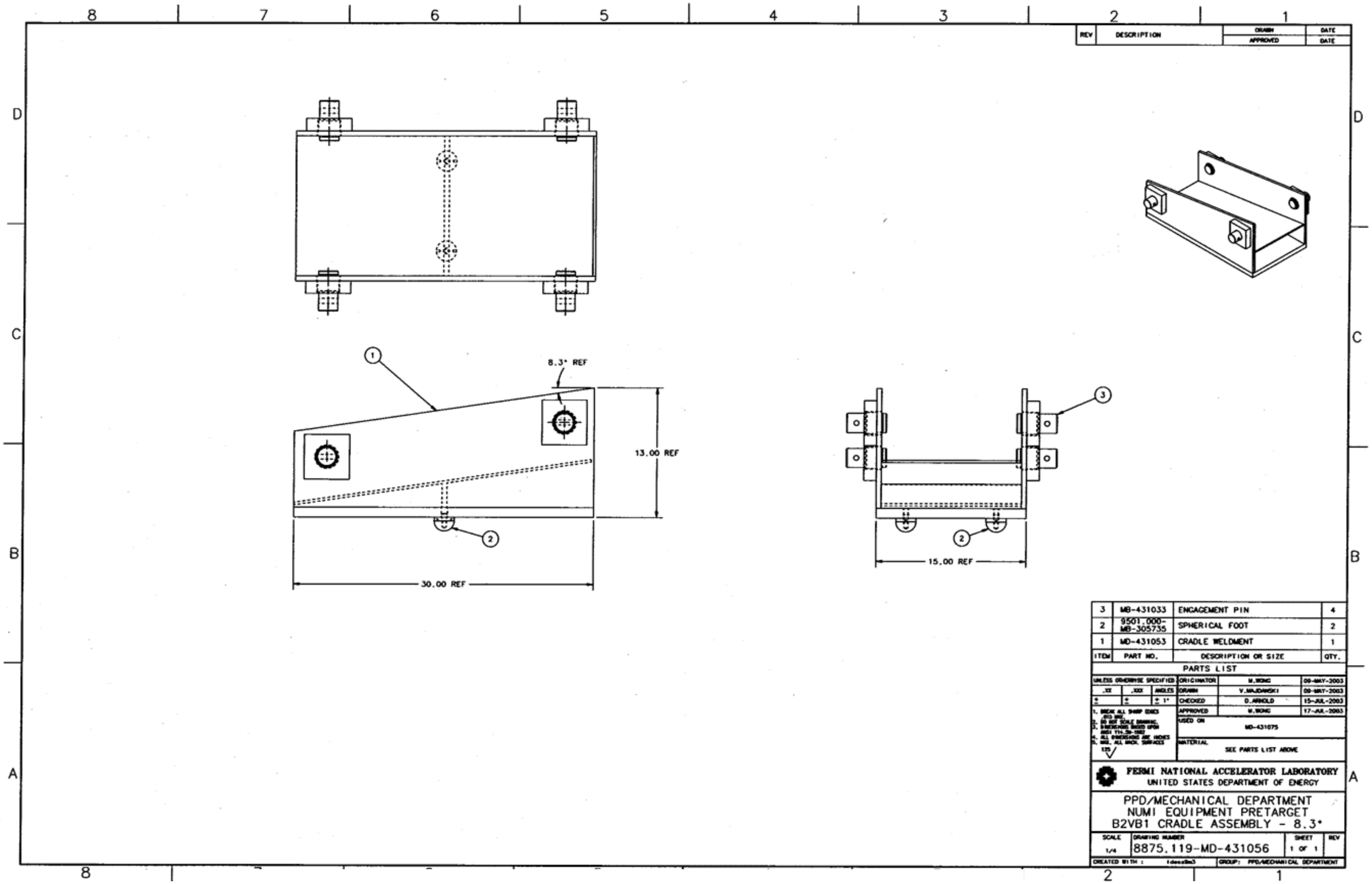


Figure 3 - Cradle for B2 Magnet Stand (Drawing 8875.119-MD-431056)

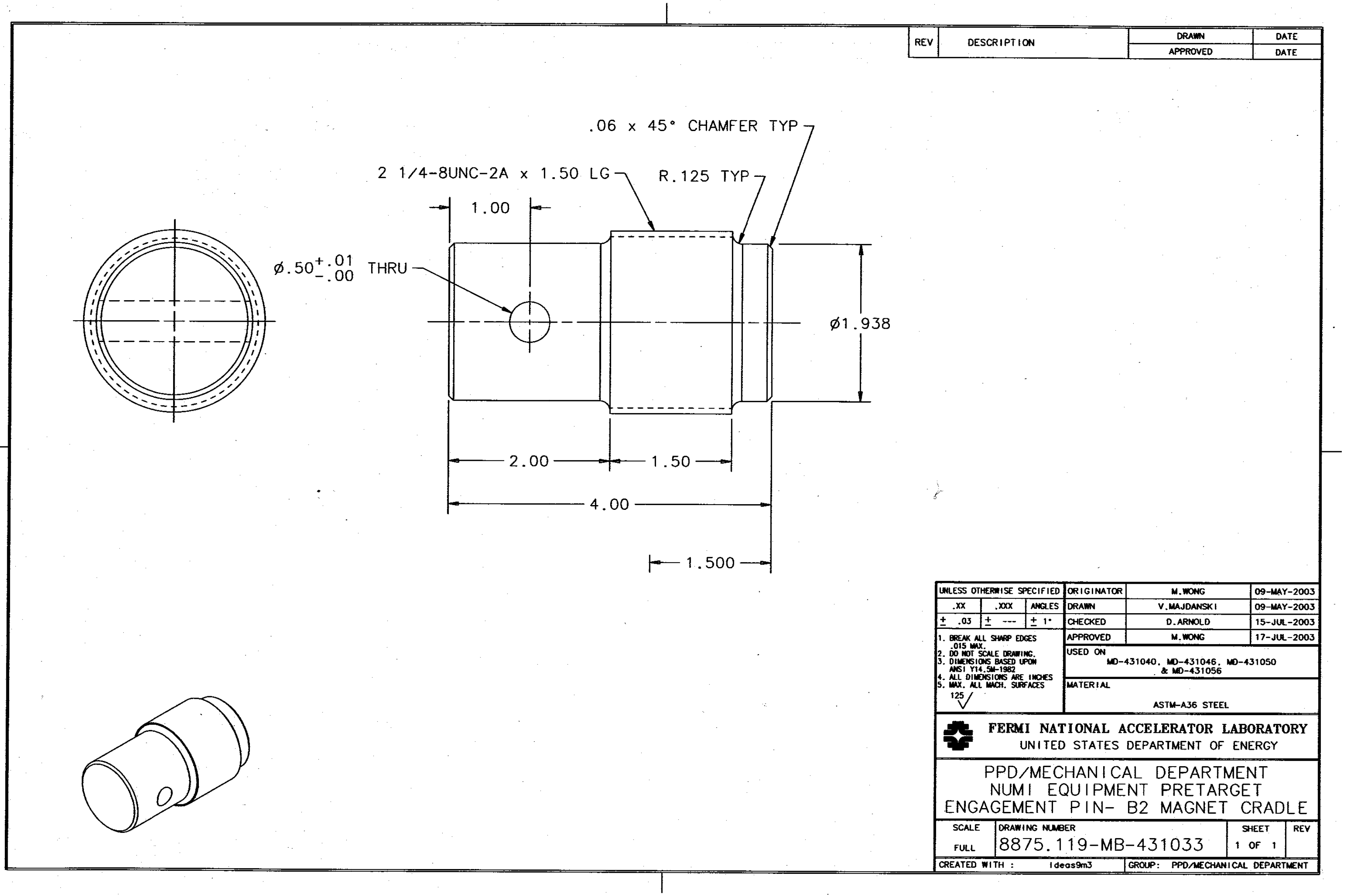


Figure 4 –Engagement Pin Used in Cradle for B2 Magnet Stand (Drawing 8875.119-MB-305735)

2 B2 Magnet Stand

Figure 1 shows a drawing of the B2 magnet (drawing number 5530-ME-388496). Figure 2 shows a typical B2 magnet stand (8875.119-MD-431075). With the weight of the B2 magnet at 26,000 lb (Table 3), the vertical stand load for each of two stands is:

P_{vert} = 26000 lb/2 = 13000 lb.

2.1 Cradle Assembly

Figure 3 shows the front and side views of the cradle assembly for the B2 magnet (drawing 8875.119-MD-431056).

2.1.1 Engagement pin shear stress, bearing strength and bending stress

Four steel (ASTM A36) engagement pins hold the B2 magnet inside the cradle. The pins extend 0.5-inch into holes in the magnet iron. With a cradle supporting a total vertical load of 13,000-pounds, each pin has a load force of F = 0.25*13000 = 3250 lb. Figure 4 shows the minor diameter of the threaded pin is 2.0094-inch (radius 1.0047-inch). Figure 5 shows how the pin is loaded.

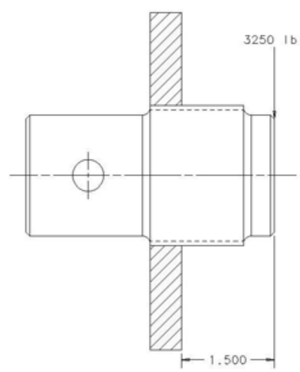


Figure 5 – Pin Loaded By Magnet Weight

Each pin experiences 1025-psi in shear:

$\tau = F/A = (3250)/(\pi*1.0047^2) = 1025$ psi

The allowable shear (ASD Section F4) is $\tau_{allow} = 0.40*\sigma_{yield} = 0.40(36000) = 14{,}400$ psi. The engagement pin design falls well within the allowable shear.

Assuming the smaller radius of the pin (r = 0.969-inch), the bending moment, deflection and bending stress on the pin are:

M = 3250 lb * 1.5 in
M = 4875 in-lb.

$\delta = Ml^2 / 3EI$
where l = 1.5 in.

$E = 28{,}000$ ksi
$I = (\pi r^4)/4 = (\pi * 0.969^4)/4 = 0.69\ in^4$
$\delta = 2x10^{-4}$ in.

$\sigma = My / I$
where $y = 0.5$ in.
$\sigma = 3233$ psi.

The allowable bending stress for a solid round bar (ASD Section F2.1) is $\sigma_{allow} = 0.75 * \sigma_{yield} = 0.75(36000) = 27{,}000$ psi. The bending stress of 3233 psi on the engagement pin is well within allowable.

