BELOW-THE-HOOK LIFTING DEVICE
Engineering Note Cover Page

Lifting Device Numbers: 138

FNAL Site No.:______ (if applicable) Div. Specific No.: ______ (if applicable) Asset No. ______ (if applicable)

ASME B30.20 Group: (check one)

- X Group I Structural and Mechanical Lifting Devices
- ˇ Group II Vacuum Lifting Devices
- ˇ Group III Magnets, Close Proximity Operated
- ˇ Group IV Magnets, Remote Operated

Device Name or Description: Horn Lifting Fixture

Device was: (check all applicable)

- ˇ Purchased from a Commercial Lifting Device Manufacturer
  mfg. name: ______
- X Designed and Built at Fermilab
- ˇ Designed by Fermilab and Built by a Vendor
  Assy drawing number: ______
- ˇ Provided by a User or Other Laboratory
- ˇ Other. Describe:______

Engineering Note Prepared by: Bob Wands Date : 1/16/03

Engineering Note Reviewed by: Ang Lee Date: 10/21/03

Lifting Device Data:

Capacity: 3000#

Fixture Weight: 120#

Service: X normal ˇ heavy ˇ severe (refer to B30.20 for definitions)

Duty Cycle: ______8, 16 or 24 hour rating (applicable to groups III, and IV)

Inspections Frequency: ______

Rated Load Test by FNAL (if applicable): Date : 3/19/03 Load: 3750#

ˇ Check if Load Test was by Vendor and attach the certificate.

Satisfactory Load Test Witnessed by: Bob Wands, Gary Trotter

Signature (of Load Test Witness): [signatures]

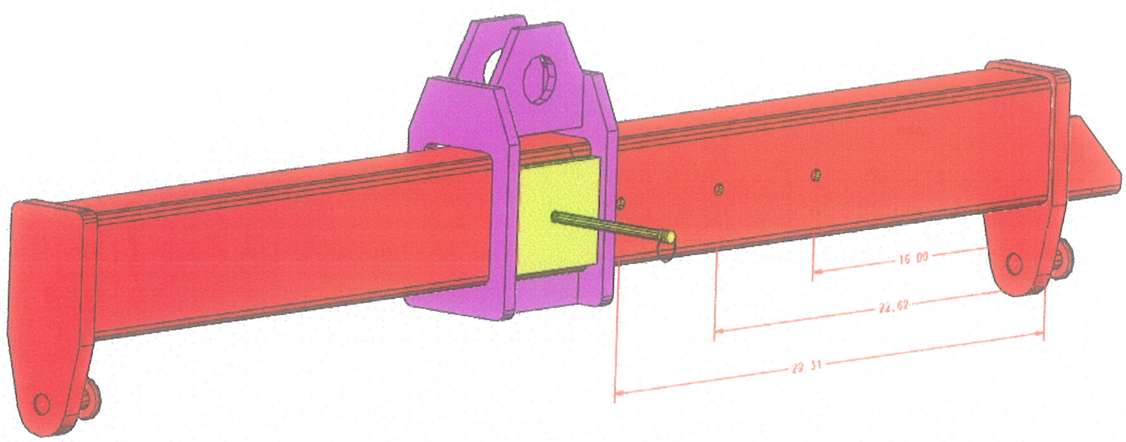
Notes or Special Information:

Horn Lifting Fixture Modification
Engineering Note # 138 from 1/16/03

1. A request has been made by Dave Pushka PPD (Mechanical Engineering) to add fixed locating positions to this lifting fixture for the remote lifting of Horns #1, #2 and Target Carrier and that I should contact Hiep Le (Accelerator Division) for these locations.

2. I contacted Hiep Le for this information and he referred me to visiting MI-8, viewing the fixture and measuring the markings on this fixture which I have done. These dimensions as shown on the Solid Model below from the inside of the downstream end plate are 16", 22 5/8" and 29 5/16".

5. I have since pulled 2D construction drawings from which I made a Solid Model of existing fixture (shown below), lifting tube assembly in red and pick point slider in magenta, I then added pin locating plates in yellow to both sides of slider weldment and a positioning pin with detent also in yellow. As the slider is captured by the tube endplates and for simplicity of aligning pinning holes, slider block should be positioned in each of the pre measured locations and line bore through these plates as well as the lifting tube after welding.



6. As the locating pins penetrate the lifting tube on the neutral axis, no additional calculations are needed for this engineering note authored by Bob Wands from 1/16/03. This statement will be inserted into original Engineering Note #138 to document changes made and to illustrate that the modifications have been given consideration in this engineering note. E. LaVallie 3/17/05.

