Analysis of Chamber Support for E906

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Version	Changes	Date
1	Release for review	June 24, 2009
2	Response to review comments: Added calculation for additional loading case of lifting for floor. Additional loading case of lifting from horizontal position added. Minor corrections made.	October 30, 2009
	Modified Figures 2,4,15	
	Modified assumption 8 and added assumption 10.	
	Added sections 3,8, 9.2, Appendix I,J, K and all included figures and tables.	
	Corrected inconsistent material reference in section 7 (now A36)	
	Corrected allowable flexural stress in Appendix A	
	Change reference bolt from A325 to A307 in appendix C.	
	Modified appendix E to account for change in load location.	

1. Introduction

The support of an existing detector chamber has been redesigned using some previously existing components along with some new components. This calculation note describes the analysis performed on this support. The existing chamber is shown in the new support configuration in Figure 1 with a close-up on the new support assembly in Figure 2. The individual hardware components are identified in Figure 4.

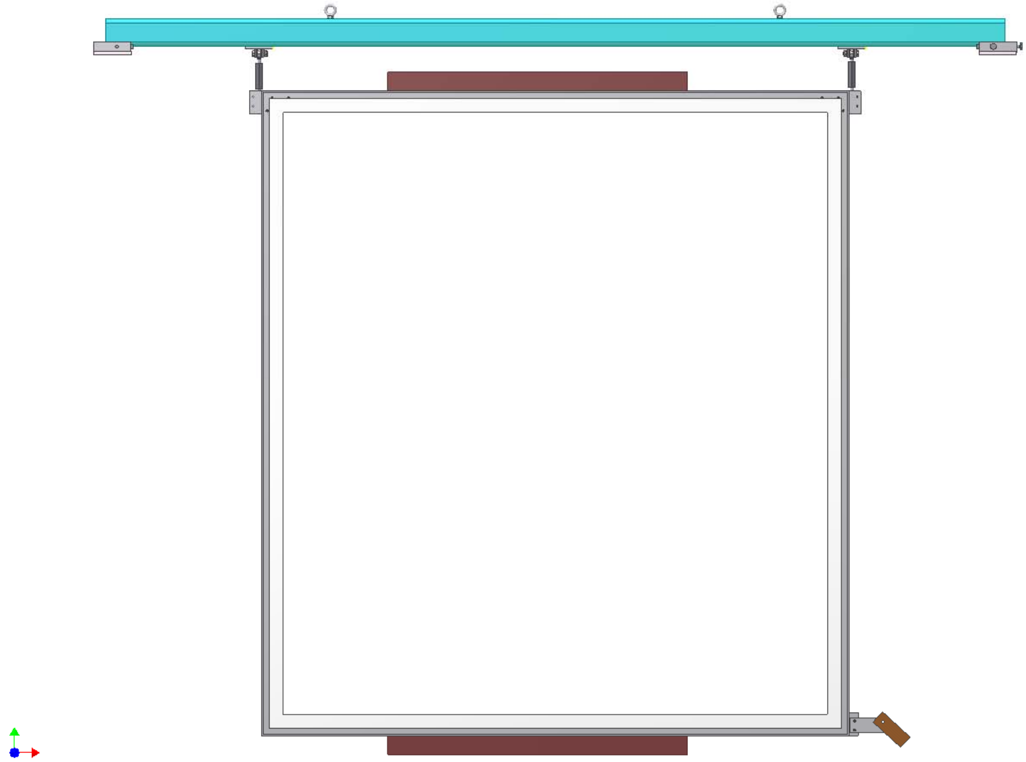

Figure 1 E906 Chamber and new support arrangement (Load case 1A).

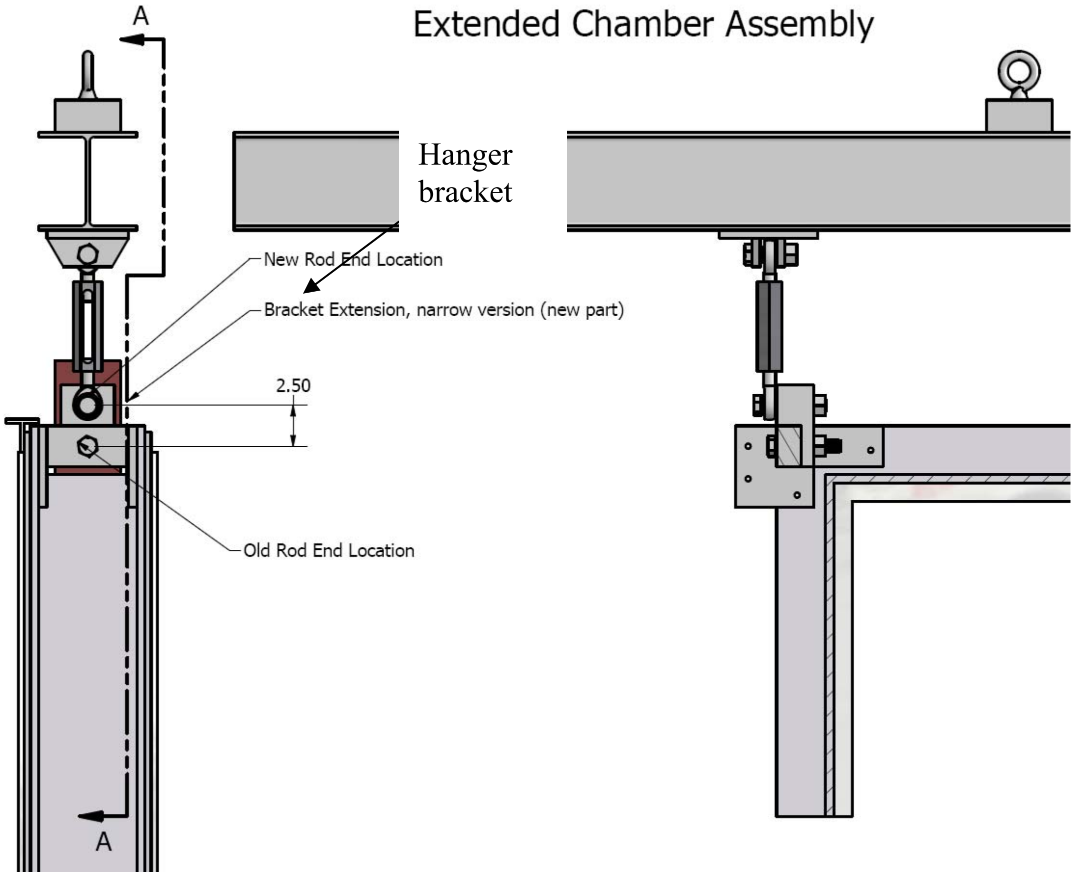

Figure 2 Close-up view of assembly mount in solid and wireframe to show hidden detail of connection (Load case 1).

As shown in the new configuration, the chamber orientation is rotated 90 degrees clockwise from the previous support configuration. The hanger brackets shown in Figure 2 are previously existing components attached to the chamber and are reused in this new configuration. The new configuration results in a different loading pattern on both the hanger bracket and the chamber. The assumptions of the analysis are listed below.

Assumptions:

1. Rod ends will be threaded at least ½ inch into adjuster nut.
2. Chamber load is applied through the cg. such that end supports equally share ½ the load.
3. No side loading, or dynamic loading is considered. Load cycles are expected to be very few such that fatigue is not considered.
4. Chamber weight is 2000 lbf.
5. Existing hanger bracket welds are at least 3/16 inch fillet welds have been properly welded using weld material with min. properties of AWS 7018 and with procedures such that full strength can be assumed as if made by AWS qualified procedures. Bracket material is assumed to be A36 steel. While it is known that

these brackets have supported the chambers previously, the new loading configuration increases the stresses on these welds. (show figure)

6. Rigging slings connected to the rigging eyebolts will make angle equal to or greater than 45 degrees between the sling and the longitudinal direction of the beam.
7. No analysis is done on the chamber to understand the effect of the new loading configurations on the chamber as compared to the previous configuration.
8. Connecting bolts are assumed to have a minimum strength of ASTM A307.
9. Newly fabricated components are assumed to be welded following AWS D1.1 structural steel welding code.
10. The chamber is lifted smoothly with no jerking or side to side swinging motions. The chamber edge is capable of supporting rotation and the lift is controlled at all times (i.e. chamber does not rotate over as c.g. crosses rotation point on way to vertical).

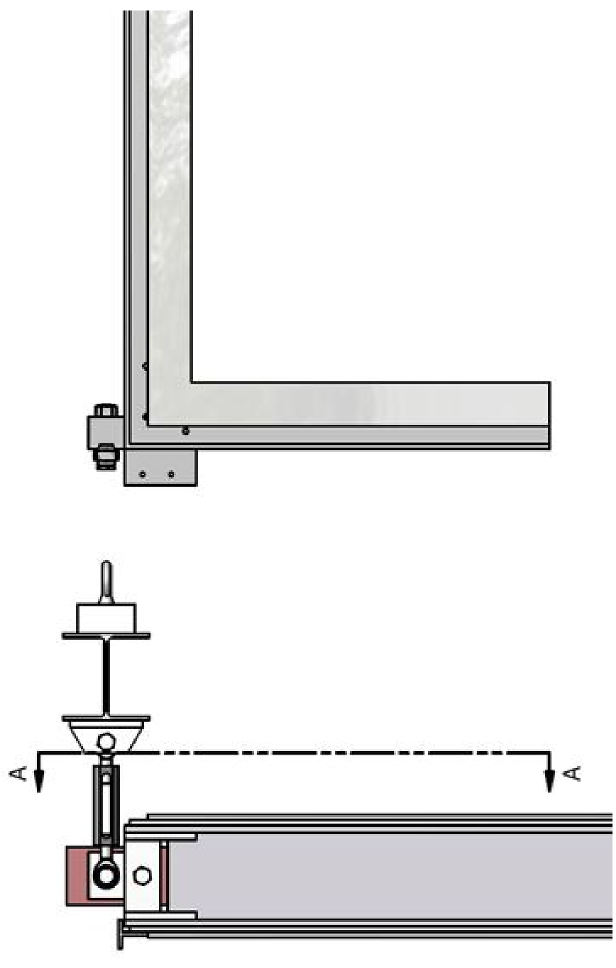

Figure 3 Mounting hardware orientation during lift of chamber from a horizontal position on the floor (Load case 2).

2. Load Cases

There are two main load cases to be considered. Load case 1 is with the chamber being supported in the vertical orientation as shown in figures Figure 1 and Figure 2. Load case 1 additionally has two alternate configurations of supporting the support beam. Load case 1A is with the beam supported simply at the ends. Load cases 1B considers the case where rigging straps are connected to eyebolts in the top of the beam.

Load cases 2 considers the loading on the support hardware during a lift of the chamber from the floor. The orientation of the hardware is shown in Figure 3 where the chamber is shown lying on a horizontal surface. During the lift, it is assumed that the chamber rotates about the edge opposite the lift point until the chamber is vertical. It is assumed that the cg is at the center of the chamber and therefore that the reaction force at the supports is along the line of the gravity vector and is equal to ½ that of load case 1.

In comparing the load cases, the direction and type of loading (i.e. axial) is the same for all of the components starting with the rod ends and looking above to the beam (see Figure 4). The magnitude of the for load case 1 is twice that for load case 2 and so only the loads for load case 1 are considered for those pieces. It is assumed that rigging straps are used for load case 2. However since the loading is 50% that of load case 1 and so no separate analysis of the beam is needed for load case 2.