2.1.2 Bending of side plates

When the magnet sits in the cradle, the side plates will experience a bending moment. The pins will extend 1.5-inch from the side plates when engaged in the magnet. The maximum bending will occur at the end of the plate furthest from the weld. With each plate loaded in bending due to 6500 lb, this gives a bending moment, deflection, and bending stress on the wall:

$M = 6500$ lb * 1.5 in
$M = 9750$ in-lb.

$\delta = Ml^2 / 3EI$
where $l = 7$ in.
$E = 28{,}000$ ksi
$I = (bh^3)/12 = (30 * 0.5^3)/12 = 0.3125\ in^4$
$\delta = 0.018$ in.

$\sigma_B = My / I$
where $y = 0.25$ in.
$\sigma_B = 7800$ psi.

The normal stress on the plate due to the load is:

$\sigma_N = Pvert / lh = 6500$ lb / (7 in.* 0.5 in)
$\sigma_N = 1800$ psi.

The allowable stress for a solid rectangular section (ASD Section F2.1) is $\sigma_{allow} = 0.75 * \sigma_{yield} = 0.75(36000) = 27{,}000$ psi. The total stress on the side plate of the cradle is $\sigma = \sigma_B + \sigma_N = 9600$ psi, which is well within allowable.

The plate experiences a bearing stress at each pin. The pins will extend 0.5-inch from the side plates when engaged in the magnet. The bearing stress at the hole is calculated from the bearing load of 3250 lb:

$\sigma_{bearing\_hole} = F / 0.5Ct$

Where C = hole circumference = pd = 6.3 in.

t = hole thickness = 0.5 in.

$\sigma_{bearing\_hole} = 2069$ psi.

According to the ASD (Section J3.7), for all connections with a single bolt in the line of force, the allowable bearing stress must be less than 1.2 times the ultimate stress. For ASTM A36 steel, $\sigma_{ultimate} = 58{,}000$ psi. The allowable bearing stress is calculated as:

$\sigma_{allow\text{-}bearing} = L_e F_u / 2d \le 1.2\sigma_{ultimate}$

where L_e = distance from the free edge to the bolt center = 3.0 in.

$F_u = \sigma_{ultimate} = 58{,}000$ psi

d = bolt diameter = 2.25 in.

$\sigma_{allow\text{-}bearing} = 38{,}667$ psi. $\le 69{,}600$ psi

According to the ASD, the hole where the engagement pin sits will experience a bearing stress within the allowable.

The holes in a side plate sit a minimum of 3-inches from the edge. For a 2-inch bolt, the minimum edge distance, according to Table J3.5 in the ASD, is 3.125-inch.

2.1.3 Stresses on spherical feet and spherical pad

The steel 2-inch diameter spherical feet (drawing 9501.000-MB-305735, item 2 in Figure 3) transfer the load from the cradle to the adjuster. The feet rest in a brass spherical pad (drawing 9501.000-MB-305047) of the adjuster assembly (see Figure 8). Each set is loaded with 6500-pound in compression. The contact stresses for components are calculated using the equations in the textbook Mechanical Engineering Design by Shigley and Mischke. For two spherical surfaces that are pressed together with a force, a circular area of contact radius a results. Given the material of each surface, the radius is calculated:

$$a = \sqrt[3]{\frac{3F}{8} * \frac{\left(1-\upsilon_1^2\right)/E_1 + \left(1-\upsilon_2^2\right)/E_2}{1/d_1 + 1/d_2}}$$

where F = pressing force = 6500 lb.

υ_1 = Poisson's ratio of spherical foot (steel) = 0.3 psi

E_1 = modulus of elasticity of spherical foot (steel) = 30,000 ksi

d_1 = radius of spherical foot (steel) = 2 in.

υ_2 = Poisson's ratio of spherical pad (brass) = 0.3 psi

E_2 = modulus of elasticity of spherical pad (brass) = 15,000 ksi

d_2 = radius of spherical pad (brass) = -2.006 in. (negative for an internal surface)

a = 0.53 in.

The maximum pressure occurs at the center of the contact area and is calculated:

$$p_{max} = \frac{3F}{2\pi a^2}$$

p_{max} = 11,076 psi.

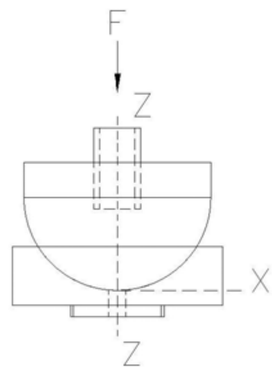


Figure 6 – Spherical Foot and Spherical Pad

The maximum normal stresses, or principal normal stresses, occur along the z-axis (see Figure 6) and are calculated as a function of the distance along the z-axis:

$$\sigma_x = \sigma_y = -p_{max}\left[\left(1 - \frac{z}{a}\tan^{-1}\frac{a}{z}\right)(1+\mu) - \frac{1}{2\left(1+\frac{z^2}{a^2}\right)}\right]$$

$$\sigma_z = \frac{-p_{max}}{1+\frac{z^2}{a^2}}$$

where z = distance from the contact point along the z-axis
μ = the Poisson's ration for the spherical surface under consideration.

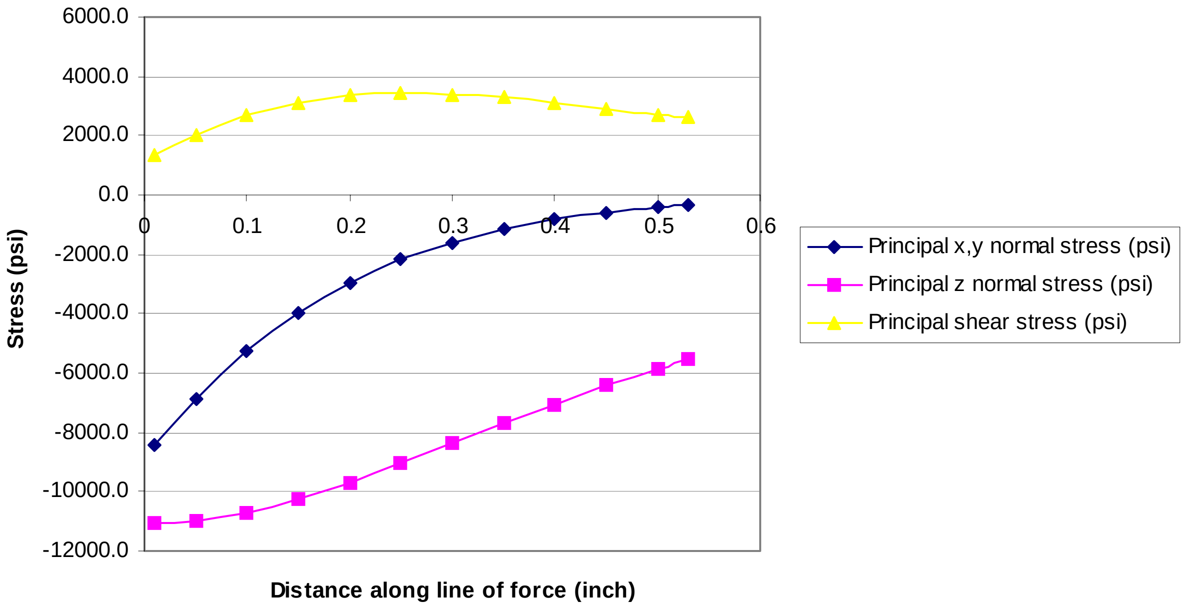
The principal shear stresses are calculated as well:

$\tau_{xy} = 0$ (since $\sigma_x = \sigma_y$)

$$\tau_{xz} = \tau_{yz} = \frac{\sigma_x - \sigma_z}{2} = \frac{\sigma_y - \sigma_z}{2}$$

Figure 7 shows the stresses at the spherical foot and spherical pad. Since Poisson's ratio is equal for both items, the stresses are identical.

Figure 7 - Principal Stresses at Spherical Ball Foot & at Spherical Pad



According with the ASD Section J8, the allowable bearing stress on the contact area is

$F_p = 0.9F_y$
where F_p = allowable bearing stress
F_y = material yield stress = 19,000 psi.
F_p = 17,000 psi.

The principal stresses on the spherical surfaces fall within the allowable limits.