MSG-EAR-03-322
January 16, 2003

Stress Analysis of Horn Lifting Fixture

Bob Wands

Introduction and Summary

The lifting fixture used for the handling of the Numi horns consists of a 6x4x0.25 in rectangular tube, supported from the 30 T crane, with pins at each end which mate with holes in two clamps around the body of the horn.

The center of gravity of the horn does not lie midway between the clamps, so the beam is supported from the crane by a sliding collar which allows the pick point to be positioned directly above the horn cg.

The maximum design load on either end of the rectangular tube is 3000 lbs, which is equal to slightly more than the total weight of horn #2 plus its stripline. The total capacity of the fixture is therefore 6000 lbs.

This analysis shows that the lifting fixture and horn collars are well within the design requirements for Fermilab lifting fixtures, which allow stresses to reach a maximum of 1/3 of the yield stress.

Geometry

The lifting fixture geometry and dimensions were taken from Drwg. 8875.126-ME-427484

Material Properties and Allowable Stresses

The 6x4x0.25 in rectangular tube is A570 Gr B steel with a minimum specified yield stress of 46 ksi. The maximum allowable stress is S = 46/3 = 15.3 ksi.

The plates which form the sliding collar and shackle engagement hole, and the plates and pins welded to the tube ends, are 1020 HR steel, with a minimum specified yield stress of 30 ksi. The maximum allowable stress is S = 30/3 = 10 ksi.

The clamps on the horn are 6061-T6 Al, with a minimum specified yield stress of 35 ksi. The maximum allowable stress is S = 35/3 = 11.6 ksi.

Weld metal shear stress for steel welds, normally limited to $F_u/3$, where F_u is the tensile strength of the weld metal (70 ksi for 70XX electrodes), is limited to $F_u/6$ = 11.6 ksi for this analysis. Stress in the parent material near the weld region is limited to the working stress of the parent material.

Stresses in Rectangular Tube

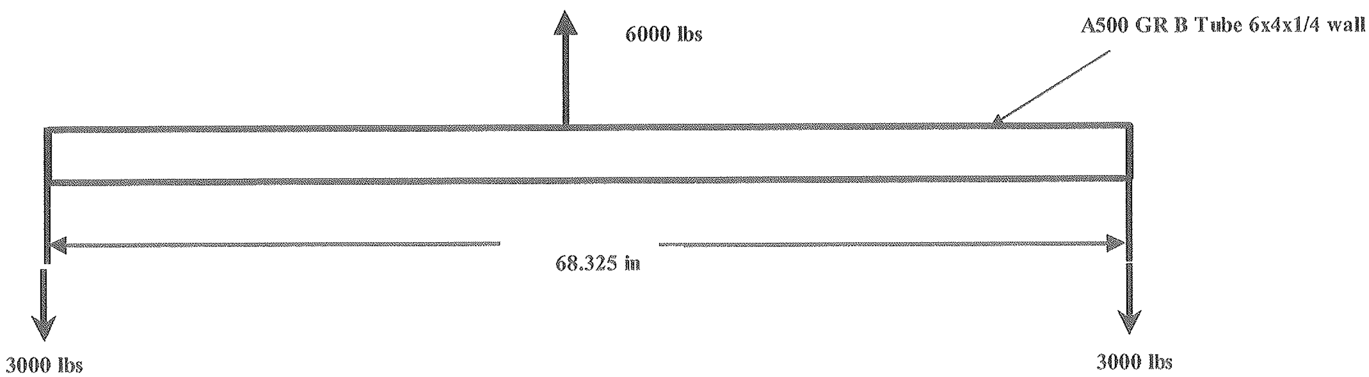


Figure 1. Loading on Rectangular Tube

The maximum moment on the beam is M = 3000(68.375/2) = 102500 lb-in, and occurs at the crane picking point. The maximum bending stress is

$$\sigma_b = Mc/I = 102500(3)/22.1$$

$$\sigma_b = 13900 \text{ psi} < 15.3 \text{ ksi}$$

The beam must meet two other criteria. First, the depth-thickness ratio of the web or webs shall meet the requirement

$$d/t \leq 257/\text{sqrt}(Fy) = 37.9$$

For this beam, d/t = 6/.5 = 12. Therefore, the criterion is satisfied.

Second, the laterally unsupported length of the compression flange for a box member must meet certain requirements, but need not be smaller than 1200(b/Fy) = 1200(4/46) = 104 in. The total length of the lifting beam is 68.375 in. Therefore, the criterion is satisfied.