Load Case	**Sub-Case for Beam**	**Description**
1	A	Chamber supported vertically in A-frame
	B	Chamber supported vertically using rigging straps
2	*n.a.*	Chamber being rotated 90 degrees from horizontal surface

Table 1 Load cases considered

3. Free Body Diagram

The free body diagram of the components in the load path are shown in Figure 4. The diagram shows load case 1, the beam supported simply at the ends which follows the configuration of the chambers supported from the KTEV A-frame. The alternate configuration of the beam during rigging, load case 1B, and the modification to the free body diagram for that configuration is shown in Figure 7. The simply supported configuration creates the larger bending moments in the beam. As can be seen from Figure 4, it is assumed that each of the two chamber support assemblies support ½ the chamber weight.

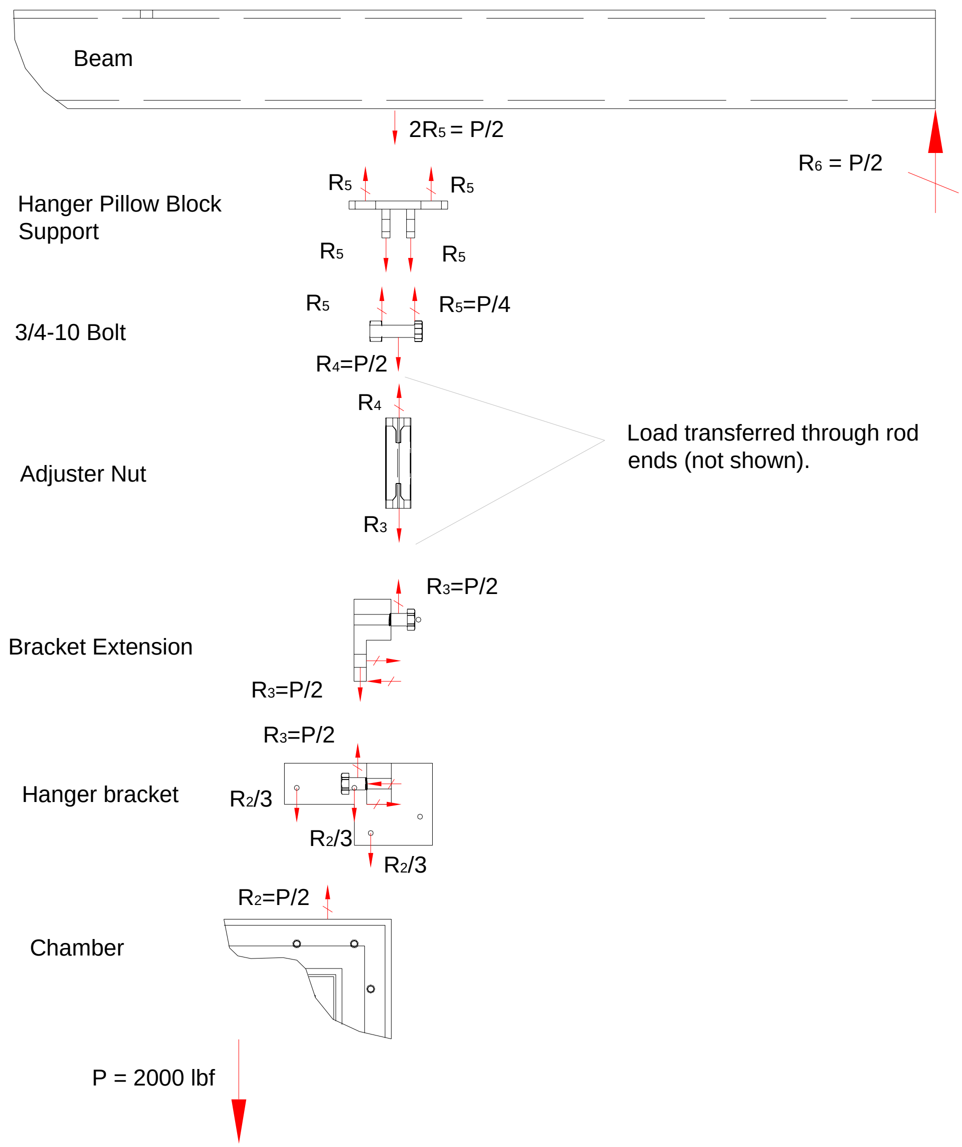

Figure 4 Free body diagram for Chamber support with beam in simply supported configuration while supported from A-frame stand (not shown) (Load Case 1A).

4. Support Beam

4.1. Load case 1A - Simply supported configuration

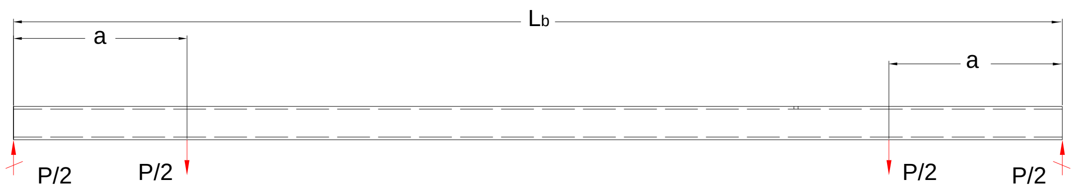

Figure 5 Loading diagram for simply supported configuration (Load case 1A)

The support beam is a W6 x 25 wide flange beam. The sizing check on this beam is done in Appendix A with the assumption that the beam is A36 steel. The actual beam stresses are found to be much less than the allowable and there is no need for stiffeners at any points along the beam. Figure 5 shows the loading for the simply supported configuration while the chamber is supported on the A-frame stand. The analysis of the beam in the simply supported configuration is given in Appendix A.

4.2. Load Case 1B - Rigging configuration

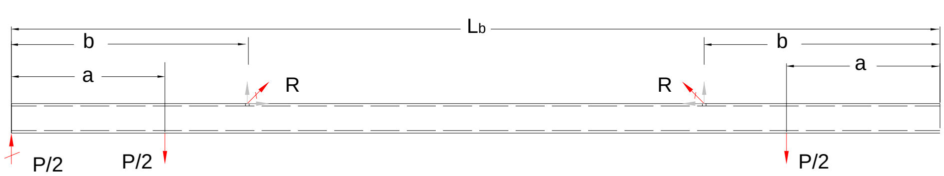

Figure 6 Loading diagram of beam during rigging (load case 1B)

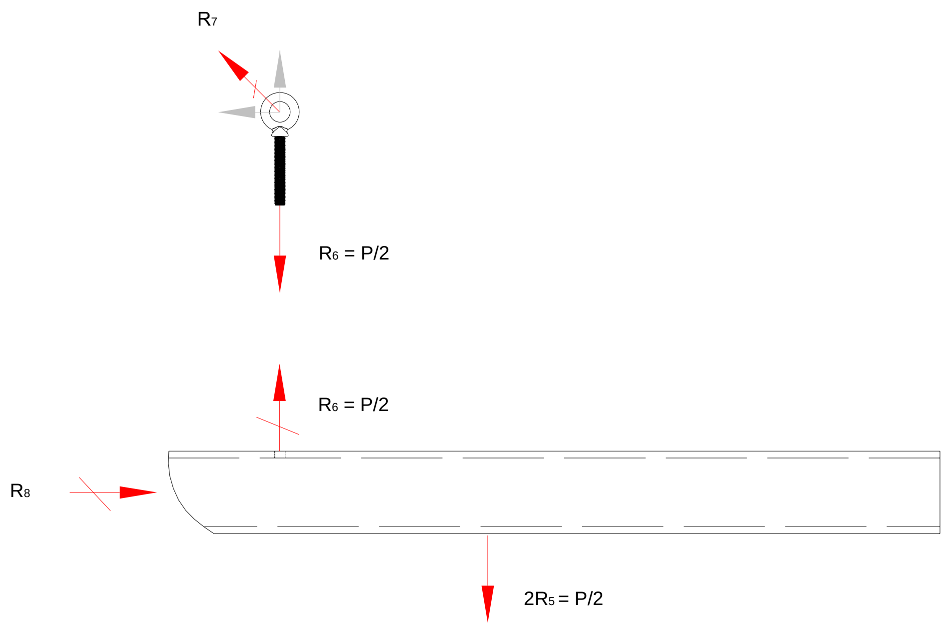

Figure 7 Free body diagram of beam during rigging.

Figure 6 and Figure 7 show the rigging configuration where eyehooks are placed at the distance *b=48* inches from the edges. The assumption for rigging is that the slings make an angle equal to or greater than 45 degrees with respect to the longitudinal axis of the beam. This would result in equal horizontal and vertical components of the reaction force of (P/2)/cos(45deg)= 1414 lbf. Appendix H checks the combined axial stress and bending that result due to the horizontal component of the rigging forces.

The eyebolt selected is a 5/8 inch lifting eye from the data sheet for these components is given in Appendix G. This lifting eye is designed for 5000lbf vertical lift and has a design limit of 1560 lbf when lifting is directed at 45 degrees. This is greater than the 1000lbf vertical load that occurs during rigging. Appendix H also contains a check on the internal thread shear of an eyebolt with 5/8" threads.

The rigging eyebolt will be threaded into a 4 inch square block that is 2inches thick to allow sufficient clearance for threads. Per the ASD code, the min. allowable fillet weld with the thicker piece equal to 2 inches is 5/16" which yields a throat of 0.22 inches. Assuming, four (4) 1 inch fillet welds located 2 inches apart along to edges of the block would give ~ 1in area. The resulting shear stress would be ~ 1000psi in the weld which is well below the allowable stresses.

5. Pillow Block Support

The pillow block support calculation is given in appendix B. The bosses will be welded using 3/16 inch fillet welds as this is the minimum fillet weld size allowed by code due to the support member thicknesses. It is recommended to provide at least 2inches weld length on each side of each weld boss (figure). This amount of weld is reasonable for fabrication and still results in very low weld stresses compared to allowable. It is assumed that welds follow AWS D1.1 requirements and procedures. Normal stresses in the bosses are very low. A check for prying reveals that the pillow block flange is sufficiently sized.

The force is transferred via a pinned connection using a ¾-10 bolt. Appendix C shows that the stresses in the bolt are much lower than the allowable and that edge distance for the hole in the pillow block support bosses is acceptable.

The pillow block support is connected to the beam using four (4) 3/8-16 bolts which would each see a load of 250lbf.

6. Rod End

Looking at Figure 4, the force transmission between each side of the adjusting block and the next member is done through a commercially purchased rod end rated at 22,800 lbs static load. The data sheet is in appendix F.

7. Adjuster Nut

The adjuster nut carries a tensile load of P/2=1000lbf in static loading as shown in Figure 4. An additional function of the adjuster nut is to provide +/-1 inch of vertical adjustment to the chamber. To prevent accidental release of the rod ends during adjustment, the adjuster nut has a slot machined through it providing a visual confirmation of thread engagement. Due to constraints on the overall size of the adjuster nut, the length of internal thread inside the adjuster is analyzed with respect to the actual design load and not following common design convention to size the internal thread such that the connecting bolt fails prior to the internal threads. That is, as designed, the internal threads will fail before the rod end thread section. This is design is considered acceptable since due to the function of the adjustment rod, the threaded ends of the rod ends will always be loosely fit into the adjuster nut without pretension. The loads should then only be that of the ½ chamber weight as shown in Figure 4.