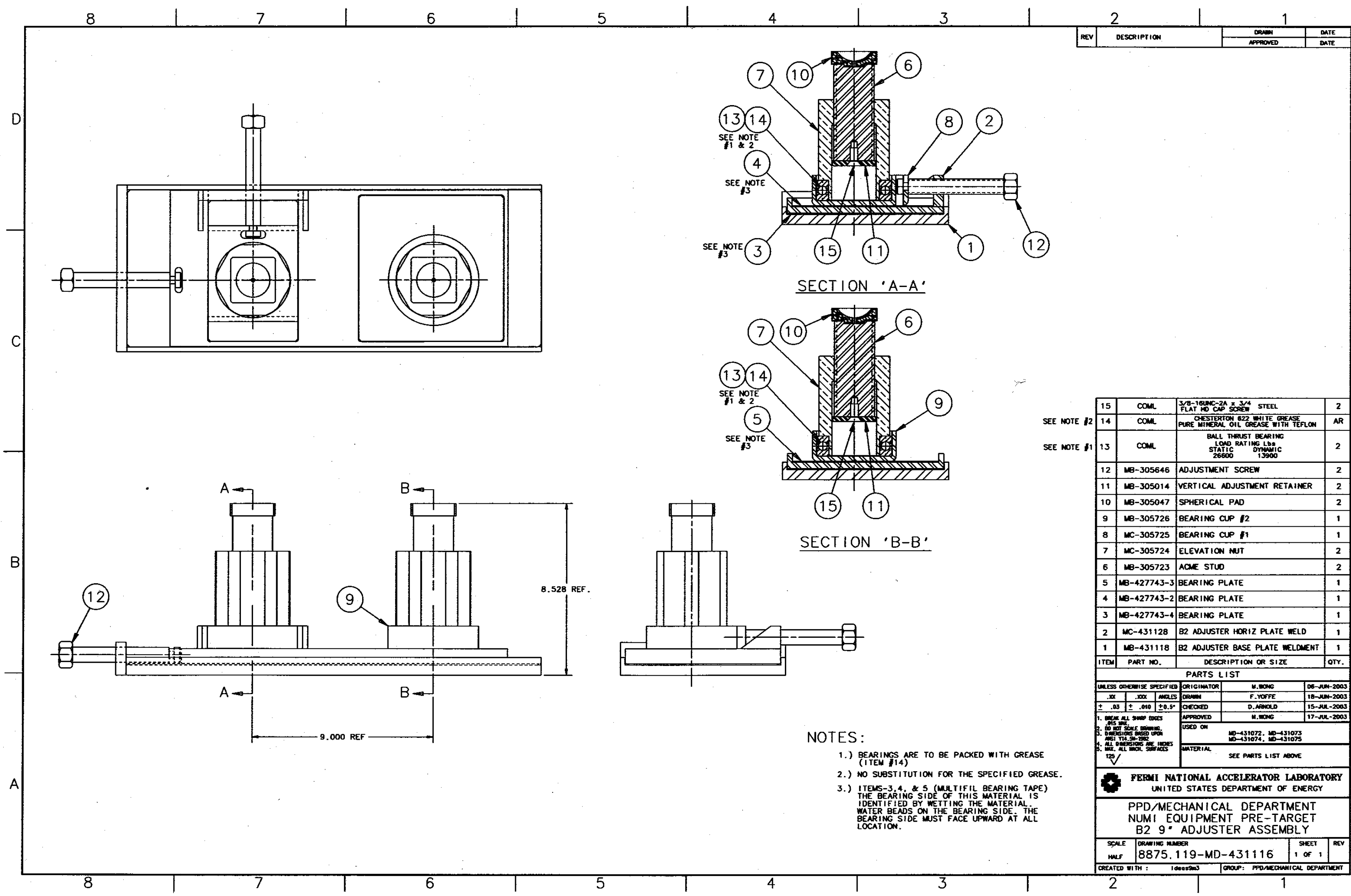


Figure 8 – Adjuster Assembly for B2 Magnet (8875.119-MD-431116)

2.2 Adjuster Assembly

Figure 8 shows the drawing of the adjuster assembly. The assembly is similar to the ones used in the Main Injector beam line.

2.2.1 Acme stud strength

Each Acme stud on the adjuster sees a load of 6500-pounds. The Acme studs have 2-4 Acme 2G threads and are made of 1020 cold finished steel (drawing 9501.000-MB-305723). Since the material of the stud is not a standard material in the LRFD, the textbook Mechanical Engineering Design is used as a guide to calculate the required torque on the screw, the bending stress on the threads, the shear stress on the threads, and the torsional shear stress on the root cylinder of the screw.

Assuming that three threads support the load, each thread supports an applied load (P_a):

$P_a = 6500 / 3 = 2167$ lb.

To calculate the required torque, the coefficient of friction in the system needs to be specified. The coefficient of friction between the threads is 0.20. The required torque T to raise the load is calculated from the torque load:

$$T = \frac{P_a d_m}{2}\left[\frac{L_{lead} + \pi\mu d_m \sec(\alpha)}{\pi d_m - \mu L_{lead}\sec(\alpha)}\right]$$

where d_m = mean (pitch) diameter = d – p/2 = 2 – 0.5(0.25) = 1.875 in.
d = diameter of external thread = 2 in.
p = pitch = 0.25 in.
L_{lead} = lead length = p = 0.25 in.
μ = friction = 0.20
α = thread angle = 29°/2 = 14.5°

T = 510 in-lb.

The required torque T_{low} to lower the load is calculated:

$$T_{low} = \frac{P_a d_m}{2}\left[\frac{\pi\mu d_m \sec(\alpha) - L_{lead}}{\pi d_m + \mu L_{lead}\sec(\alpha)}\right]$$

T_{low} = 331 in-lb.

The term $\pi\mu d_m = 1.2$. Having $\pi\mu d_m > L_{lead}$ indicates that the screw is self-locking.

To analyze the threads in bending, the threads are treated as beams and load applied at the end of the beam. Assuming that three threads are under a total load of 6500-lb, the bending stress at the root of the external thread is:

$$\sigma_{bend\_thread} = \frac{M}{S}$$

where $M = F * \frac{p}{2}$

$$S = n\frac{bt^2}{6} = n\frac{\pi d\left(p/2\right)^2}{6}$$

F = total load on stud = 6500 lb.

n = number of threads = 3

σ_{bend_thread} = 16,552 psi

The shear stress at the thread is

$$\sigma_{v\_thread} = \frac{F}{A} = \frac{F}{n\pi d(p/2)}$$

σ_{v_thread} = 2759 psi

According to the ASD Section F2.2, the allowable bending stress is $\sigma_b = 0.60\sigma_y$, where σ_y is the yield stress of the material.

$\sigma_b = 0.60\sigma_y$

where σ_y = yield stress of 1020 steel (hot rolled) = 30,000 psi

σ_b = 18,000 psi > σ_{bext}

The allowable shear stress, according to the ASD Section F4, is $\sigma_v = 0.40\sigma_y$.

$\sigma_v = 0.40\sigma_y$

σ_v = 12,000 psi. > σ_{vext}

The torsional shear stress σ_{vt} is calculated on the root cylinder of the screw based on the torque needed to raise the load:

$$\sigma_{vt} = \frac{16T}{\pi d_m^3}$$

σ_{vt} = 394 psi. < σ_v

2.2.2 Acme stud compressive buckling load

The Acme stud is 4.75-inch long. With its basic minor diameter of 1.75-inch, the critical buckling load of the stud, assuming one end is fixed and the other end has a load of 6500-pounds, is

$$P_{cr} = \frac{\pi^2 EI}{4L^2}$$

where E = modulus of elasticity of 1020 steel = 28,000 psi
I = moment of inertia = $\pi r^4/4 = 0.46\ in^4$
L = 4.75 in.

$P_{cr} = 1.4x10^6$ lb.

2.2.3 Elevation nut thread strength

Assuming that three threads are under load, the bending stress at the root of the external thread is:

$$\sigma_{b\_int} = \frac{6F}{n\pi d_{int}\left(p/2\right)}$$

where d_{int} = 2.04 in.

σ_{b_int} = 16,228 psi

The shear stress at the thread is

$$\sigma_{v\_int} = \frac{F}{n\pi d_{int}\left(p/2\right)}$$

σ_{v_int} = 2705 psi

According to the ASD Section F2.2, the allowable bending stress is $\sigma_b = 0.60\sigma_y$, where σ_y is the yield stress of the material.

$\sigma_b = 0.60\sigma_y$

where σ_y = yield stress of 1020 steel = 57,000 psi

σ_b = 34,200 psi > σ_{b_int}

The allowable shear stress, according to the ASD Section F4, is $\sigma_v = 0.40\sigma_y$.

$\sigma_v = 0.40\sigma_y$

σ_v = 22,800 psi. > σ_{v_int}

The length of the threaded part of the elevation nut is based on the minimum length of engagement Q is calculated based on the design of the Acme bolt and nut according to Machinery's Handbook:

$Q = JL_e$

where L_e = minimum length of engagement for mating internal and external Threads of the same material
J = factor for the relative strength of the external and internal threads of Different materials

$$L_e = \frac{2A_{t-screw}}{\pi K_n\left[0.5 + 0.57735n\left(E_s - K_n\right)\right]}$$

where K_n = maximum minor diameter of the internal thread = 1.75 in.
n = number of threads per inch = 4
$A_{t\text{-}screw}$ = stress area of Acme stud = 2.45 in^2

L_e = 1.3 in.

$$J = \frac{A_{t-screw}\sigma_{t-screw}}{A_{t-nut}\sigma_{t-nut}}$$

where $\sigma_{t\text{-}screw}$ = tensile strength of the screw = 75,000 psi
$A_{t\text{-}nut}$ = stress area of adjusting nut = 2.61 in.
$\sigma_{t\text{-}nut}$ = tensile strength of adjusting nut = 68,000 psi

J = 1.04

Q = 1.4 in.

2.2.4 Vertical and angular adjustment torque on Acme threads

Refer to the engineering note Main Injector 150 Line, B2 Magnet Support and Adjuster Design Documentation, T. Anderson (February 21, 1995) for calculations on the vertical and angular adjustment torque on the Acme threads of the stud and adjusting nut.

2.2.5 Horizontal and longitudinal adjustment torque on adjusting screw

Refer to the engineering note Main Injector 150 Line, B2 Magnet Support and Adjuster Design Documentation, T. Anderson (February 21, 1995) for calculations on the horizontal and longitudinal adjustment torque of the adjusting screw (drawing 9501-MB-305646). The assumption was made that analyzing the torque for the adjusting screw, which have Unified coarse threads, can be done in the same way as analyzing the torque for a power screw.