FE Model of End Bracket, Pin, and Horn Clamp

An FE model of the end bracket, pin, and horn clamp is shown in Fig. 2a.

The model uses deformable contact elements between the pin and clamp plate to realistically simulate the pin contact.

The pin and end bracket are connected only at the back surface of the bracket (where the two parts will be welded), and gap elements with an initial clearance of zero are used at all other locations on the pin/bracket interface.

The resulting deformation and stresses is shown in Fig. 2b.

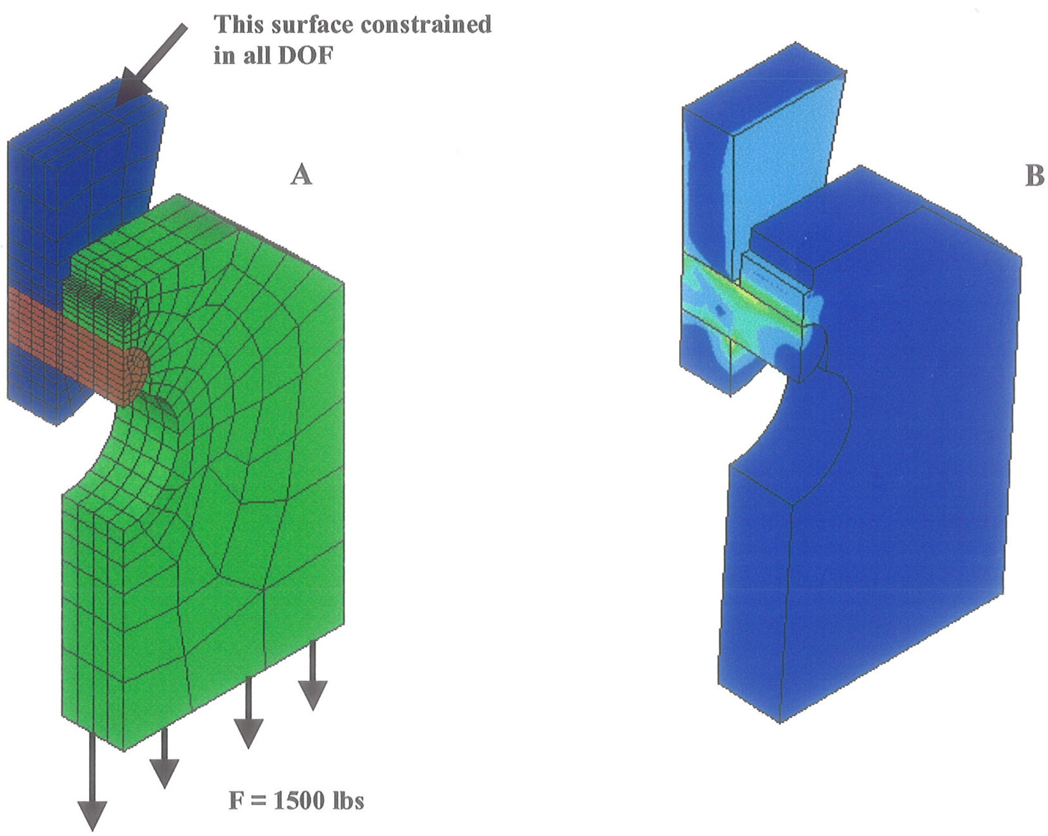


Figure 2. FE Model of End Bracket, Pin , and Horn Clamp

End Bracket Stresses

The end bracket is subjected to normal forces and moments on a section through the pin hole. These forces, as calculated by the FEA, are shown in Fig. 3. The normal force of 4554 lbs is larger than the applied load of 3000 lbs on the assembly. This is because, for equilibrium, the section chosen must include the pin and its weld to the bracket, because it is these components that provide the 1554 lbs reaction necessary to balance the external loads.

The area A and moment of inertia Ixx of the bracket across the section shown are:

$$A= 2(1.5-.5) = 2 \text{ in}^2$$

$$Ixx = 2(1^3)/12 = 0.167 \text{ in}^4$$

The normal stress S_n and bending stresses S_b are:

$$S_n = P/A = 4554/2 = 2277 \text{ psi}$$

$$S_b = Mc/Ixx = 2400(.5)/.167 = 7185 \text{ psi}$$

The sum of normal and bending stresses is then 9462 psi. This is less than the 10 ksi stress limit for the bracket material

