BELOW-THE-HOOK LIFTING DEVICE
Engineering Note Cover Page

Lifting Device Numbers:

FNAL Site No.: 132 (if applicable) Div. Specific No.: ______ (if applicable) Asset No. ______ (if applicable)

ASME B30.20 Group: (check one)
- ˇ 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: NuMI Module Lifting Fixture

Device was: (check all applicable)
- ˇ Purchased from a Commercial Lifting Device Manufacturer
  mfg. name: ______
- ✓ 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: 11/28/01

Engineering Note Reviewed by: [signature] Date: 11/28/01

Lifting Device Data:

Capacity: 60000 lbs

Fixture Weight: 1260 lbs

Service: ˇ 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: Normal Service

Rated Load Test by FNAL (if applicable): Date: 3/26/02 Load: 75,000 lbs

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

Satisfactory Load Test Witnessed by: GARRETT R TROTTER

Signature (of Load Test Witness): Donald R Trotter

Notes or Special Information:

November 28, 2001

Stress Analysis of Numi Horn Module Lifting Fixture

Bob Wands

Introduction and Summary

The Numi horn module will be lifted by a fixture which is designed to allow for variation of the center of gravity of the module. This is accomplished by mating gear racks. One pair of racks is mounted on a large rectangular frame, which contains the pins that engage the module lifting hooks. The other pair of racks is mounted on a separate beam, which is held by the crane hook. The beam can be moved along the length of the frame until the appropriate location is reached for the existing center of gravity. Lifting the beam engages the teeth of the racks.

Fig. 1 illustrates how the lifting fixture is adjusted for variations in the center of gravity of the module. The lifting fixture will be used with a 30 ton crane. The worst possible loading is assumed to occur when the beam is pushed as far toward one end of the frame as possible. This loading is assumed to be 30000 lbs/pin on the two pins nearest the beam.

In addition to this load case, the symmetric case (with the lifting beam centered on the side tubes) was also run, to look at the stresses in the side tube only.

Geometry

The lifting fixture geometry was taken from Drwg # 8875.111-ME-406392 and its associated details.