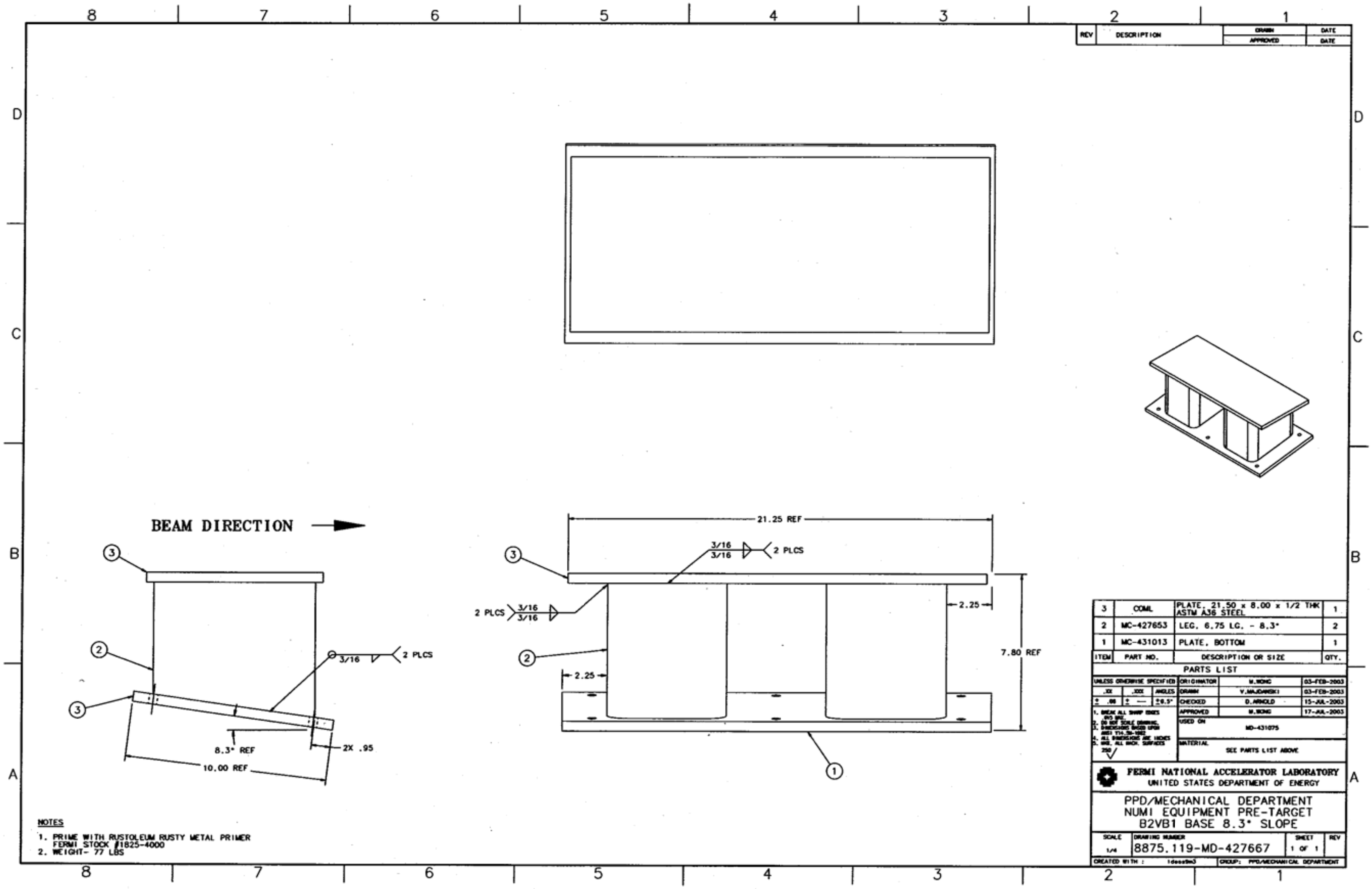


Figure 9 – Base of B2 Magnet Assembly (drawing MD-427667) for 8.°3 Slope

2.3 Base assembly

Figure 9 shows the drawing of the base for the B2 magnet stand that sits at an 8.3° slope (drawing MD-427667), which is the largest slope for the B2 magnet stand. The stand holds a load of half of the weight of B2 magnet, 13,000-pounds.

2.3.1 Deflection and bending stresses in top plate

The top plate in the base assembly has a distributed load of half of the weight of the B2 magnet divided by the distance between the legs of the base:

$w = P/L$

where $P = 13{,}000$ lb.

$L = 4.75$ in.

$w = 2737$ lb/in.

Assuming that the plate is held at the legs so that the plate ends are considered fixed, the plate has a maximum deflection at its center

$$y_{max} = \frac{wL^4}{384EI}$$

where E = modulus of elasticity of A36 steel = 28,000 ksi

b = cross section width = 21.25 in.

h = cross section height = 0.5 in.

I = moment of inertia for cross section = $bh^3/12 = 0.22$ in^4

$y_{max} = 0.0006$ in.

The maximum bending moment that the plate experiences is calculated:

$$M = \frac{wL^2}{12}$$

$M = 5146$ in-lb.

The largest normal stresses in the plate are:

$$\sigma_{max} = \frac{My}{I}$$

where y = half the thickness of the plate (farthest from the centroid) = 0.25 in.

$\sigma_x = 5712$ psi.

According to ASD Section F2.1, a beam that is bent about is weaker axis has an allowable stress of $\sigma_{allow} = 0.75*\sigma_{yield} = 0.75(36000) = 27{,}000$ psi.

2.3.2 Bending load on the legs

Figure 10 shows the dimension of the leg for the base.

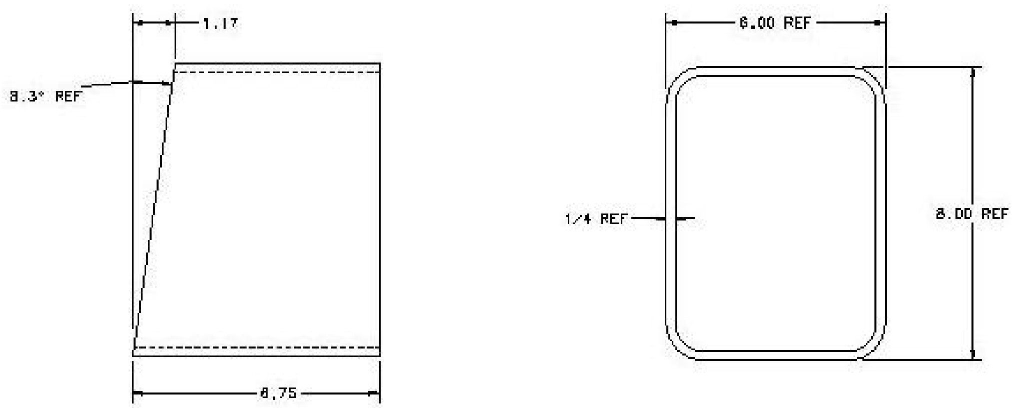
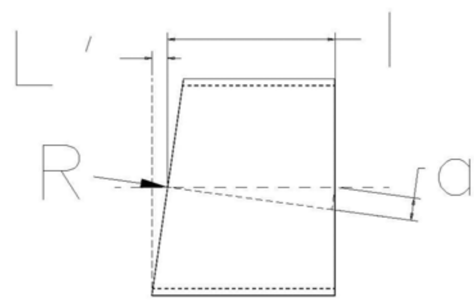


Figure 10 – Dimensions of Leg for 8.3° Slope Base

Each stand has two legs, and each leg experiences 6500-pounds along its axis due to the weight of the magnet. With the leg cut and resting at an angle, the leg experiences some bending. The equivalent bending moment on the leg is calculated from the reactionary force at the angled part of the leg:

$R = 6500 \cos 8.3°$
$R = 6432$ lb.



$M = R*a$
$a = l*\sin 8.3°$
$l = L - L'$
where $L = 6.75$-inch
$L' = 4*\sin 8.3°$
$L' = 0.58$ in.
$l = 6.17$ in.
$a = 0.89$ in.
$M = 5729$ in-lb.

The stress in the leg is calculated:

$$\sigma = \frac{My}{I}$$

where $y = 4$ in.
$I = (bd^3 - b_i d_i^3)/12 = 62.6\ \text{in}^4$

$\sigma = 366$ psi.

The allowable bending stress (ASD Section F2.2) is $\sigma_{allow} = 0.60*\sigma_{yield} = 0.60(36000) =$ 21,600 psi. Thus, the bending that the leg experiences is well within the allowable under normal load conditions by the weight of the magnet.

Suppose the magnet, while sitting on the stands, is loaded with a force that is perpendicular to the beam line, as if an object is accidentally bumped up against the magnet. The force will cause a bending moment in the stands such that a leg is under compression. Let the force amount be 10% of the weight of the B2 magnet, so that F = 2600-pounds. If the force is applied at a central spot relative to the height of the magnet, a distance of 25.375-inches from the top of the base of the magnet stand. Each stand is experiencing an equivalent moment that is applied at the center of the top surface of the stand:

$M_{accident} = F_{accident} * D$

where $F_{accident} = 2600$ lb. / 2

D = distance from force to top of base = 25.375 in.

$M_{accident} = 32{,}988$ lb.

The moment at the top of the stand is the equivalent of one of the legs experiencing a force along its axis (the other leg is in tension in the same direction):

$F_{leg} = M_{accident} / x$

where x = distance from center of stand to axis of leg = 5.375 in.

$F_{leg} = 6137$ lb.

As previously analyzed, the force is causing a bending moment in the leg. Following the same analysis as above, the stress caused by the bending moment in the leg is due to the combined forces from the magnet weight and the accidental load:

$R_{total} = (6500+6137) \cos 8.3°$

$R_{total} = 12{,}505$ lb.

$M_{total} = R_{total}*a$

$M_{total} = 11{,}129$ in-lb.

$$\sigma_{total} = \frac{M_{total}\,y}{I}$$

$\sigma_{total} = 711$ psi.

The total stress on the leg due to the magnet weight and the accidental load of 10% of the magnet weight is well within the allowable bending stress.

2.3.3 Shear load on anchor bolts

The Hilti Kwik II anchor bolts that hold the base to the tunnel floor will have a shear force acting on them. The magnitude of the shear force is calculated:

$V = 6500 \sin 8.3°$
$V = 938$ lb.

The anchor bolts have a diameter of ½ inch and an allowable shear load of 1940 lb each. Thus for just one bolt the shear load is within allowable according to the manufacturer.

2.3.4 Tensile load on anchor bolts

As described in Section 2.3.2, the stands are designed to accommodate an accidental loading on the magnet of 10% of its weight. Such a loading scenario would cause the anchor bolts to be loaded in tension. The combined tensile load on the two anchor bolts is 6137-pounds. Thus, each anchor bolt would experience a tensile load of 3068-pounds. While the allowable tensile load for a carbon steel bolt of ½ inch diameter is 2080-pounds (with 6-inch embedment), the ultimate tensile load is 7800-pounds. Since the bolts were selected based on static loading from the magnet weight, they will be used. However, it is recognized that a large accidental loading of 10% of the magnet weight as described in Section 2.3.2 will load the bolts within the ultimate tensile load.

2.3.5 Torque requirement and thread strength of leveling bolts

Three bolts are threaded through the bottom plate of the base and are used to help level the base during installation. The bolts are ¾ -13UNC 2-inch long. During installation of the magnet stand, bolts will be in compression due to the weight of the base assembly and the adjuster assembly (the cradle is attached to the magnet). Thus the bolts must support a weight of 200-pounds. Assuming one bolt supports the entire load, let the applied load P_a = 200-pounds. With the coefficient of friction between steel surfaces being $\mu = 0.2$, the required torque T to raise the load is calculated from the torque load:

$$T = \frac{P_a d_m}{2}\left[\frac{L_{lead} + \pi\mu d_m \sec(\alpha)}{\pi d_m - \mu L_{lead} \sec(\alpha)}\right]$$

where d_m = mean (pitch) diameter = 0.6875 in.
d = diameter of external thread = 0.75 in.
L_{lead} = lead length = 0.077 in.
μ = friction = 0.20
α = thread angle = 60°/2 = 30°

$T = 43$ in-lb.

The term $\pi\mu d_m = 1.1$. Having $\pi\mu d_m > L_{lead}$ indicates that the screw is self-locking.