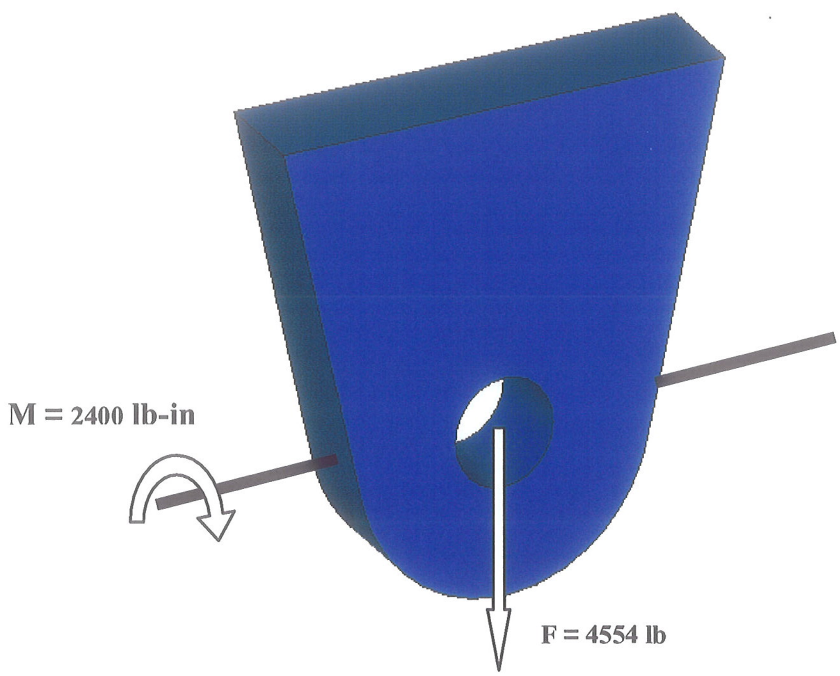


Figure 3. Forces and Moments from FE Model of Assembly

Pin Bearing Stress on End Bracket

The average bearing stress on the end bracket due to the pin load is

$$S_{bearing} = 3000/(1) = 3000 \text{ psi}$$

This is the average stress, found by assuming a uniform application of the load over the available bearing area of 1 in^2; concentrations occur at the lower surface of the pin hole nearest the horn clamp. But these concentrations are not relevant in this analysis.

Welds between End Bracket and Beam

The end bracket is attached to the 6x4 box beam with ¼ in fillet weld around three sides. Considering only the two 6 inch long sides, the available stress area is

$$A = 6(2)(0.25)(0.707) = 2.1 \text{ in}^2$$

The shear stress under a load of 3000 lbs is

$$S = 3000/A = 1428 \text{ psi}$$

This is well within the shear stress limit of 11.6 ksi for the weld metal.

The shear stress on the parent material is lower, since the shear area is the full 0.25 in leg of the weld. Therefore, parent material stress limits are satisfied.

Pin Stresses

The pins which engage the hole in the horn clamps are in single shear. For a pin diameter of 1 inch, the nominal shear stress under a 3000 lb load is $\tau = 3000/(\pi 0.5^2) = 3820$ psi. This is smaller than the maximum allowable shear stress of 5 ksi.

The forces on the pin transmit both a moment (due to the offset of the load from the plane of the bracket) and a normal force to the bracket.

Horn Clamp Stresses

The stresses in the horn clamp are shown in Fig. 4. No stress exceeds 8500 psi. Therefore, no stress exceeds the 11.6 ksi limit for the aluminum clamp.

Horn Clamp Bolts

The horn clamp is attached to the horn with four 3/8x16 Gr. 8 bolts. The proof strength of a Grade 8 material is 120 ksi. The stress area of a single bolt is 0.0775 in^2, giving a total stress area of 0.31 in^2. The stress for a load of 3000 lbs is $\sigma = 3000/0.31 = 9677$ psi, well within the capacity of the four bolts.