The slot in the adjuster nut (Figure 8) is such that the slot edge starts ½ inch away from the edge giving a ½ inch thread length of complete thread. The calculation in appendix D show very low stresses with the material assumed to be A36 or similar low carbon steel.

It is necessary to verify that the material removed from the inspection slot does not weaken the adjuster nut. In addition to verifying the net section area is sufficient to handle the tensile loads, there is also an applied torque when the adjusting action is raising the load. The torque required to raise the load is conservatively calculated to be 344 in-lbf using the worst case value for steel on steel friction of μ=0.8 as shown in appendix D. This torque along with the axial load was applied to an FEA analysis, the results of which are shown in Figure 9 and Figure 10. The stresses are well below the 35 ksi yield stress of A36 with only highly localized peak stresses at 6000 psi.

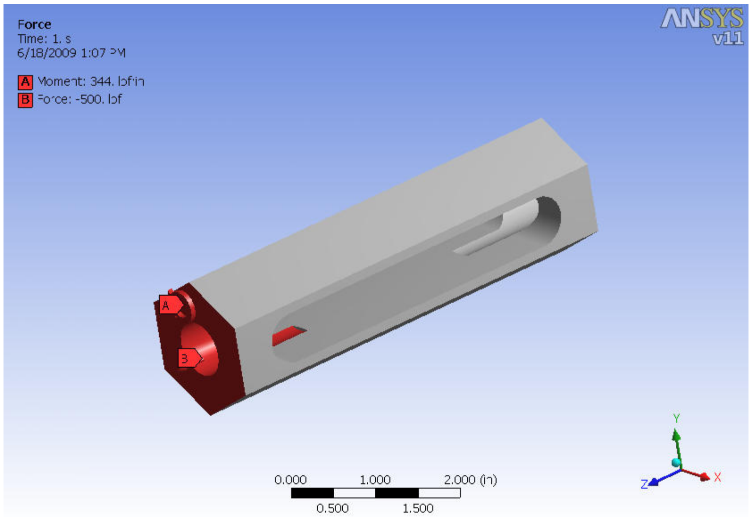

Figure 8 Adjuster nut and applied loads. Label A shows the face of the applied torque and label B shows the face of the applied tensile loads (thread geometry not modeled). The face opposite label A is held fixed.

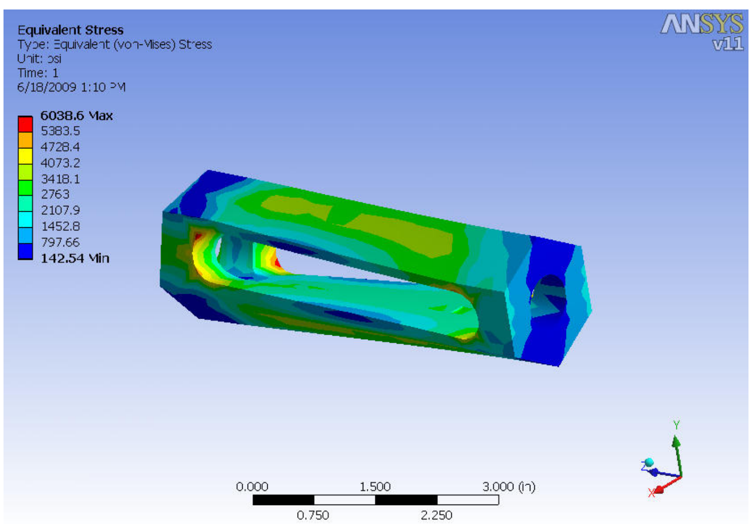

Figure 9 Von Mises stress on Adjuster nut with tensile load and raising torque applied to one end while restraining oppositre end.

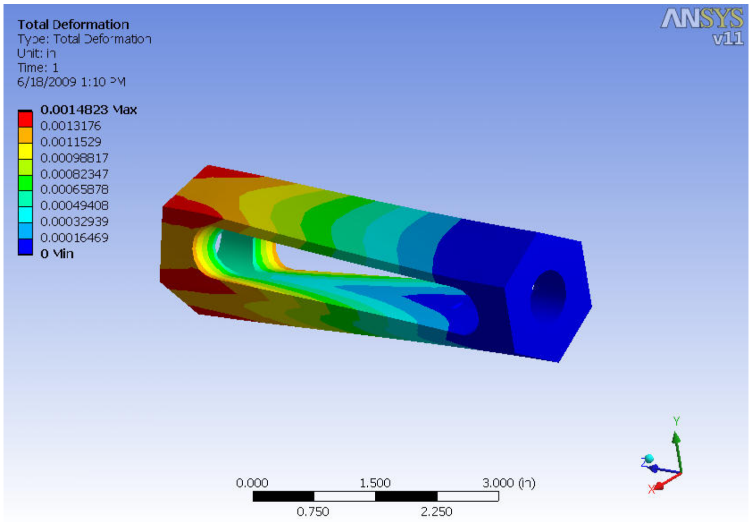

Figure 10 Deformation of Adjuster nut with tensile load and raising torque applied to one end while restraining opposite end.

8. Bracket Extension

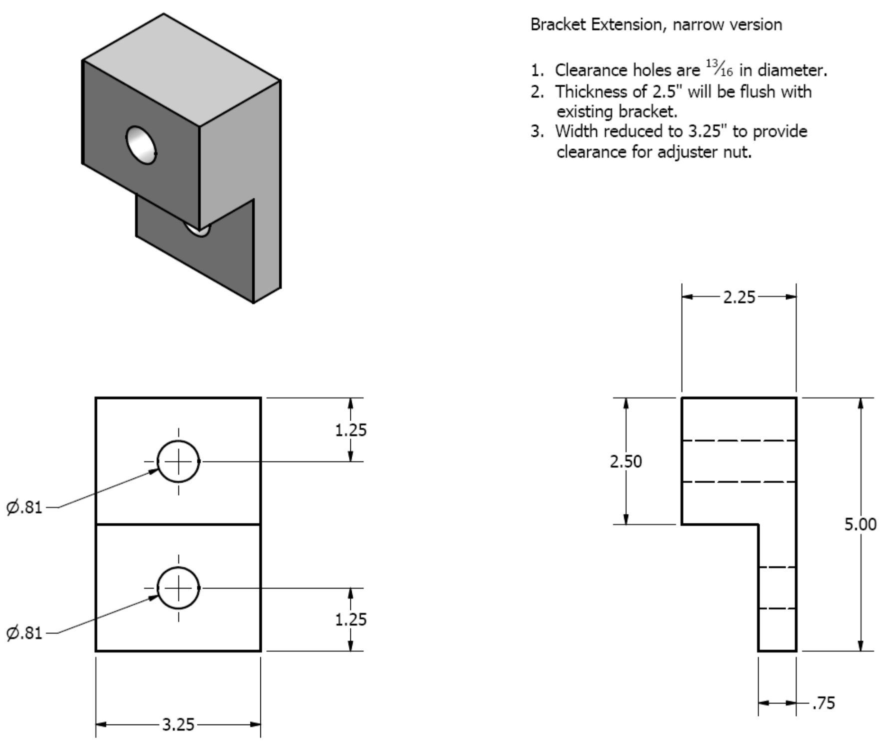

Figure 11 Bracket Extension

8.1. Load case 1

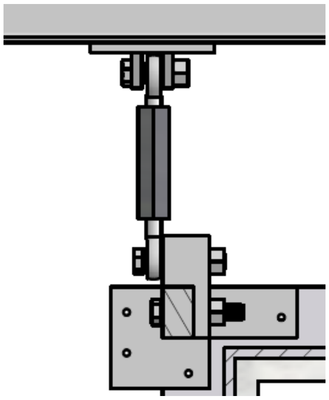

Figure 12 Bracket extension in Load case 1 showing vertical loading.

The bracket extension shown in Figure 11 is assumed to be made of A36 or steel. The detailed calculation can be found in appendix I. In load case 1, the force is transmitted vertically through the bracket as is shown in Figure 12. The force is transferred through 3/4in bolts at both the top and the bottom. A rod end connects to the top bolt in the bracket and creates a remote force to the bracket that is not inline with the reaction force going through the lower hole. This creates a moment in the bracket. This moment creates a combined axial and shear load in the lower bolt. The bolt forces are found to be about 50% of the acceptable load of an A307 bolt. The lower section is checked for block shear and has a strength significantly higher than needed. The bracket is checked also for bending stresses. The highest bending stresses are found in the section at the lower bolt hole. The bending stress if about 50% the allowable for A36 steel.

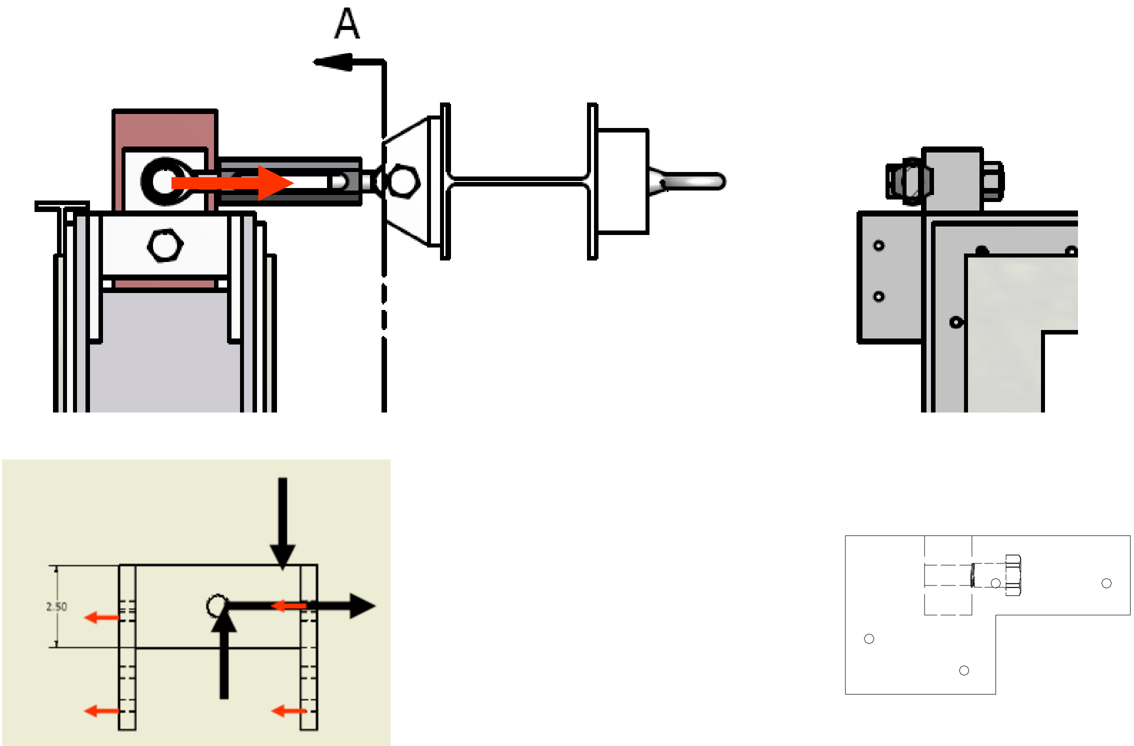

Figure 13 Bracket extension in Load case 2 showing lifting position.