For this analysis, the threads are treated as beams in bending and the load applied at the end of the beam. Assuming that three threads are under load, the bending stress at the root of the external thread is:

$$\sigma_{bext} = \frac{6F\left(p/2\right)}{\pi n_{load} d b^2}$$

where F = total load on the bolt = 200 lb.
n_{load} = 3
p = pitch = 0.010 in.
b = width of thread at root = 0.0541 in.

σ_{bext} = 2900 psi

The shear stress at the thread is

$$\sigma_{vext} = \frac{F}{\pi n_{load} d b}$$

σ_{vext} = 490 psi

According to the ASD Section F2.2, the allowable bending stress is $\sigma_b = 0.60\sigma_y$, where σ_y is the yield stress of the material.

$\sigma_b = 0.60\sigma_y$
where σ_y = yield stress of carbon steel = 36,000 psi
σ_b = 21,600 psi > σ_{bext}

The allowable shear stress, according to the ASD Section F4, is $\sigma_v = 0.40\sigma_y$.

$\sigma_v = 0.40\sigma_y$
σ_v = 14,400 psi. > σ_{vext}

The torsional shear stress σ_{vt} is calculated on the root cylinder of the screw:

$$\sigma_{vt} = \frac{16T}{\pi d_m^3}$$

σ_{vt} = 674 psi. < σ_v

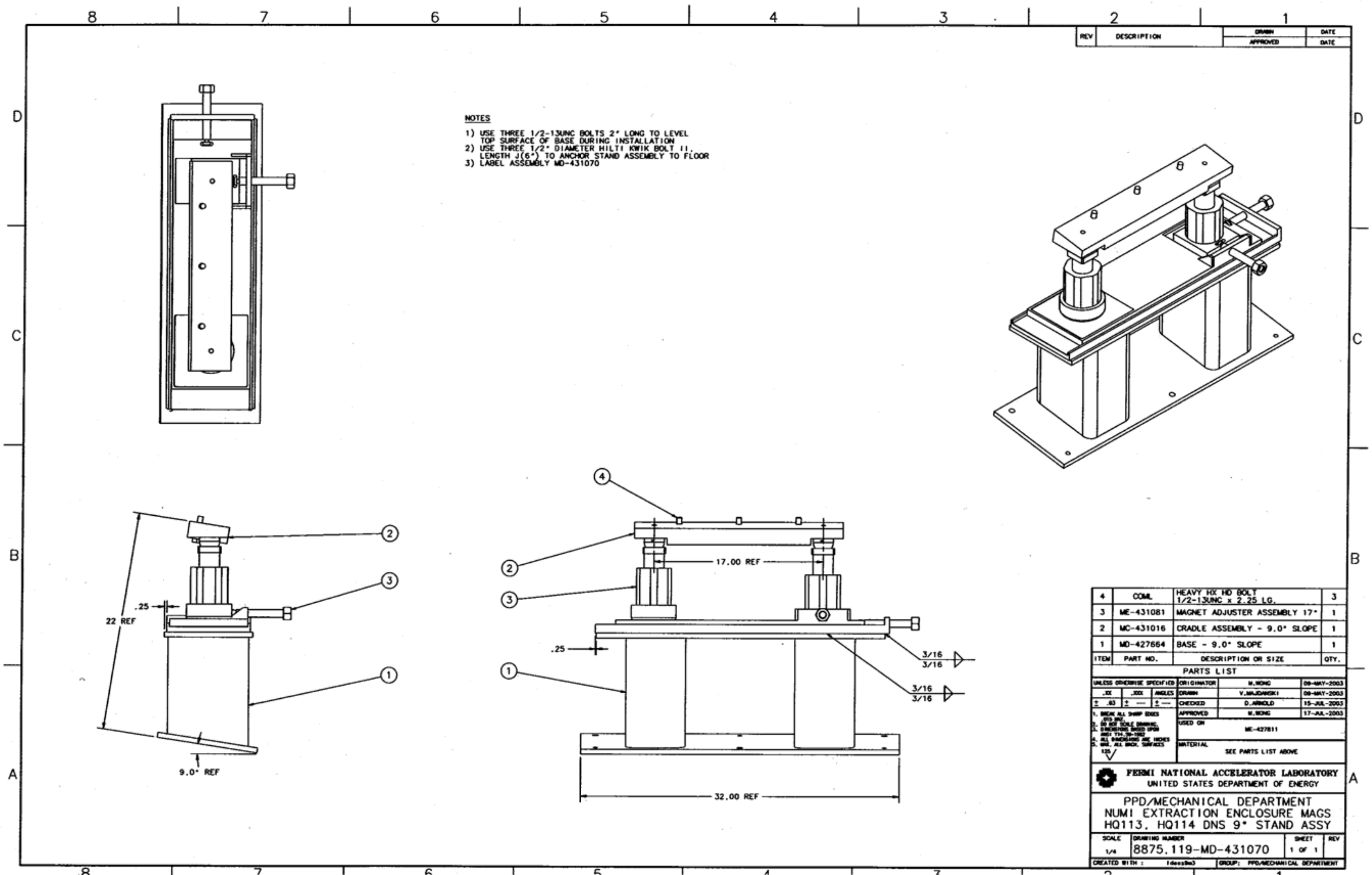


Figure 12 – Drawing of Typical Quad Magnet Stand (Drawing 8875.119-MD-431070)

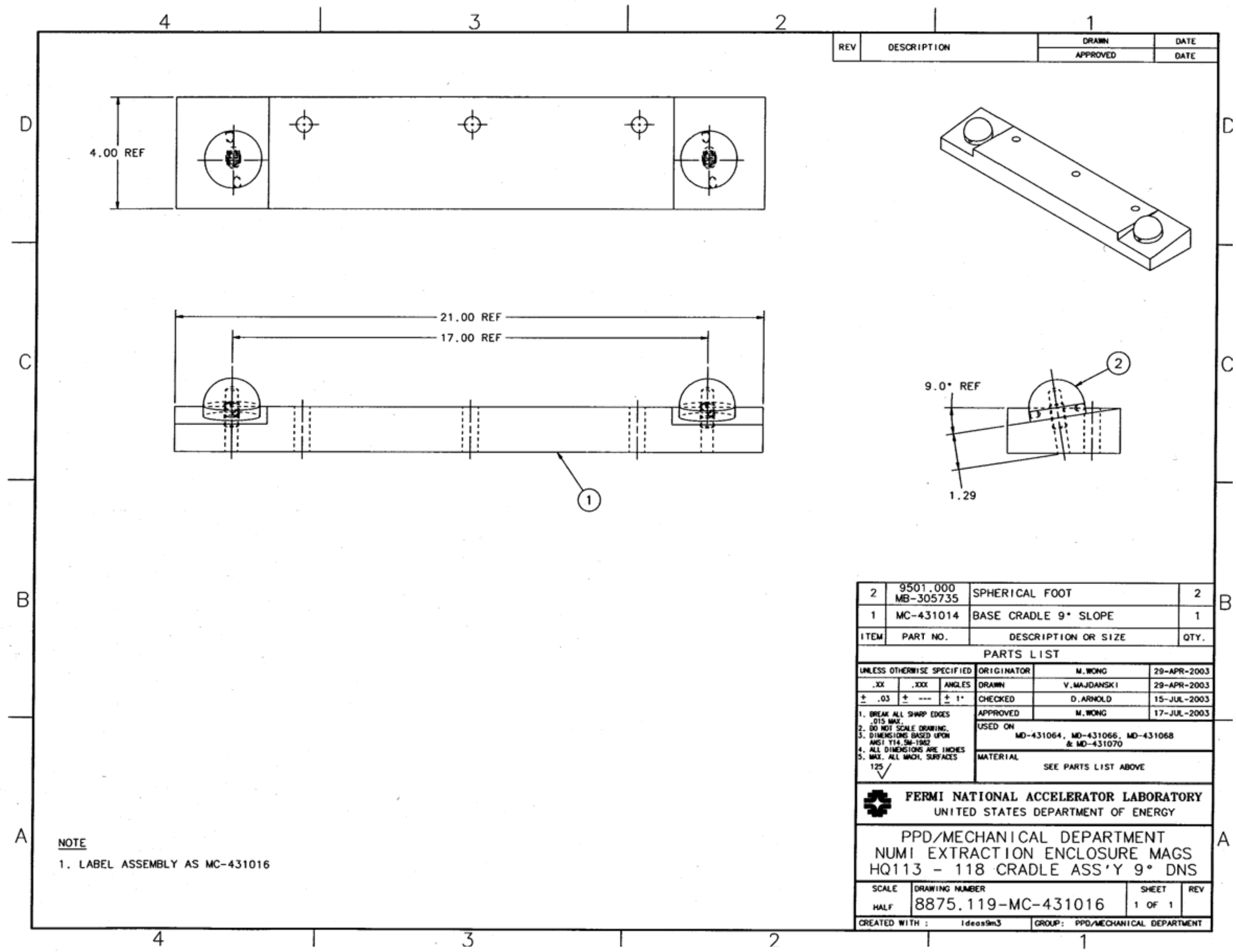


Figure 13 – Cradle Assembly of Quad Magnet Stand (Drawing 8875.119-MC-431016)