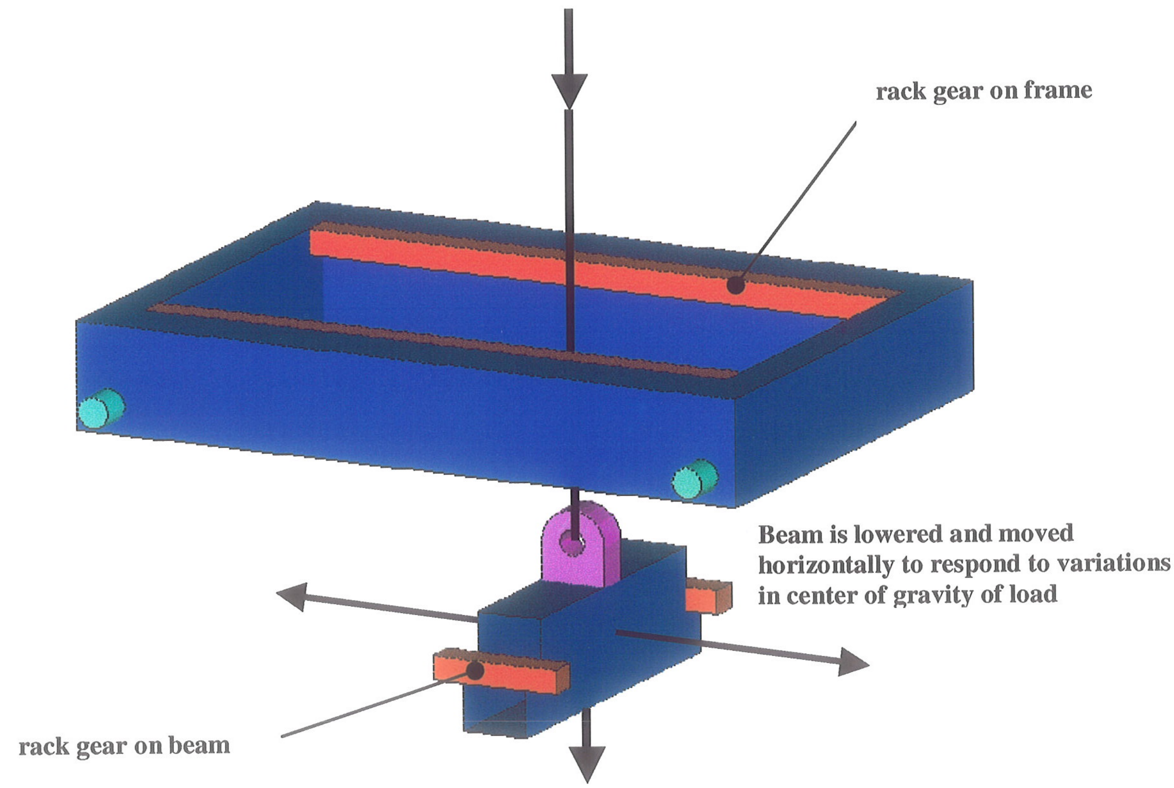
Allowable Stresses

The tubing in the frame is A500 Grade B structural steel, with a minimum specified yield strength $S_y = 46$ ksi. The pins are 4130 cold drawn steel, with a vendor's stated yield strength of 87 ksi. All other components area A36 structural steel, with a minimum specified yield strength of 36 ksi.

Consistent with Fermilab lifting fixture design requirements, stresses in the lifting fixture will be limited to $S_y/3$, except near concentrations where stresses in excess of $S_y/3$ are expected, and will not jeopardize the load-carrying capacity of the ductile steel members.

Stresses will, for the most part, be linearized across a section passing through the component. This procedure gives primary membrane plus primary bending stresses, and effectively eliminates consideration of peak stresses, which are relevant only for fatigue.

Allowable stresses are summarized in Table I for the components identified in Fig. 2.



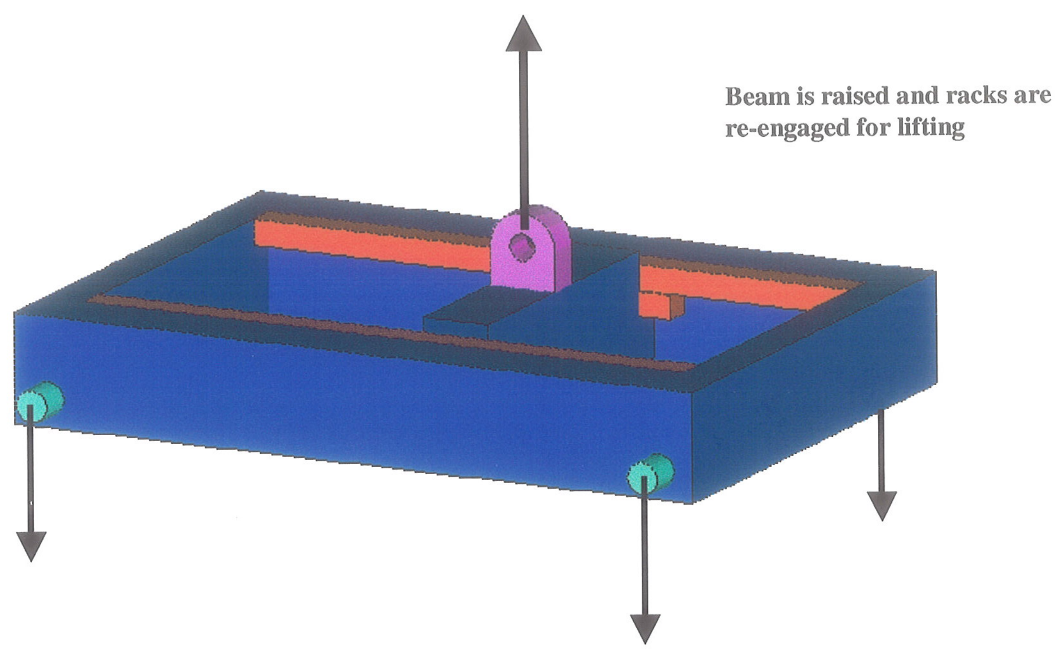


Figure 1. Adjustment of Lifting Fixture to Compensate for Change in CG of Module

Table I. Component Material and Allowable Stress

Component	Material	Minimum Yield Stress (ksi)	Maximum Normal Stress (ksi)	Maximum Shear Stress (ksi)
Frame Tubes	A500 Tube	46	15.3	7.65
Racks	A36 Plate	36	12	6
Pins	4130 Cold Drawn Bar	87	29	14.5
Lifting Eye	A36 Plate	36	12	6
Lifting Beam	A36 Plate	36	12	6