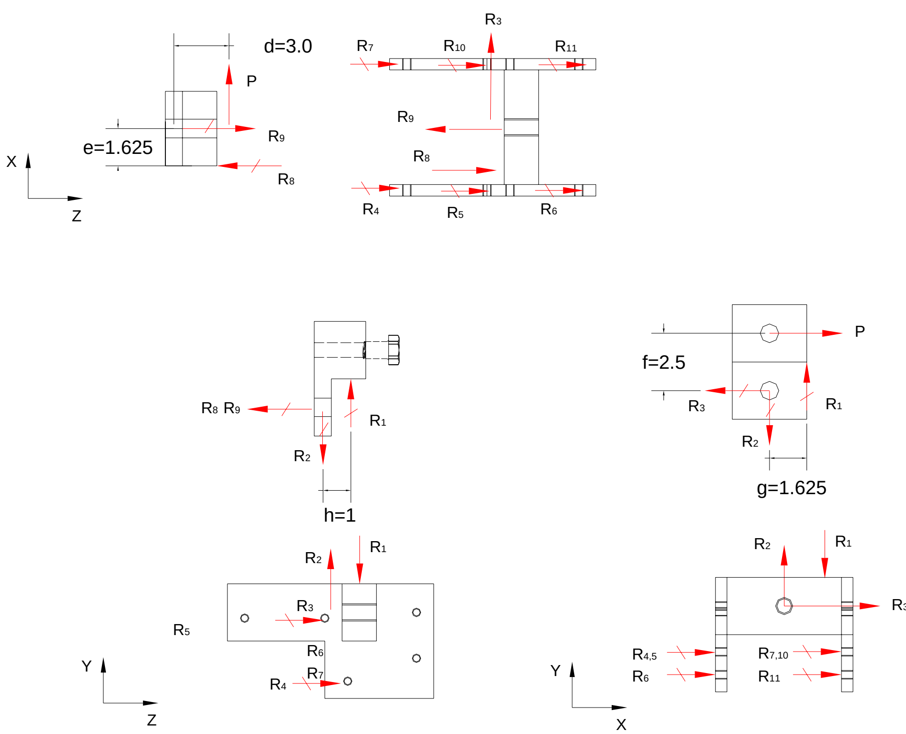

Figure 14 Free body diagram showing interaction of extension bracket and hanger bracket in load case 2

8.2. Load case 2

The bracket extension is shown in Figure 13 for load case 2 with the free body diagram shown in Figure 14. As can be seen in the figure, it is shown in the position of maximum side loading. The single bolt connection does not restrain the rotation caused by the side load P and so there is a reaction supplied by the top boss of te extension bracket in contact with the hanger bracket. This is depicted in Figure 14. The detailed calculation is shown in appendix J for load case 2. The bolt located in the lower part of the bracket is the worse loaded bolt and is loaded in combined axial tension and shear. No friction is assumed in this connection. The axial tensile stress is 2 ksi compared to the allowable 12ksi (for A307 bolt) when shear effects are included. The total reaction force through the hole is 900 lbf which is lower than that of load case 1 and so block shear does not need to be checked. The couple created by R_1, R2 creates bending about the x-axis. The maximum stress (ingnoring stress concentrations) at the section with the hole is 1.1ksi much less than the allowable 23ksi. The load P also causes torsion about section A (see figure in appendix J. The maximum shear stress for this is 3ksi which is accepatble.

9. Hanger Bracket

9.1. Load case 1

The hanger bracket connects directly to the chamber as shown in Figure 2 and Figure 4. Figure 15 also shows more detail of this hanger including the eccentric load path. The hanger is connected to the chamber by 3/8-16 bolts where the load transfer is assumed to be evenly shared by each bolt such that the load, P/12 is applied vertically at each bolt location. As Figure 4 and Figure 15 show, the reaction is applied eccentrically through a ¾-10 bolt. The stress on the 3/8 inch bolts as well as the fillet weld connection are determined using a standard elastic method. The bolt stresses and weld stresses are both found to be acceptable as shown in appendix E.

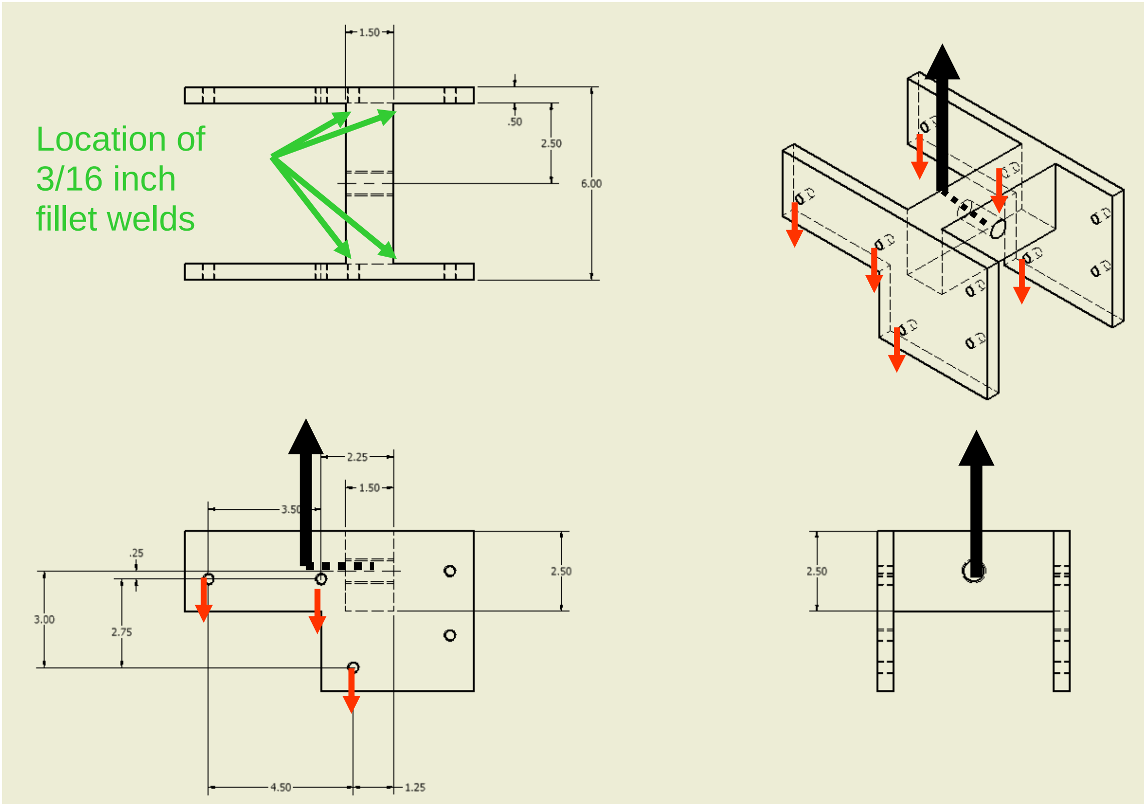

Figure 15 Hanger bracket dimensions and forces. Black arrow depicts eccentric reaction loading through 3/4-10 bolt. Red arrows loading through 3/8-16 bolts.

9.2. Load case 2

The free body diagram for the hanger bracket for load case 2 is shown in Figure 16. The loading depicts the situation where the chamber is horizontal and is the worst side loading case. Under the assumption that the chamber cg is in the geometric center of the chamber, the lifting load at each support is ½ the applied load of load case 1. The loading force is transferred from the extension bracket through a ¾-10 bolt in single shear. The load is assumed to act through the center of the extension bracket and is offset from the connecting face.

The free body diagram shows three dimensional forces and reactions on the bracket due to the side loading of load case 2. The vertical load component as well as the vector sum of x- and Y- direction loads is less than that of load case 1 . For this reason, the analysis of the weld stress due to the shear force and moment due to the offset load are lower than that of load case 1 and need not be repeated. A check on the bolt forces was done by creating an FEA model. The loading and the resultant forces are shown in appendix J. The loading assumed fixed displacement at the chamber connection. No analysis of the chamber has been made to validate this claim. Two bolts were examined, the bolt with the highest shear and the bolt with the highest axial load. The bolts were eveluated by determined the reduced axial allowable stress after shear effects were takin into account. The stress was compared to low strength A307 bolts since the existing bolts are not known. The stresses are found to be acceptable but the margin in the case of the highest shear loaded bolt is small with the actual axial stress of 3.7 ksi compared to the allowable of 5.1 ksi. This margin nay be greater if grade 5 or A325 bolts are used.

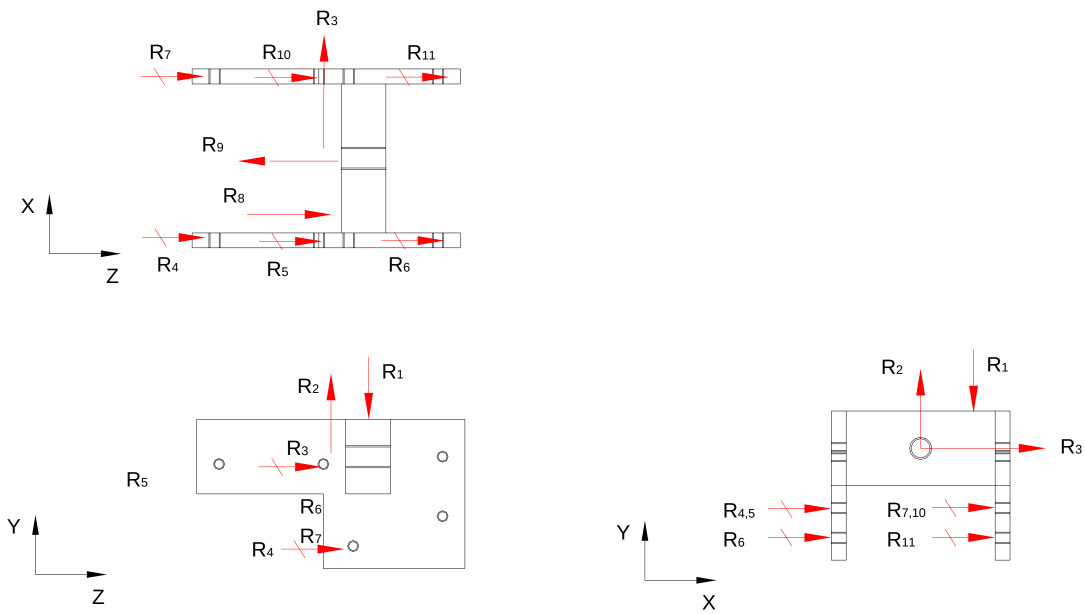

Figure 16 Free body diagram of hanger bracket in load case 2

10. Conclusion

The analysis described above concludes that the new support design is adequate for the design loads assumed with often generous safety factors. The analysis of the existing hanger bracket is predicated on assumptions of material and welding. The actual material used must be verified and checked with this assumption. Additionally, any possible information available regarding the original weld specification should be sought and compared to the assumptions contained within this analysis. Short of this verification, the parts should be inspected and the compared with assumptions based on best available engineering judgment. Short of better supporting information, the hanger brackets should be load tested at 110% of the design load prior to service.