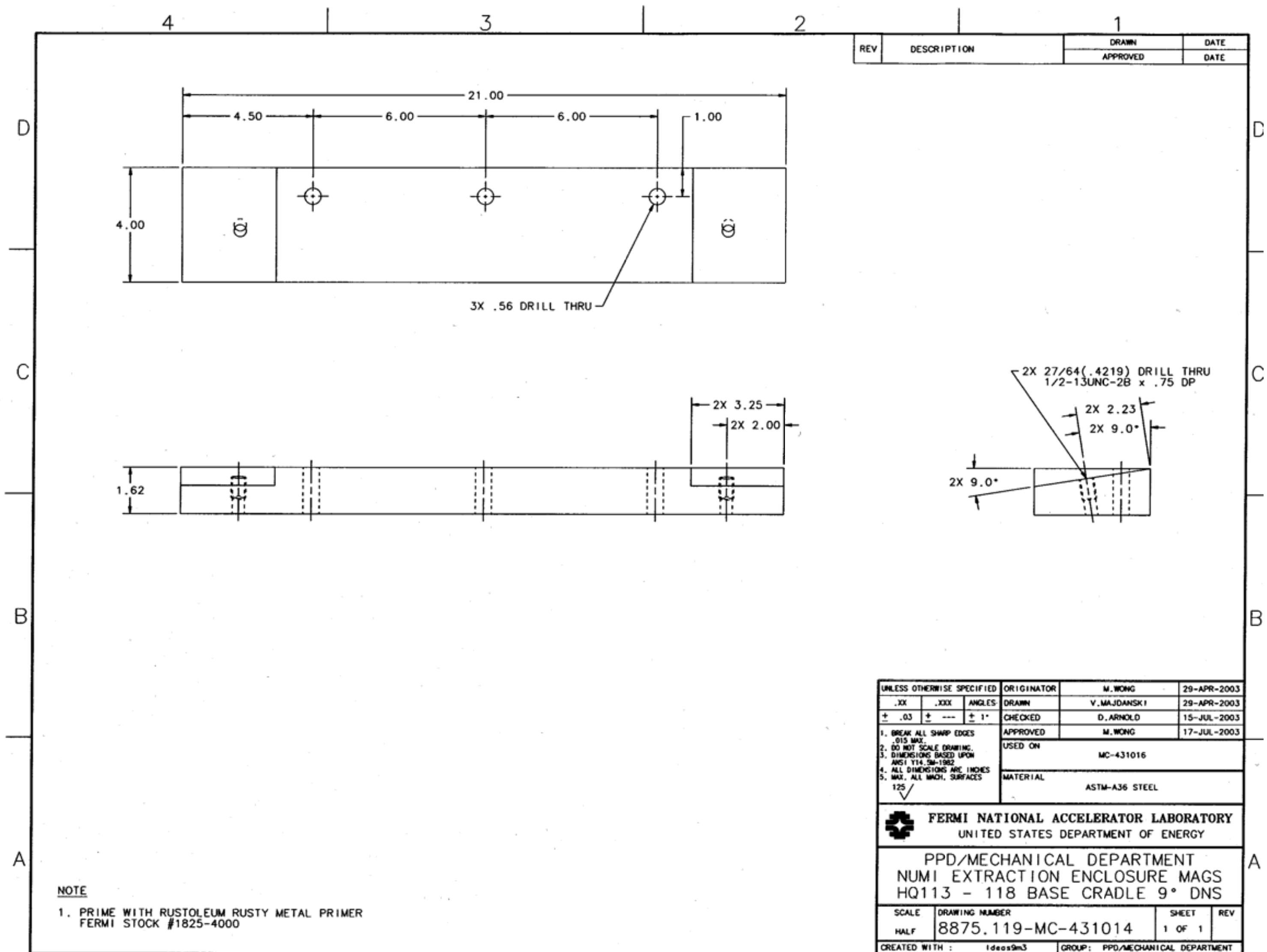


Figure 14 – Cradle Bar (Drawing 8875.119-MC-431014)

3 Quad magnet stand

Figure 12 shows a typical magnet stand (8875.119-MD-431070) that will support the quad magnets 3Q120 and 3Q60. Each magnet will have two stands. The analysis of the stand is based on the weight of the heavier quad magnet, which is 8050-pounds (Table 3). Each stand holds a vertical load of:

$P_{vert} = 8050 / 2 = 4025$ lb.

Since the components of the adjuster assembly are the same designs as the ones used in the B2 magnet stand, it is assumed that the components are designed within allowable for the quad magnet stand. See Section 2.2 for the analysis.

3.1 Cradle assembly

Figure 13 shows the front and side views of the cradle assembly for the 3Q120 and 3Q60 magnets sloped at approximately 9° (drawing 8875.119-MD-431016).

3.1.1 Bar deflection

The cradle base, as shown in Figure 14 (drawing 8875.119-MD-431014), is essentially a beam in bending due to the weight of the magnet. The load that the bar sees is evenly distributed along its length, resulting in a uniform load of:

$w = P / b$

where P = load of 3Q120 magnet on the cradle = 4025 lb.
b = width of the magnet = 17 in.

$w = 237$ lb/in.

The maximum bending moment that the cradle bar experiences is

$$M = \frac{wL^2}{8}$$

where L = distance between supports = 17 in.

$M = 8553$ in-lb.

The cross section can be analyzed by splitting it into two areas, A_1 and A_2 as shown Figure 15. To determine the normal stresses that the bar experiences, the moment of inertia is calculated as the sum of the centroidal moment of inertias of the two areas:

$$I_{xc} = I_{xc1} + I_{xc2}$$

$$I_{xc1} = I_{x1} + A_1 d_1^2$$
$$I_{xc2} = I_{x2} + A_2 d_2^2$$

$$I_{x1} = \frac{bh_1^3}{12} \qquad I_{x2} = \frac{bh_2^3}{36}$$

where $b = 4$ in.
$h = 1.62$ in.
$h_2 = b \tan\theta = 4 \tan(9°) = 0.63$ in.
$h_1 = h - h_2 = 0.99$

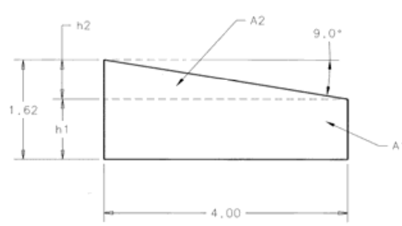


Figure 15 – Simple Cross Section

$I_{x1} = 0.323$ in^4
$I_{x2} = 0.028$ in^4

$A_1 = b*h_1 = 3.95$ in^2
$A_2 = 0.5bh_2 = 1.27$ in^2

The location of the centroid of the bar cross section is determined, based on Figure 16:

$y_1 = 0$
$$y_2 = \frac{h_2}{3} + \frac{h_1}{2} = 0.705 \text{ in.}$$
$A = A_1 + A_2 = 5.21$ in^2
$Q_x = y_1A_1 + y_2A_2 = 0.895$ in^3
$$\bar{y} = \frac{Q_x}{A} = 0.17 \text{ in.}$$

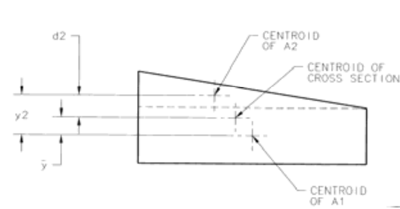


Figure 16 – Centroid of Cross Section

The values d_1, d_2 are then determined:

$$d_1 = \bar{y} = 0.17 \text{ in.}$$
$$d_2 = \frac{h_1}{2} - \bar{y} = 0.535 \text{ in.}$$

Finally, the moment of inertia for the cross section can be calculated:

$I_{xc1} = I_{x1} + A_1d_1^2 = 0.323 + (3.95)(0.17)^2 = 0.44$ in^4
$I_{xc2} = I_{x2} + A_2d_2^2 = 0.028 + (1.27)(0.535)^2 = 0.39$ in^4

$I_{xc} = I_{xc1} + I_{xc2} = 0.83$ in^4

Knowing this, the normal stress due to the bending moment is calculated. The largest stresses will take place at the location of the cross section that is farthest from the centroid, which is the top point in the section at a distance $y_{max} = 0.955$ inch.

$$\sigma_x = \frac{My_{max}}{I}$$

where $M = 8553$ in-lb.
$y_{max} = 0.955$ in.
$I = 0.83$ in^4

$\sigma_x = 9841$ psi.

According to ASD Section F2.1, a beam that is bent about is weaker axis has an allowable stress of $\sigma_{allow} = 0.75*\sigma_{yield} = 0.75(36000) = 27{,}000$ psi.

3.1.2 Shear load on support bolt

On each cradle assembly, three bolts hold the magnet in place. Each carbon steel bolt is ½ - inch UNC, with a minor diameter d = 0.4056-inch (radius r = 0.2028-inch). For a magnet at a 9° angle, the total shear load at one cradle assembly is F = 2034-pounds. The shear stress on one bolt is calculated based on one third of the shear load:

$\tau = F/A = (2034/3)/(\pi*0.2028^2) = 5247$ psi

The allowable shear (ASD Section F4) is $\tau_{allow} = 0.40*\sigma_{yield} = 0.40(36000) = 14{,}400$ psi. A support bolt in the cradle assembly falls well within the allowable shear.

3.1.3 Bearing load on spherical feet

The spherical feet in the cradle assembly for the quad magnets are the same size as but support less weight than the feet analyzed in section 2.1.3. Thus, it is assumed that the surface of the feet will experience principal stresses within the allowable limits as stated in section 2.1.3.