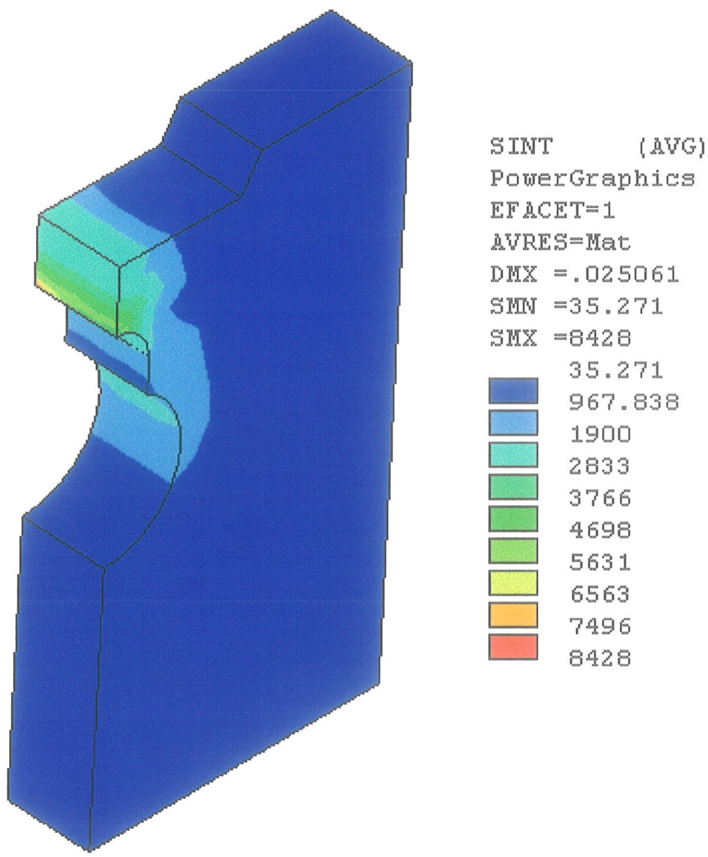


Figure 4. Stresses in Horn Clamp from FE Model of Assembly

Beam Sliding Bracket Stresses

An FEA model of the bracket that slides along the 6x4 tube, allowing it to accommodate the off-center CG of the horn was created. This model is shown in Fig. 5a

The maximum stresses from the FE model are shown in Fig. 5b, and do not exceed 9000 psi. Therefore, no stress exceeds the 10 ksi limit for the plate material.

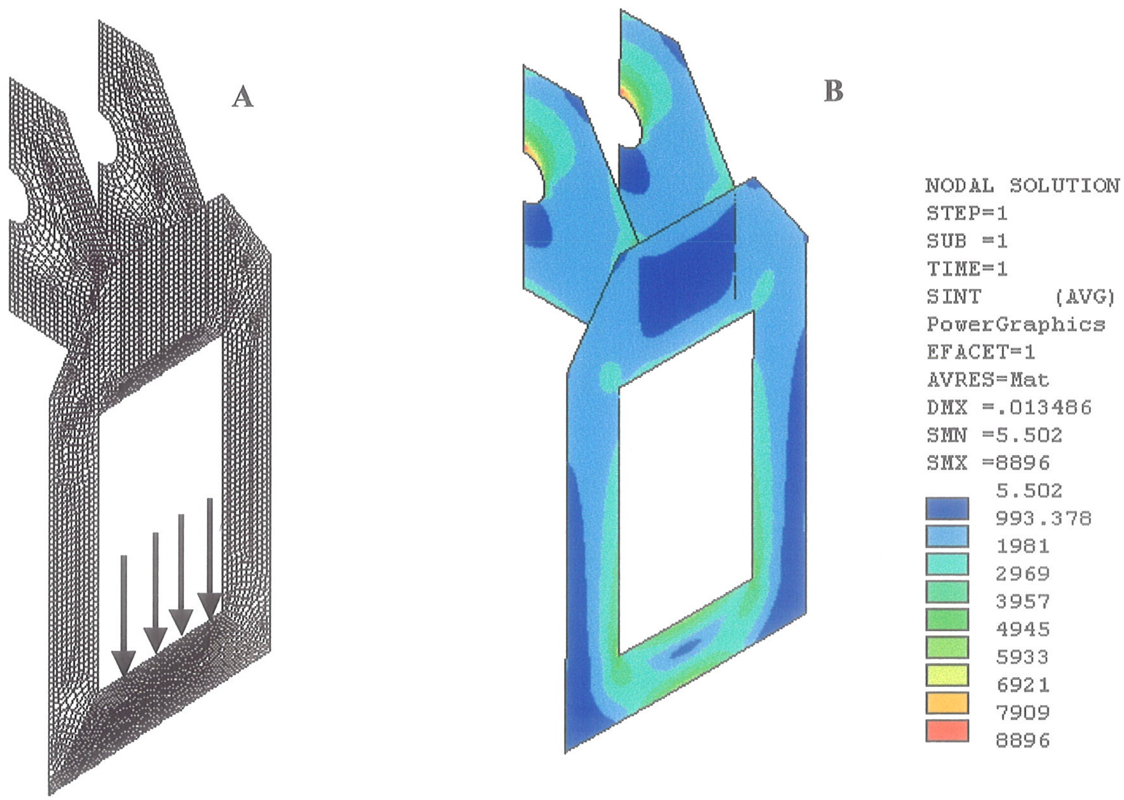


Figure 5. FE Model of Sliding Bracket