Appendix A-Support Beam

Properties for a W6x25

$A := 7.34in^2$	Section area
$t_f := 0.455in$	Flange thickness
$b_f := 6.08in$	Flange width
$d := 6.375in$	Overal depth of beam
$t_w := 0.320in$	web thickness
$I_{xx} := 53.4in^4$	Moment of inertia about XX axis
$S_{xx} := 16.7in^3$	Section Modulus about XX axis
$r := 2.7in$	Radius of gyration about XX axis
$r_T := 1.66in$	
$h := d - 2 \cdot t_f = 5.465\,in$	Clear distance between flanges

Assume A36 steel

$F_y := 36 \cdot ksi$

$F_u := 58ksi$

$E := 30 \cdot 10^6 psi$

Check that no deduction is necessary for bolt holes

$A_{fg} := b_f \cdot t_f = 2.766\,in^2$	Gross flange area
$d_b := 0.5in$	Bolt clearance hole diameter

$A_{fn} := A_{fg} - 2 \cdot d_b \cdot t_f = 2.311\,in^2$ Net flange area

$0.5 \cdot F_u \cdot A_{fn} = 6.703 \times 10^4\,lbf$ must be greater than $0.6F_y \cdot A_{fg} = 5.975 \times 10^4\,lbf$

ASD B10-1

Check for compact section: width thickness ration must be less that limiting value.

$\frac{b_f}{2 \cdot t_f} = 6.681$ Width to thickness ratio of compression flange on W shape

$\frac{65}{\sqrt{\frac{F_y}{ksi}}} = 10.833$ Limiting width to thickness ratio for compact section, Table ASD B5.1

Section is compact

determine Lc (in inches) from F1-2

$L_c := \frac{76 \cdot \frac{b_f}{in}}{\sqrt{\frac{F_y}{ksi}}} = 77.013$ $\frac{20000}{\left(\frac{d}{\frac{A_{fg}}{in}}\right) \cdot \frac{F_y}{ksi}} = 241.081$

$L_b := 189in = 15.75 \cdot ft$

$L_b > L_c$

Allowable bending stress, F1-1

$C_b := 1$ Conservative assumption

$r_T = 1.66\,in$ $l := L_b = 189\,in$

since $\frac{l}{r_T} = 113.855$ < $\sqrt{\frac{510 \cdot 10^3 \cdot C_b}{\frac{F_y}{ksi}}} = 119.024$

use F1-6

$$F_b := \left[\frac{2}{3} - \frac{\frac{F_y}{ksi}\cdot\left(\frac{l}{r_T}\right)^2}{1530\cdot 10^3\cdot C_b}\right]\cdot F_y = 1.302\times 10^4 psi$$

$$\rho_{beam} := 25\frac{lbf}{ft}$$

$$W_{beam} := \rho_{beam}\cdot L_b = 393.75\,lbf$$

$$a := 31.26in$$

Mass of beam is 20% of load. Nonetheless, we neglect it for simplicity at the moment under the assumption that overall stresses are very low for this beam.

$$P := 2000lbf$$

$$V_{max} := \frac{P}{2} = 1\times 10^3 lbf$$

Maximum shear at x<a, x>b

$$M_{max} := \frac{P\cdot a}{2} = 3.126\times 10^4\cdot in\cdot lbf$$

Maximum moment constant at a<x<b

$$f_b := \frac{M_{max}}{S_{xx}} = 1.872\cdot ksi$$

Bending stress is acceptable, is there a deflection requirement?

Check shear stress

if $\frac{h}{t_w} = 17.078$ is less than $\frac{380}{\sqrt{\frac{F_y}{ksi}}} = 63.333$

$$F_v := 0.4\cdot F_y = 14.4\cdot ksi$$

Allowable shear stress , F4-1

$$f_v := \frac{\frac{P}{2}}{A} = 0.136\cdot ksi$$

Average shear stress at location of maximum shear load.
Value is less than allowable.

Note that bending and shear stresses are very low and adding the beam weight would not raise these significantly.

Transevers stiffeners are not needed per section F5 as h/t < 260 and fv < Fv.

Appendix B - Pillow block support

Weld connection of boss

$l_w := 2in$	Length of weld
$t_{wn} := \frac{3}{16} in$	Nominal weld size, note this is minimum weld allowed by code
$t_w := \frac{t_{wn}}{\sqrt{2}} = 0.133\,in$	Fillet weld throat
$n_w := 2$	Number of weld paths per boss
$A_w := n_w \cdot t_w \cdot l_w = 0.53\,in^2$	Effective shear area of fillet weld
$P_b := \frac{P}{4} = 500\,lbf$	Load applied to single boss
$f_{vw} := \frac{P_b}{A_w} = 942.809\,psi$	Shear stress in boss weld

Normal Stress in boss taken through section with clearance hole

$l_b := 5in$

$t_b := 0.5in$

$d_{bb} := 0.5in$

$A_b := (l_b - d_{bb}) \cdot t_b = 2.25\,in^2$

$$f_{nb} := \frac{P_b}{A_b} = 222.222\,psi$$

Bolt connection of piilow block to beam

Assume 3/8-16 bolts

Check for prying using method 1 of ASD section on hanger type connections

Find minimum t for pillow block support flange such that prying force is insignificant.

$T := 250lbf$ applied tension per bolt

$b := 3in$ distance from bolt centerline to center of bracket (conservativ value)

$p := 5in$ length of flange

$F_y = 3.6 \times 10^4\,psi$ yield strength of material

$t_{reqd} := \sqrt{\frac{8T \cdot b}{p \cdot F_y}} = 0.183\,in$ required thickness of pillow block flange

actual thickness is 1/2 inch which is greater than minimum required.

Appendix C - Pinned connection to boss (pinned with 3/4-10 bolt)

$F_{vblt} := 12ksi$

Allowable shear stress when threads are in the load plane for an A307 bolt in single shear. Table 7-1 13 th Ed.

$d_{blt} := 0.75in$

bolt diameter

$A_{blt} := 0.4418in^2$

Area of a 3/4-10 bolt

$$f_{vblt} := \frac{\frac{P}{4}}{A_{blt}} = 1.132 \cdot ksi$$

Actual shear stress is less than allowable shear stress.

$$l_v := \frac{2\frac{P}{4}}{F_u \cdot t_b} = 0.034\,in$$

Minimum edeg distance in boss from clearance hole for 3/4-10 bolt connection, J3-6.

$L_e := 0.75in$

$$F_{pblt} := \frac{L_e \cdot F_u}{2 \cdot d_{blt}} = 29 \cdot ksi$$

Allowable bearing stress at boss hole, J3-3

Appendix D- Adjuster Nut

$S_y := 36ksi$ Material properties for A36

$S_u := 58ksi$ Low estimate of UTS for A36

3/4-10 threaded rod

$D_m := 0.7353in$ min. diameter of class 2A thread, 3/4-10

$D_p := 0.6773in$ pitch diameter

$p := \frac{1}{10}in$ Thread pitch

$A_s := \frac{\pi}{4} \cdot D_p^{\,2} = 0.36\,in^2$ Tensile shear area

$$\sigma_a := \frac{\frac{P}{2}}{A_s} = 2.776 \times 10^3 psi$$

$n := 3$ desired number of threads to sustain load

$L_t := p \cdot n = 0.3\,in$ min. length of thread engagement

$A_t := \pi \cdot D_p \cdot L_t = 0.638\,in^2$ Shear area on internal treads

$$\tau := \frac{\frac{P}{2}}{0.87 \cdot A_t} = 1.801 \times 10^3 psi$$ Internal thread shear stress

$$\frac{S_y}{\sqrt{3}} = 2.078 \times 10^4 psi$$ Allowable shear stress based on yield

factor of safety on thread yield

$$\frac{S_y}{\sqrt{3} \cdot \tau} = 11.543$$

$L := 0.1in$	Screw lead
$\alpha := 30°$	1/2 thread angle
$\mu := 0.8$	Worst case friction, steel on steel, unlubicrated per Machinery Handbook

Torque required to raise the load assuming no collar friction

$$T := \frac{P}{2} \cdot \frac{D_p}{2} \cdot \left(\frac{\cos(\alpha) \cdot L + \mu \cdot \pi \cdot D_p}{\pi \cdot D_p \cdot \cos(\alpha) - \mu \cdot L} \right) = 343.667 \cdot in \cdot lbf$$

Appendix E- Hanger bracket

Fastener data

$A_{bm} := 0.0678 in^2$ Minor area for a 3/8-16 UNC fastener

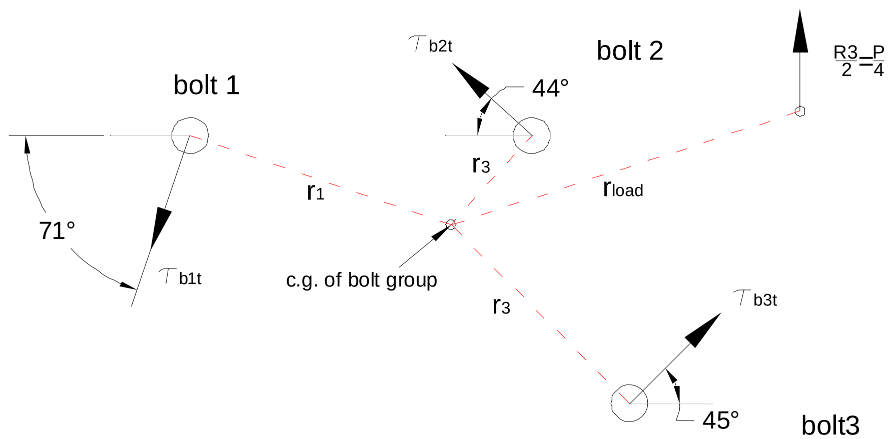

Figure E1. Layout of bolt group for hanger bracket and identification of variables used in determination of torsional componet of bolt stress.

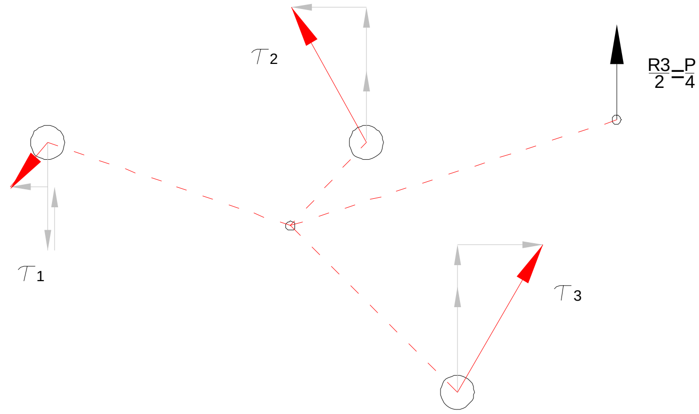

Figure E2 Resultant shear stress for bolt group after adding torsional and vertical components.

Determine cg location of fastener group

Fastener and applied load x,y locations

$x_1 := 0in$ $x_2 := 3.5in$ $x_3 := 4.5in$ $x_{load} := 0.25in$

$y_1 := 0in$ $y_2 := 0in$ $y_3 := -2.75in$ $y_{load} := 0.25in$

$$x_{bcg} := \frac{x_1 + x_2 + x_3}{3} = 2.667\,in$$

location of bolt group cg

$$y_{bcg} := \frac{y_1 + y_2 + y_3}{3} = -0.917\,in$$

distance from fastener group cg to individual fasteners

$$r_1 := \sqrt{\left(x_{bcg} - x_1\right)^2 + \left(y_{bcg} - y_1\right)^2} = 2.82\,in$$

$$r_2 := \sqrt{\left(x_{bcg} - x_2\right)^2 + \left(y_{bcg} - y_2\right)^2} = 1.239\,in$$

$$r_3 := \sqrt{\left(x_{bcg} - x_3\right)^2 + \left(y_{bcg} - y_3\right)^2} = 2.593\,in$$

$$r_{load} := \sqrt{\left(x_{bcg} - x_{load}\right)^2 + \left(y_{bcg} - y_{load}\right)^2} = 2.684\,in$$

Calculate polar moment of inertia of bolt group using parallel axis theorem

$$J_{bg} := \left(r_1\right)^2 \cdot A_{bm} + \left(r_2\right)^2 \cdot A_{bm} + \left(r_3\right)^2 \cdot A_{bm} = 1.099\,in^4$$

$$T_{bg} := \frac{P}{4} \cdot r_{load} = 1.342 \times 10^3 \cdot in \cdot lbf$$

Torque restrained by bolt group, P is load of detector; divided by 2 over two hangers, divided again by 2 with two bolt groups per hanger.