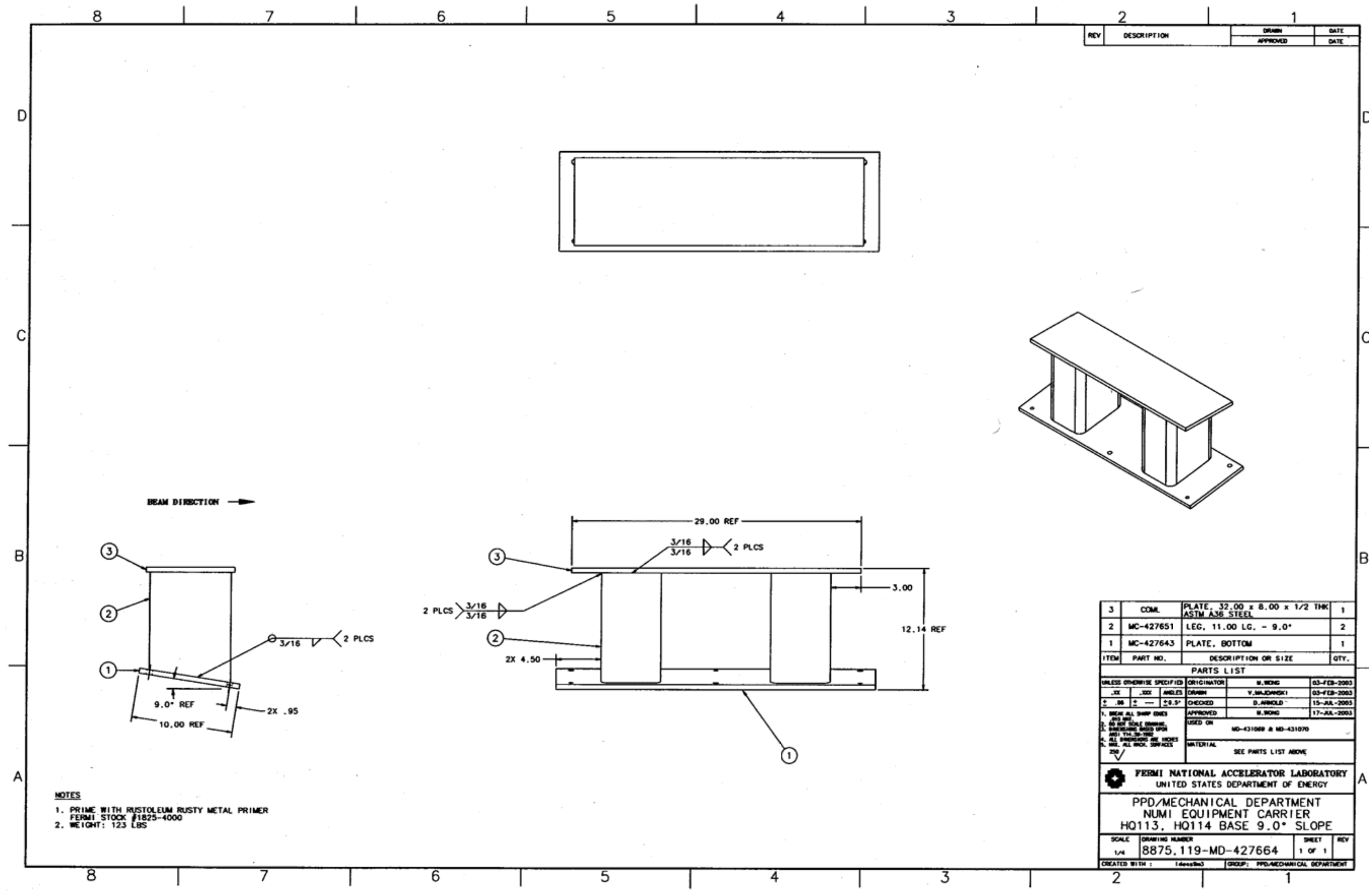


Figure 17 – Base Assembly of Quad Magnet (Drawing 8875.119-MD-427664)

3.2 Base assembly

Figure 17 shows drawing 8875.119-ME-427657 of the base assembly for the quad magnet stand. This stand has the largest slope of 8.5° at its bottom plate and is the tallest at 18.8-inches. Since the weight on this base is smaller than base in the B2 magnet stand and the slope is the same, the shear load on the anchor bolts and the threat strength of the leveling bolts are assumed to be within allowable, as analyzed in sections 2.3.2 and 2.3.3

3.2.1 Deflection and bending stresses in top plate

The top plate in the base assembly has a distributed load of half of the weight of the 3Q120 divided by the distance between the legs of the base:

$w = P_{vert} / L$

where $P_{vert} = 4025$ lb.

$L = 11$ in.

$w = 366$ lb/in.

Assuming that the plate is held at the legs so that the plate ends are considered fixed, the plate has a maximum deflection at its center

$$y_{max} = \frac{wL^4}{384EI}$$

where E = modulus of elasticity of A36 steel = 28,000 ksi

b = cross section width = 32 in.

h = cross section height = 0.5 in.

I = moment of inertia for cross section = $bh^3/12 = 0.33$ in^4

$y_{max} = 0.0015$ in.

The maximum bending moment that the plate experiences is calculated:

$$M = \frac{wL^2}{12}$$

M = 3690 in-lb.

The largest normal stresses in the plate are:

$$\sigma_{max} = \frac{My}{I}$$

where y = half the thickness of the plate (farthest from the centroid) = 0.25 in.

$\sigma_x = 2795$ psi.

According to ASD Section F2.1, a beam that is bent about is weaker axis has an allowable stress of $\sigma_{allow} = 0.75*\sigma_{yield} = 0.75(36000) = 27,000$ psi.

3.2.2 Bending load in the legs

The analysis of the bending load in the legs is the similar to the analysis in Section 2.3.1. The dimensions of each leg is the same as the one in the B2 magnet (8x6x¼ wall ASTM A500 Grade B tubing) except that the length L = 17.75-inch and the slope is 8.5°. Each leg is loaded in axial compression by 2012.5-pounds. The equivalent bending moment on the leg is calculated from the reactionary force at the angled part of the leg (refer to Figure 11):

$R = 2012.5 \cos 8.5°$
$R = 1990$ lb.

$M = R*a$
$a = l*\sin 8.5°$
$l = L - L'$
where $L = 17.75$-inch
$L' = 4*\sin 8.5°$
$L' = 0.59$ in.
$l = 17.2$ in.
$a = 2.5$ in.
$M = 5048$ in-lb.

The stress in the leg is calculated:

$$\sigma = \frac{My}{I}$$

where $y = 4$ in.
$I = (bd^3 - b_i d_i^3)/12 = 62.6\ in^4$

$\sigma = 322.6$ psi.

The allowable bending stress (ASD Section F2.2) is $\sigma_{allow} = 0.60*\sigma_{yield} = 0.60(36000) =$ 21,600 psi. Thus, the bending that the leg experiences is well within the allowable.

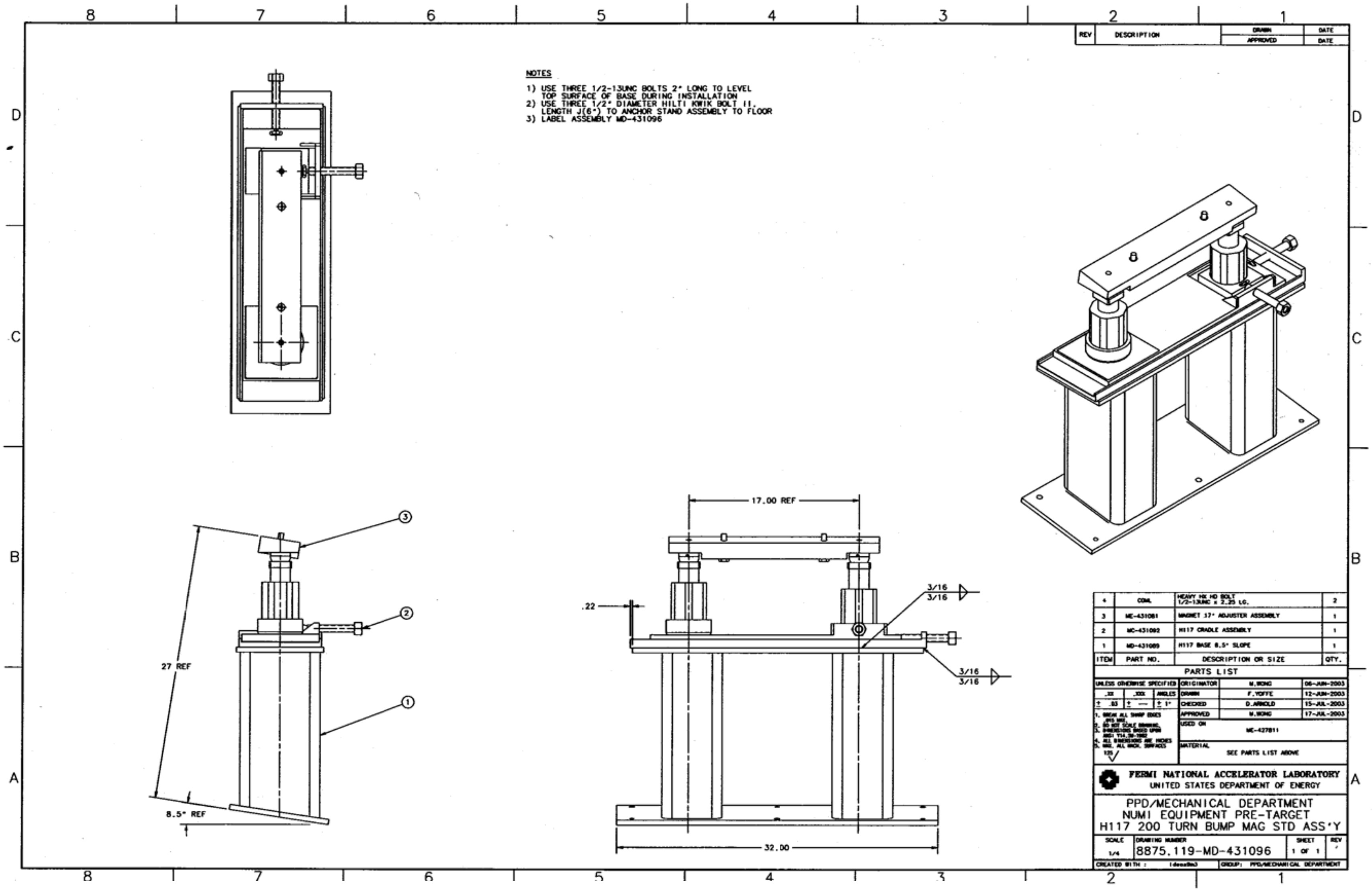


Figure 18 – Stand Assembly for 200 Turn Bump Magnet (Drawing 8875.119-MD0431096)