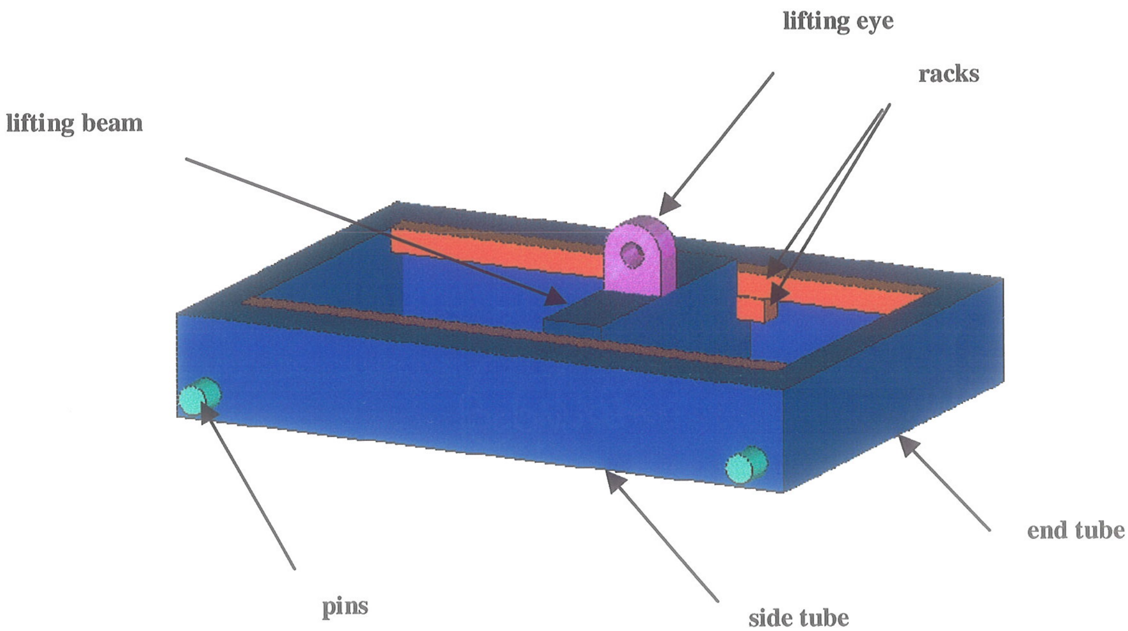


Figure 2. Components of Lifting Fixture

Welds are made with E70XX rod, with an ultimate strength of 70 ksi. From the AISC Specification for the Design, Fabrication and Erection of Structural Steel for Buildings, both normal and shear stresses in groove and fillet welds is limited to $0.3S_u$, or 21 ksi. Because the base metal working stresses are defined as $S_y/3$, rather than the $0.6S_y$ normally allowed for such stresses in structural steel design, the working stresses in the welds should also be reduced to maintain a similar margin of safety. A rational de-rating of this stress would be use $0.2S_u$, or 1/5 of the ultimate strength of the weld metal. The working stress for weld metal, in both normal and shear directions, is therefore 14 ksi.

Finite Element Model

The finite element model is shown in Fig. 3. It consists of 5500 4-node plate elements to model the A500 tubing, and 4700 8-node solid elements to model the racks, lifting eye, and pins. The lifting beam is placed as far toward one set of pins as possible. Seven components are shown in the figure, and each will be specifically addressed by the analysis.

The pins are inserted into the tubes, but are only welded at one side. The side nearest the load is not welded, but relies on the high-tolerance of the fit to ensure adequate bearing. Fig. 4 shows the details of this pin-to-tube connection.

In order to understand the effective length of the pin, an FE model created previously for the analysis of the module hooks was used. This model, shown in Fig. 5, places the hook its maximum distance from the tube, subjecting the pin to its maximum bending moment. In addition, interface elements allow the pin to deform and assume a natural contact pattern against the hook, which effectively shortens the lever arm. This effective lever arm length can be calculated from the known moment applied to the pin. This effective length is then used to locate the loading points on the pins in the full model of Fig. 3.