Shear stress in fasteners due to torsional loading

$$\tau_{b1t} := \frac{T_{bg} \cdot r_1}{J_{bg}} = 3.443 \times 10^3 psi$$

$$\tau_{b2t} := \frac{T_{bg} \cdot r_2}{J_{bg}} = 1.513 \times 10^3 \text{psi}$$

$$\tau_{b3t} := \frac{T_{bg} \cdot r_3}{J_{bg}} = 3.166 \times 10^3 \text{psi}$$

Assume shear stress in fasteners due to vertical shear is evenly divided by three bolts

$$\tau_{bv} := \frac{\frac{1}{3} \cdot \frac{P}{4}}{A_{bm}} = 2.458 \times 10^3 \text{psi}$$

Vertical shear load on each fastener

By inspection, worse loaded fasteners are either is bolts 2 or 3

$$\tau_3 := \sqrt{\left(\frac{\tau_{b3t}}{\sqrt{2}}\right)^2 + \left(\frac{\tau_{b3t}}{\sqrt{2}} + \tau_{bv}\right)^2} = 5.203 \times 10^3 \text{psi}$$

$$\tau_2 := \sqrt{\left(\tau_{b2t} \cdot \cos(42°)\right)^2 + \left(\tau_{b2t} \cdot \sin(42°) + \tau_{bv}\right)^2} = 3.648 \times 10^3 \text{psi}$$

$$F_{vblt} := 12\text{ksi}$$

Allowable shear stress when threads are in the load plane for an A307 bolt in single shear. Table 7-1 in 13th ed. ASIC.

Welds on Hanger bracket

$r_{ld} := 1.25in$

$T := \frac{1}{2} \cdot \frac{P}{2} \cdot r_{ld} = 625 \cdot in \cdot lbf$ Torque on weld group

$V := \frac{1}{2} \cdot \frac{P}{2} = 500\,lbf$ Shear force on weld group

$t_w := \frac{3}{16} \cdot in \cdot \frac{1}{\sqrt{2}} = 0.133\,in$ fillet weld throat thickness

$L_w := 2.5in$ length of fillet weld

$A_w := 2 \cdot t_w \cdot L_w = 0.663\,in^2$ area of two fillet welds

$$I_{xx} := \frac{2 \cdot t_w \cdot L_w^{\ 3}}{12} = 0.345\,in^4$$

$$I_{yy} := \frac{2 \cdot L_w \cdot t_w^{\ 3}}{12} = 9.711 \times 10^{-4}\,in^4$$

$J_w := I_{xx} + I_{yy} = 0.346\,in^4$ Polar moment of inertia of weld group

$r_{wmax} := 2.125in$

$$\tau_{wt} := \frac{T \cdot r_{wmax}}{J_w} = 3.836 \times 10^3\,psi$$

$$\tau_{wv} := \frac{V}{A_w} = 754.247\,psi$$

$$\tau_w := \sqrt{\tau_{wt}^{\ 2} + \tau_{wv}^{\ 2}} = 3.909 \times 10^3\,psi$$

Weld stress is lower than allowable weld stress as shown below

Allowable weld stress

$SY_w := 70ksi$ Electrode classification E7018 welding rod

Fillet weld stress must be the lesser of:

$0.3 \cdot SY_w = 2.1 \times 10^4 psi$ Weld material

$0.4 \cdot F_y = 1.44 \times 10^4 psi$ Base material

$SA_w := .4 \cdot F_y = 1.44 \times 10^4 psi$ Allowable weld stress for fillet weld

AURORA BEARING COMPANY
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E-Mail: customerservice@aurorabearing.com Web site: www.aurorabearing.com

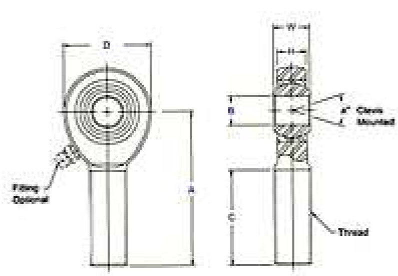
All Categories > Browse By Catalog > Commercial > Male Rod Ends > Inch Male Rod Ends > XM & XB Series Male Rod Ends, Extra Strength - Heavy Duty Shank (Ptfe Liners Available) > Item # XM-12-1

Item # XM-12-1 XM & XB Series Male Rod Ends, Extra Strength - Heavy Duty Shank (Ptfe Liners Available)

XM & XB Series Male Rod Ends, Extra Strength - Heavy Duty Shank (Ptfe Liners Available)

- Body - Carbon Steel protective coated for corrosion resistance.
- Race - Alloy steel, heat treated protective coated for corrosion resistance. (Carbon steel with Ptfe liners).
- Ball - Alloy steel, heat treated, hard chrome plated.
- All sizes solid shank type.
- Grease Fitting available for all sizes. Units are supplied without grease fittings. When a grease fitting is required, specify by adding suffix "Z" or "F". Ex.: XM-6Z = Zerk Type Fitting XM-6F = Flush Type Fitting.
- Load ratings apply only to rod ends without grease fittings. For load ratings with fittings, please consult our engineering department.
- Studded configuration available upon request. Specify by adding suffix "S". Ex.: XM-8S.
- Ptfe Liners available upon request. Specify by adding suffix "T". Ex.: XM-8T.

larger image



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Specifications

Part Number	XM-12-1
Thread Type	Right Hand
Thread Configuration	Male
B - Ball Bore [inch]	.7500
W - Ball Width [inch]	.875

FORGED EYE BOLT WARNINGS AND APPLICATION INSTRUCTIONS

Regular Nut Eye Bolt G-291 | Shoulder Nut Eye Bolt G-277 | Machinery Eye Bolt S-279 / M-279

⚠ WARNING

- Loads may slip or fall if proper eye bolt assembly and lifting procedures are not used.
- A falling load can seriously injure or kill.
- Read and understand both sides of these instructions, and follow all eye bolt safety information presented here.
- Read, understand, and follow information in diagrams and charts below before using eye bolt assemblies.

Important Safety Information - Read & Follow

Inspection/Maintenance Safety:

- Always inspect eye bolt before use.
- Never use eye bolt that shows signs of wear or damage.
- Never use eye bolt if eye or shank is bent or elongated.
- Always be sure threads on shank and receiving holes are clean.
- Never machine, grind, or cut eye bolt.

Assembly Safety:

- Never exceed load limits specified in Table I & Table 2.
- Never use regular nut eye bolts for angular lifts.
- Always use shoulder nut eye bolts (or machinery eye bolts) for angular lifts.
- For angular lifts, adjust working load as follows:

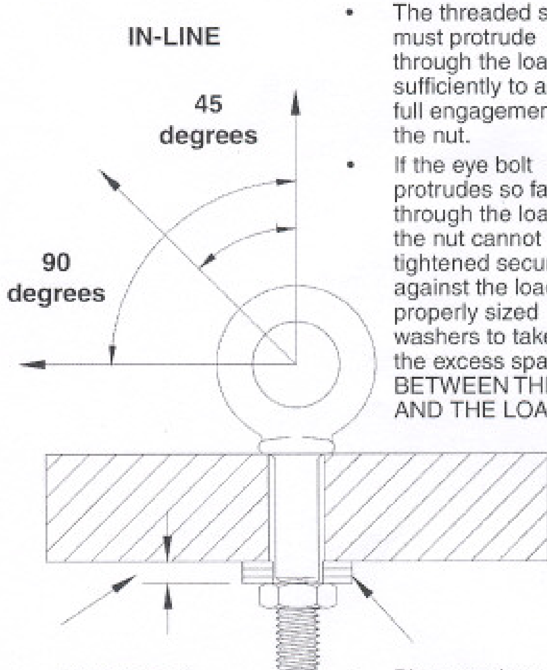
Direction of Pull (in-line)	Adjusted Working Load
45 degrees	30% of rated working load
90 degrees	25% of rated working load

- Never undercut eye bolt to seat shoulder against the load.
- Always countersink receiving hole or use washer with sufficient I.D. to seat shoulder.
- Always screw eye bolt down completely for proper seating.
- Always tighten nuts securely against the load.

Table I (In-Line Load)	
Size (in.)	**Working Load Limit (lbs.)**
1/4	650
5/16	1,200
3/8	1,550
1/2	2,600
5/8	5,200
3/4	7,200
7/8	10,600
1	13,300
1-1/8	15,000
1-1/4	21,000
1-1/2	24,000
1-3/4	34,000
2	42,000
2-1/2	65,000

Shoulder Nut Eye Bolt — Installation for Angular Loading

- The threaded shank must protrude through the load sufficiently to allow full engagement of the nut.
- If the eye bolt protrudes so far through the load that the nut cannot be tightened securely against the load, use properly sized washers to take up the excess space BETWEEN THE NUT AND THE LOAD.



- Thickness of spacers must exceed this distance between the bottom of the load and the last thread of the eye bolt.
- Place washers or spacers between nut and load so that when the nut is tightened securely, the shoulder is secured flush against the load surface.

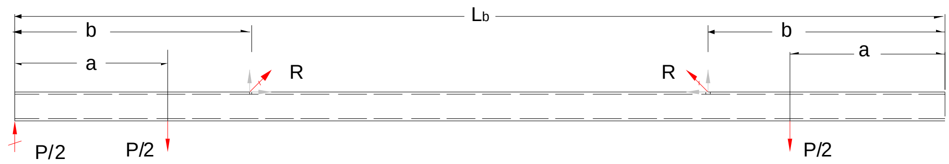
Table 2 (In-Line)	
Metric Size	**Working Load Limit - kg**
m6	200
m8	400
m10	640
m12	1000
m16	1800
m20	2500
m24	4000
m27	5000
m30	6000
m36	8500
m42	14000
m48	17300
m64	29500

Rev. 5

H - Head Width [inch]	.687
A - Base to Center [inch]	3.375
D - Head Diameter [inch]	2.000
Ball Dia. [inch]	1.312
C - Thread Length [inch]	1.875
Thread Size	3/4-16
Thread Class	UNF-3A
a - Misalignment Angle Clevis Mounted [degrees]	12
Ultimate Radial Static Load Capacity [Lbs.]	22,803
Approximate Brg. Weight [Lbs.]	0.918
Options	Stud
Lubrication Options ?	Zerk Fitting, Flush Fitting, or Ptfe Liner

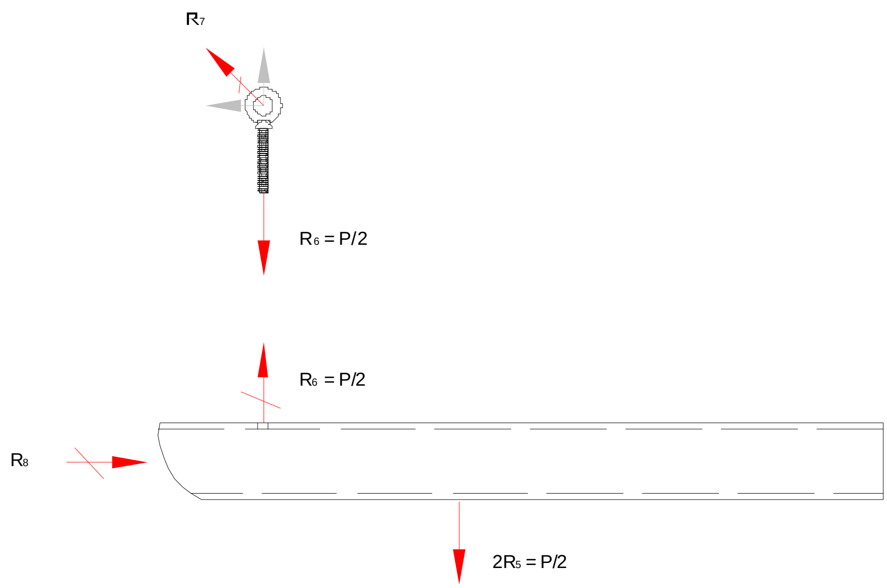
Print Back

Appendix H -Beam Support during rigging



Loading diagram of beam during rigging

Beam section between rigging points sees axial load component due to sling angle. Assume worst case slings are at 45 degrees.