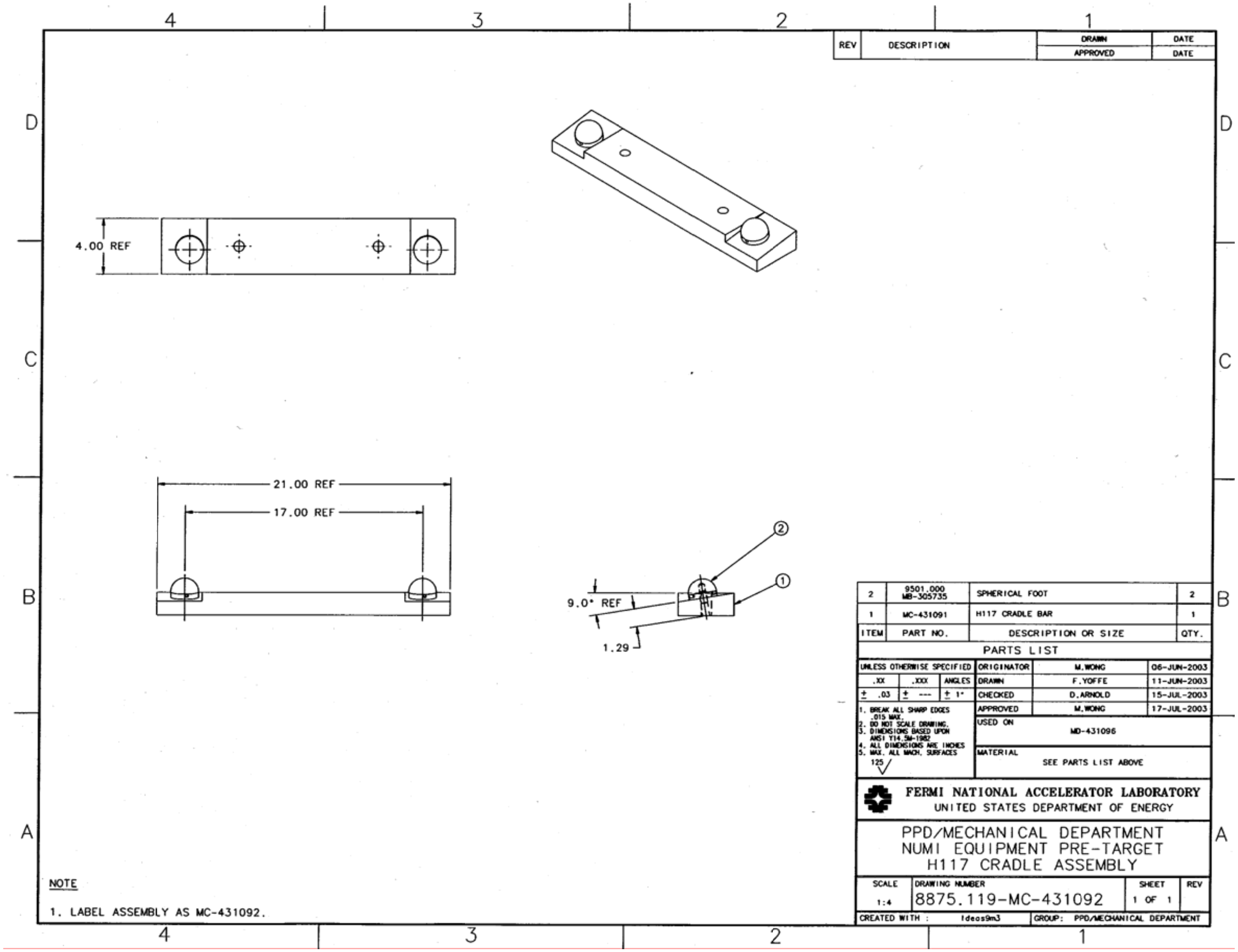


Figure 19 – Cradle Assembly for 200 Turn Bump Magnet Stand (Drawing 8875.119-MD-431092)

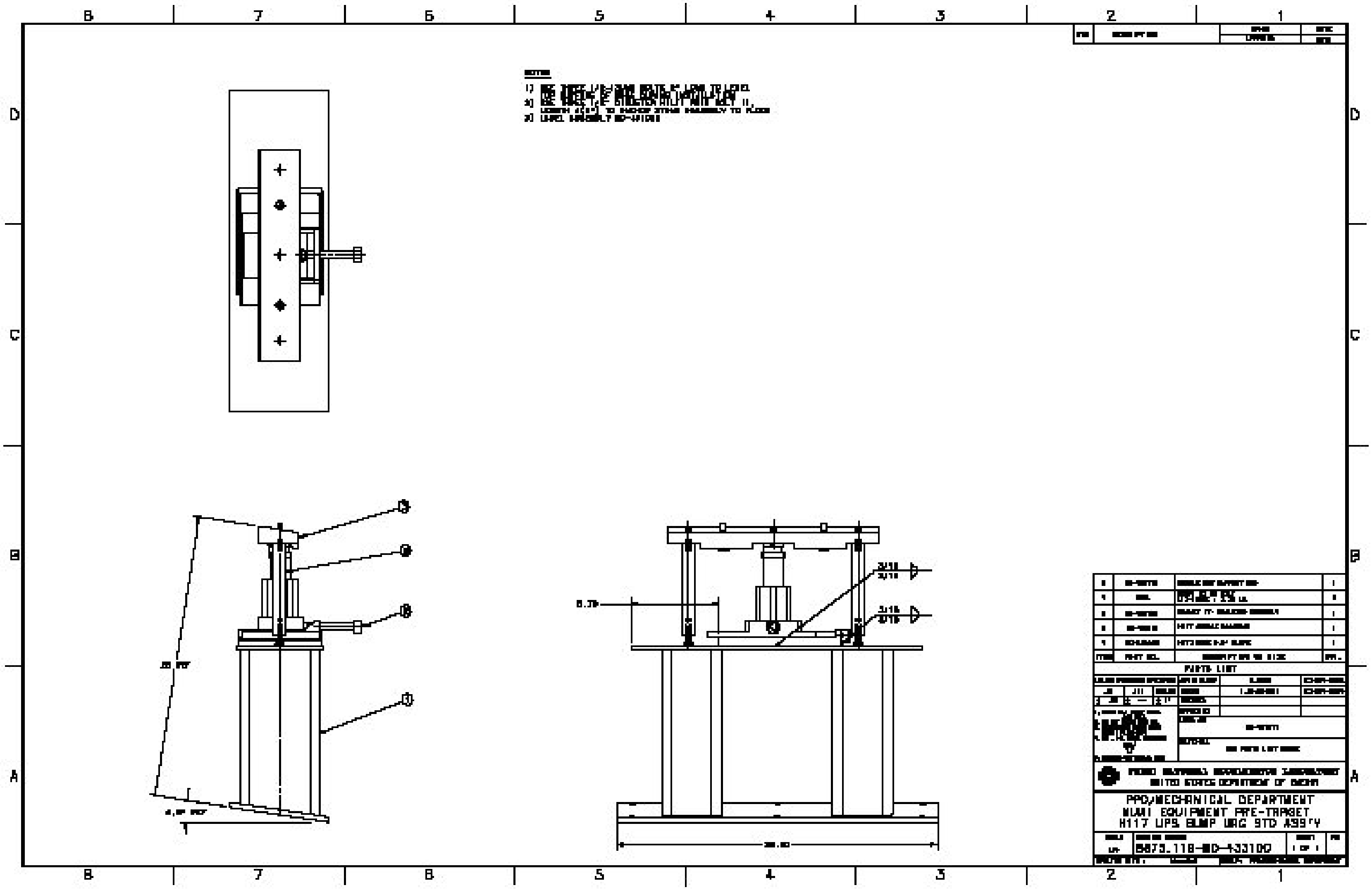


Figure 20 – Upstream Stand for H117 Magnet (Drawing 8875.119-MD-433100)

4 200 Turn Bump Magnet (SYTRIM) Stand

Figures 18 and 20 show drawings 8875.119-MD-431096 and –433100 of the magnet stand assemblies for the 200 turn bump magnet. MD-433100 supports the upstream end of the magnet while MD-431096 supports the downstream end. Each stand supports 1050-pounds. Assembly MD-431096 will use the same adjuster assembly as the one used by the quad magnet stand. Since the load on the adjuster of 1050-pounds is less than the load on the adjuster by the quads, it is assumed that the downstream adjuster can safely hold the 200 turn bump magnet. The base assembly is a similar design as the one used for the quad magnets. Due to the lower load by the 200 turn bump magnet, it is assumed that the base assembly design is within allowable.

4.1 Stability of magnet on three-point support

Let an external load F act on the upstream end of the magnet. Determine the load required to push the magnet off of the stand, summing the moments about stand labeled O (Figure 21).

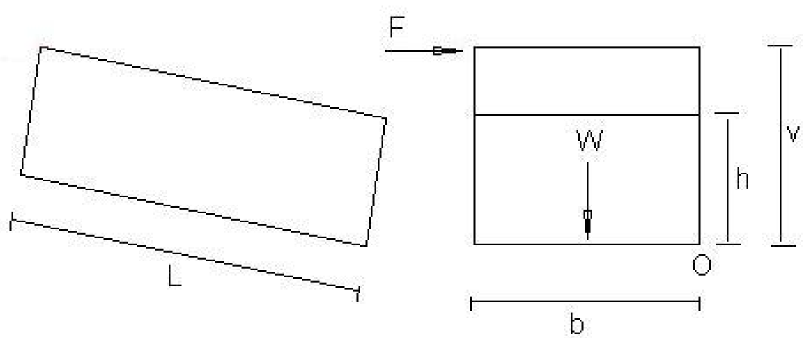


Figure 21 – Magnet Diagram

W*0.5b = F*v

Where W = 900 lb. (weight of the magnet)
b = 16 in.
v = 21.7 in.

F = 332 lb.

Figure 22 shows a diagram of the support triangle made up of the supports. Determine the force F' that is perpendicular to the line OP required to push the magnet off of the support R.

F'*h = W*d

Where h = 12.5 in. (height of magnet)
d = 3.9 in. (shortest distance from magnet's center of gravity to line OP)

F' = 140 lb.

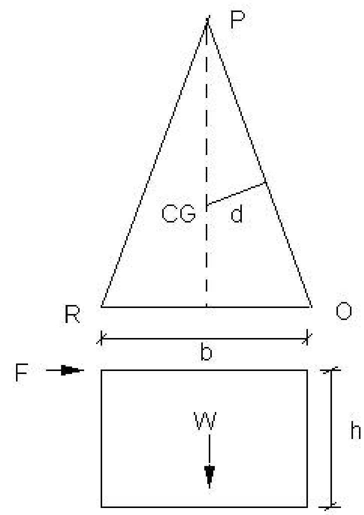


Figure 22 – Diagram of Forces and Moment Arms about Line OP

Two round steel bars, each of 1.25-inch diameter and 9.18-inch length, are placed at the upstream stand to keep the magnet from becoming destabilized due to an unexpected load of magnitude F' = 140-pounds.

4.2 Cradle assembly

Figure 19 shows drawing 8875.119-MD-431092 of a typical cradle assembly. The dimensions of the cradle bar are the same as the cradle bar used in the quad magnet assembly. With the 200 turn bump magnet cradle bar supporting 1050-pounds, it is assumed that its deflection and stresses are lower than the ones in the quad magnet stand and thus can safely support the magnet. The same assumption is also made for the bearing load on the spherical feet.

4.2.1 Shear load on the support bolt

On each cradle assembly, two bolts hold the end of the magnet in place. Each carbon steel bolt is ¾ - inch UNC, with a minor diameter d = 0.6273-inch (radius r = 0.31365-inch). For a magnet at a 9° angle, the total shear load at one cradle assembly is F = 164-pounds. The shear stress on one bolt is calculated based on one half of the shear load:

$\tau = F/A = (164/2)/(\pi * 0.31365^2) = 265$ psi

The allowable shear (ASD Section F4) is $\tau_{allow} = 0.40 * \sigma_{yield} = 0.40(36000) = 14{,}400$ psi. A support bolt in the cradle assembly falls well within the allowable shear.