$R_6 := \frac{P}{2} = 1 \times 10^3 \, \text{lbf}$ Vertical component of rigging reaction

Assuming angle between beam axis and slings is 45degrees:

$R_8 := R_6 = 1 \times 10^3 \, \text{lbf}$ Horizontal component of rigging reaction

$r := 2.7 \text{in}$ Radius of gyration about XX axis

$E = 3 \times 10^7 \, \text{psi}$

$K := 2$

$b := 48 \text{in}$ Rigging point distance from end.

$L_b = 189 \, \text{in}$

$L_{bc} := L_b - 2 \cdot b = 93 \, \text{in}$ Length of compression section in beam

$$F_a := \frac{12 \cdot \pi^2 \cdot E}{23 \left(\frac{K \cdot L_b}{r} \right)^2} = 7.882 \times 10^3 \, \text{psi}$$ Allowable axial stress

$f_a := \frac{R_8}{A} = 136.24\,psi$ Actual axial stress

Since fa/Fa <0.15:, it is only needed to check equation H1-3 to check the axial compressior and bending.

$d = 6.375\,in$ depth of beam

$e := \frac{d}{2}$ eccentricty of compressive load applied by rigging eyebolt

$M_r := R_8 \cdot e = 3.188 \times 10^3 \cdot in \cdot lbf$ Moment due to eccentric bending moment =CClockwise

$M_p := \frac{P}{2} \cdot (b - a) = 1.674 \times 10^4 \cdot in \cdot lbf$ moment between supports due to vertcal loading-clockwise

MomentResulting moment in section b<x<Lb-b

$M := M_p - M_r = 1.355 \times 10^4 \cdot in \cdot lbf$ Resulting moment

$f_b := \frac{M}{S_{xx}} = 811.527\,psi$ Actual bending stress

$F_b = 1.302 \times 10^4\,psi$ Allowable bending stress caculated above

check combined stress

$\frac{f_a}{F_a} + \frac{f_b}{F_b} = 0.08$ Is less than 1 per H1-3)

Check on thread strength in 2inch thick block welded to be in which eyebolt is threaded into.

$S_y := 36ksi$ Material properties for A36

$S_u := 58ksi$ Low estimate of UTS for A36

5/8" eyebolt

$D_p := 0.5663in$ pitch diameter

$L_t := 0.625in$ min. length of thread engagement

$A_t := \pi \cdot D_p \cdot L_t = 1.112\,in^2$ Shear area on internal treads

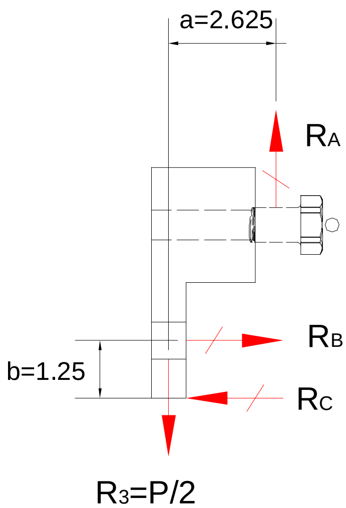
$$\tau := \frac{\frac{P}{2}}{0.87 \cdot A_t} = 1.034 \times 10^3 psi$$ Internal thread shear stress

$$\frac{S_y}{\sqrt{3}} = 2.078 \times 10^4 psi$$ Allowable shear stress based on yield

factor of safety on thread yield

$$\frac{S_y}{\sqrt{3} \cdot \tau} = 20.107$$

Appendix I - Bracket Extension - Load Case 1



$a := 2.625in$

$b := 1.25in$

$P = 2 \times 10^{3} lbf$

Sum forces in x:

$$R_b - R_c = 0$$

Sum focres in y

$$R_a := \frac{P}{2} = 1 \times 10^{3} lbf$$

Sum moments

$$R_b := \frac{a \cdot R_a}{b} = 2.1 \times 10^{3} lbf$$ axial bolt force

Check 3/4' bolt load

$R_b = 2.1 \times 10^{3} lbf$ axial bolt force

$\frac{P}{2} = 1 \times 10^{3} lbf$ shear force on bolt

For A325 3/4inch bolts:

allowable axial bolt force is 9950 lbf.
allowable shear is 5300 lbf in single shear through the threads

For A307, 3/4" bolt,
allowable axial load is 4975 lbf
allowable shear load is 2650 lbf in single shear.

Calculate bolt stress to take into account combined axial and shear loading

$$A_{blt} := 0.442\text{in}^2$$

$$f_{nt} := \frac{R_b}{A_{blt}} = 4.751 \times 10^3\,\text{psi}$$

$$f_{nv} := \frac{P}{2A_{blt}} = 2.262 \times 10^3\,\text{psi}$$

$F_{nt} := 11200\text{psi}$ allowable tensile stress for A307, 3/4" bolt.

$F_{nv} := 6000\text{psi}$ allowable shear stress for A307, 3/4" bolt.

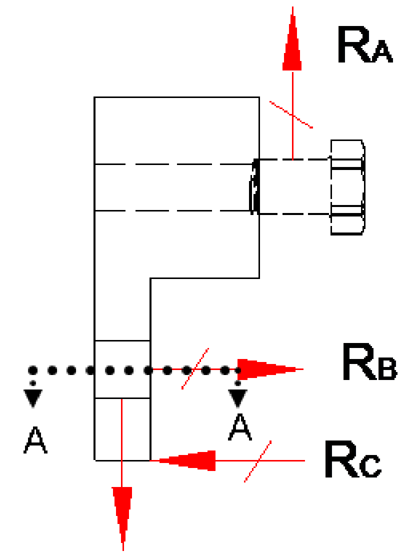
$$F_{ntprime} := 1.3 \cdot F_{nt} - \frac{2 \cdot F_{nt}}{F_{nv}} \cdot f_{nv} = 6.114 \times 10^3\,\text{psi}$$

nominal tensile stress modified to include effects of shear. J3-3b.

Bolt stress is OK.

Check bending stress.

Maximum bending stress occurs at the cross section of the fastener



Section A-A

Section inertia about z axis at section A-A

$$I := \frac{(3.25in - 0.81in)\cdot(0.75in)^3}{12} = 0.086\,in^4$$

$c := 0.375in$ distance from midplane to extreme fiber

$$M := \frac{P\cdot a}{2} = 2.625 \times 10^3 \cdot in\cdot lbf$$

$$f_b := \frac{M\cdot c}{I} = 1.148 \times 10^4 psi$$ bending stress

$$F_b := 0.66\cdot F_y = 2.376 \times 10^4 psi$$ allowable bending stress

$$A := (3.25in - 0.81in)\cdot(0.75in) = 1.83\,in^2$$

$$f_a := \frac{P}{2\cdot A} = 546.448\,psi$$ axial stress

$$F_y = 3.6 \times 10^4 \text{psi} \qquad F_u = 5.8 \times 10^4 \text{psi}$$

use lower of

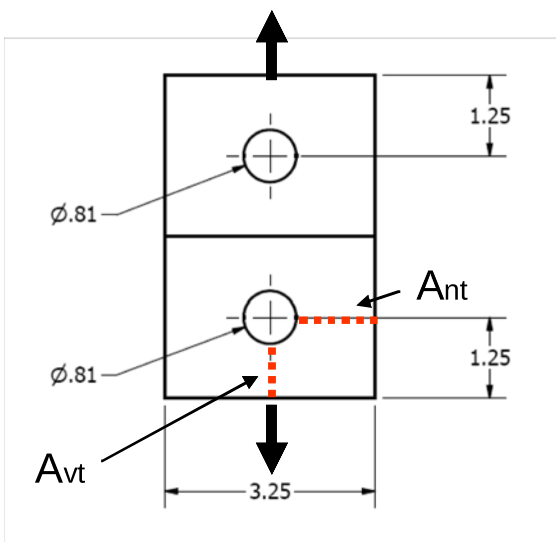
$$F_a := \frac{F_u}{2} = 2.9 \times 10^4 \text{psi}$$

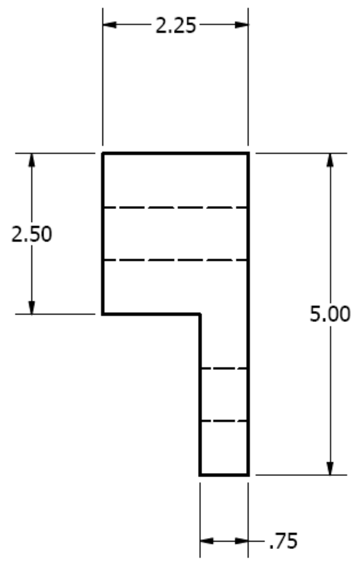
for allowable stress in tension

$$F_a := \frac{F_y}{1.67} = 2.156 \times 10^4 \text{psi}$$

bending stress is acceptable. Axial stress is trivially low.

Check for block shear





$$A_{nt} := \frac{(3.25\text{in} - 0.81\text{in})}{2} \cdot 0.75\text{in} = 0.915\,\text{in}^2$$

$$A_{nv} := \frac{(2.5\text{in} - 0.81\text{in})}{2} \cdot 0.75\text{in} = 0.634\,\text{in}^2$$

$$A_{gv} := \frac{2.25in}{2} \cdot 0.75in = 0.844\,in^2$$

$$U_{bs} := 1$$

$F_u = 5.8 \times 10^4 psi$ $\qquad$ $F_y = 3.6 \times 10^4 psi$

allowable shear strength is lesser of

$$R_n := \frac{0.6 \cdot F_u \cdot A_{nv} + U_{bs} \cdot F_u \cdot A_{nt}}{2} = 3.756 \times 10^4 lbf$$

J4-5 (ASIC 13th ed.)

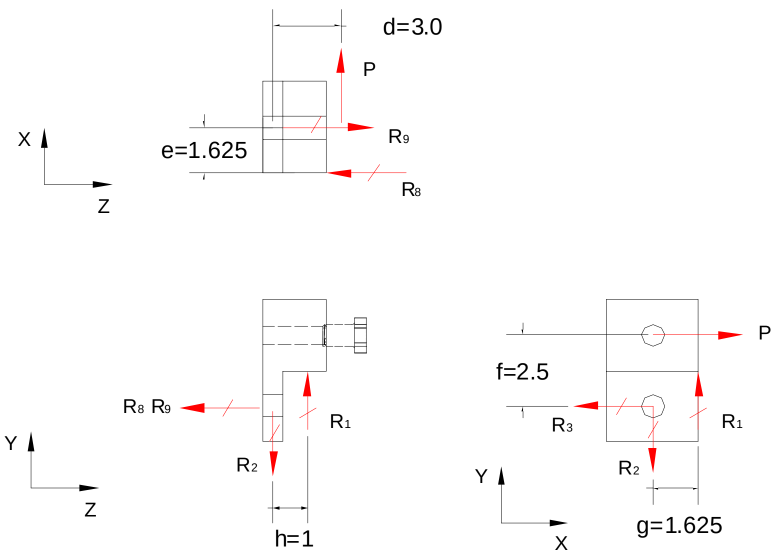
$$R_n := \frac{0.6 \cdot F_y \cdot A_{gv} + U_{bs} \cdot F_u \cdot A_{nt}}{2} = 3.565 \times 10^4 lbf$$

$$r_n := \frac{P}{2} = 1 \times 10^3 lbf$$

actual shear is much less than allowable

Note block shear is checked for thinner section and by inspection the thicker top section is acceptable.

Appendix J Bracket extension - Load case 2



Calculate reaction forces

$P := 500\text{lbf}$

$g := 1.625\text{in}$ $\quad d := 3\text{in}$

$f := 2.5\text{in}$ $\quad e := 1.625\text{in}$

sum forces in X $\quad R_3 := P = 500\,\text{lbf}$

sum forces moments about Z $\quad R_1 := \frac{f}{g}\cdot P = 769.231\,\text{lbf}$

sum foces in Y $\quad R_2 := R_1 = 769.231\,\text{lbf}$

sum moments about Y $\quad R_9 := \frac{d}{e}\cdot P = 923.077\,\text{lbf}$

sum forces in Z $\quad R_8 := R_9 = 923.077\,\text{lbf}$

vector sum at lower hole $\quad R_t := \sqrt{R_2^{\,2} + R_3^{\,2}} = 917.451\,\text{lbf}$

Bolt Forces

Forces R3 and R9 are taken by 3/4in bolt

$A_{blt} := 0.442in^2$

$f_{nt} := \frac{R_8}{A_{blt}} = 2.088 \times 10^3 psi$

$f_{nv} := \frac{R_3}{2A_{blt}} = 565.611 psi$

$F_{nt} := 11200psi$ allowable tensile stress for A307, 3/4" bolt.

$F_{nv} := 6000psi$ allowable shear stress for A307, 3/4" bolt.

$F_{ntprime} := 1.3 \cdot F_{nt} - \frac{2 \cdot F_{nt}}{F_{nv}} \cdot f_{nv} = 1.245 \times 10^4 psi$ nominal tensile stress modified to include effects of shear. J3-3b.

Bolt stress is OK.

Tensile stress in bottom section due to R2 using net area at cross section of hole

$f_a := \frac{R_2}{2A_{nt}} = 420.345 psi$ $A_{nt} = 0.915 in^2$

Shear stress due to R3

$f_v := \frac{R_3}{A_{nv}} = 788.955 psi$ $A_{nv} = 0.634 in^2$

Stresses are very low, by inspection it is not necessary to check combined effects .
Block shear calculation in appendix I revealed allowable foce of 34kips which is much higer tha
combined effects of R2 and R3.

Check bending stress at section A-A due to couple R1 R2

Maximum bending stress occurs at the cross section of the fastener

Section inertia about z axis at section A-A

$$I := \frac{(3.25in - 0.81in)\cdot(0.75in)^3}{12} = 0.086\,in^4$$

$c := 0.375in$ distance from midplane to extreme fiber

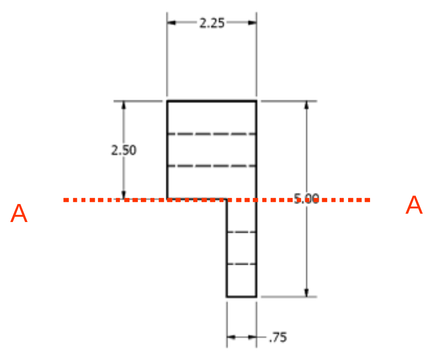
$h := 1in$

$$M := \frac{P\cdot h}{2} = 250\cdot in\cdot lbf$$

$f_b := \frac{M\cdot c}{I} = 1.093\times 10^3 psi$ bending stress

$F_b := 0.66\cdot F_y = 2.376\times 10^4 psi$ allowable bending stress

Check torsional shear stress at section A due to load P



$P = 500\,lbf$ $d = 3\,in$

$T_p := P\cdot d = 1.5\times 10^3\cdot in\cdot lbf$

dimensions of section for use in formula

$a := \frac{3.25}{2} in$ $\quad$ $b := \frac{0.75}{2} in$

$$\tau_{max} := \frac{3 \cdot T_p}{8 \cdot a \cdot b^2} \cdot \left[1 + 0.6095 \cdot \frac{b}{a} + 0.8865 \cdot \left(\frac{b}{a}\right)^2 - 1.8023 \cdot \left(\frac{b}{a}\right)^3 + 0.91 \cdot \left(\frac{b}{a}\right)^4 \right] = 2.876 \times 10^3 psi$$

formula from Roark's 20-4

stress is acceptable

Appendix K

$A_{blt} := 0.0678in^2$ Minor area for a 3/8-16 UNC fastener

Bolt with highest shear loading

$V_x := 317lbf$

$V_y := 131lbf$

$V_{res} := \sqrt{V_x^2 + V_y^2} = 343.001\,lbf$

$N := 255lbf$

$f_{nt} := \frac{N}{A_{blt}} = 3.761 \times 10^3 psi$

$f_{nv} := \frac{V_{res}}{2A_{blt}} = 2.53 \times 10^3 psi$

$F_{nt} := 11200psi$ allowable tensile stress for A307, 3/4" bolt.

$F_{nv} := 6000psi$ allowable shear stress for A307, 3/4" bolt.

$F_{ntprime} := 1.3 \cdot F_{nt} - \frac{2 \cdot F_{nt}}{F_{nv}} \cdot f_{nv} = 5.116 \times 10^3 psi$ nominal tensile stress modified to include effects of shear. J3-3b.

Bolt stress is OK.

Bolt with highest tensile load

$V_x := 21lbf$

$V_y := 101\text{lbf}$

$V_{res} := \sqrt{V_x{}^2 + V_y{}^2} = 103.16\,\text{lbf}$

$N := 411\text{lbf}$

$f_{nt} := \dfrac{N}{A_{blt}} = 6.062 \times 10^3\,\text{psi}$

$f_{nv} := \dfrac{V_{res}}{2A_{blt}} = 760.767\,\text{psi}$

$F_{nt} := 11200\text{psi}$ allowable tensile stress for A307, 3/4" bolt.

$F_{nv} := 6000\text{psi}$ allowable shear stress for A307, 3/4" bolt.

$F_{ntprime} := 1.3 \cdot F_{nt} - \dfrac{2 \cdot F_{nt}}{F_{nv}} \cdot f_{nv} = 1.172 \times 10^4\,\text{psi}$ nominal tensile stress modified to include effects of shear. J3-3b.

Bolt stress is OK.

Appendix K – Reactions forces for LC2 of Hanger bracket

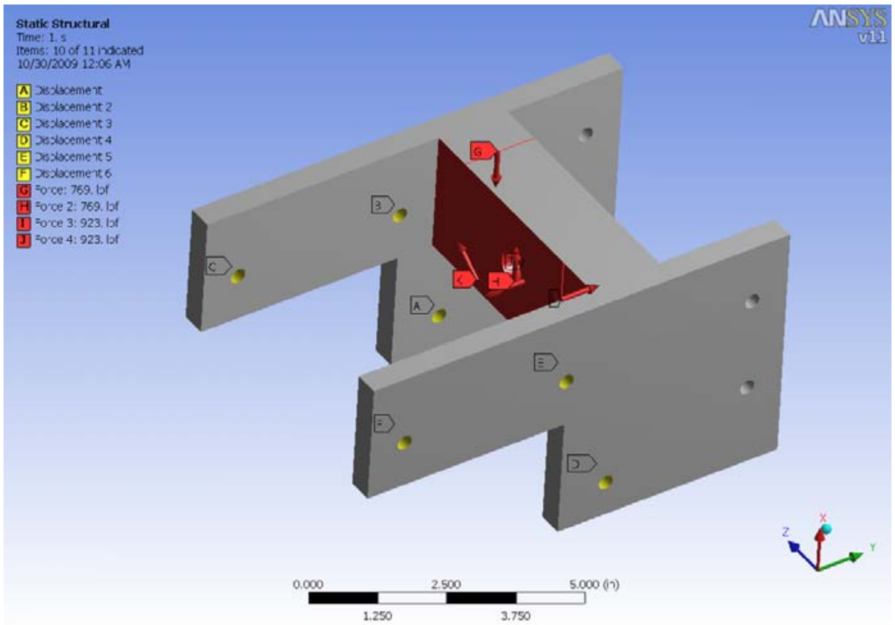


Figure 1 Applied boundary conditions for hanger bracket

TABLE 14
Model > Static Structural > Solution > Probes

Object Name	*Force Reaction*	*Force Reaction 2*	*Force Reaction 3*	*Force Reaction 4*	*Force Reaction 5*
State	Solved				
Definition					
Type	Force Reaction				
Location Method	Boundary Condition				
Boundary Condition	Displacement	Displacement 2	Displacement 3	Displacement 4	Displacement 5
Options					
Result Selection	All				
Display	All Time Points				
Maximum Value Over Time					
X Axis	82.714 lbf	-21.467 lbf	-12.616 lbf	-232.25 lbf	-317.33 lbf
Y Axis	92.132 lbf	101.03 lbf	2.8131 lbf	-47.696 lbf	-131.23 lbf
Z Axis	-29.357 lbf	-411.14 lbf	10.187 lbf	-90.949 lbf	-255.41 lbf
Total	127.25 lbf	423.91 lbf	16.457 lbf	253.95 lbf	427.96 lbf

Minimum Value Over Time					
X Axis	82.714 lbf	-21.467 lbf	-12.616 lbf	-232.25 lbf	-317.33 lbf
Y Axis	92.132 lbf	101.03 lbf	2.8131 lbf	-47.696 lbf	-131.23 lbf
Z Axis	-29.357 lbf	-411.14 lbf	10.187 lbf	-90.949 lbf	-255.41 lbf
Total	127.25 lbf	423.91 lbf	16.457 lbf	253.95 lbf	427.96 lbf

TABLE 15
Model > Static Structural > Solution > Probes

Object Name	*Force Reaction 6*
State	Solved
Definition	
Type	Force Reaction
Location Method	Boundary Condition
Boundary Condition	Displacement 6
Options	
Result Selection	All
Display	All Time Points
Maximum Value Over Time	
X Axis	1.1467 lbf
Y Axis	-17.29 lbf
Z Axis	7.6685 lbf
Total	18.949 lbf
Minimum Value Over Time	
X Axis	1.1467 lbf
Y Axis	-17.29 lbf
Z Axis	7.6685 lbf
Total	18.949