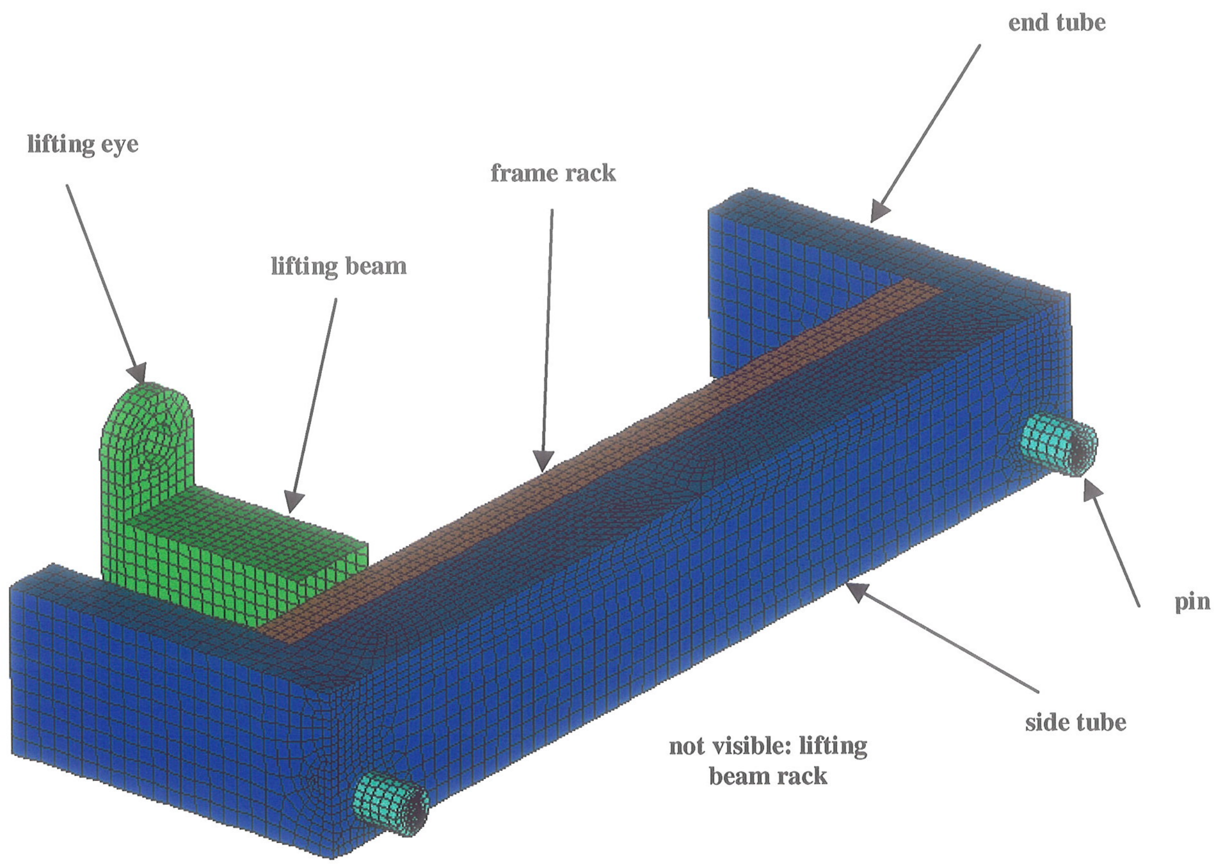


Figure 3. Finite Element Model of Lifting Fixture

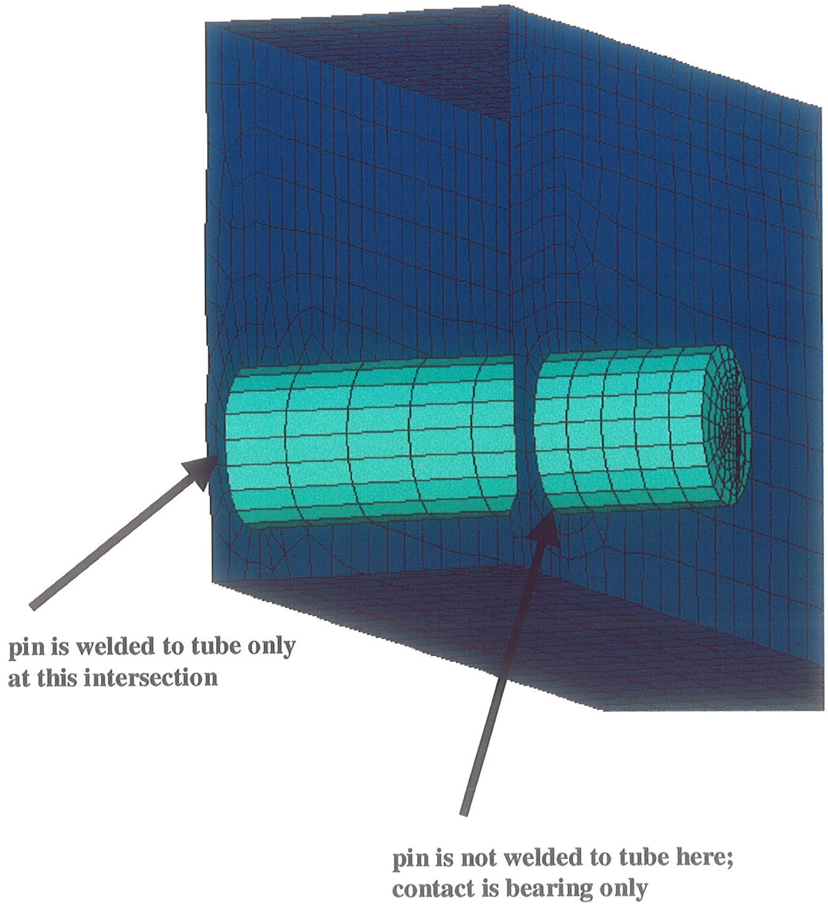


Figure 4. Detail of Pin-toTube Connection

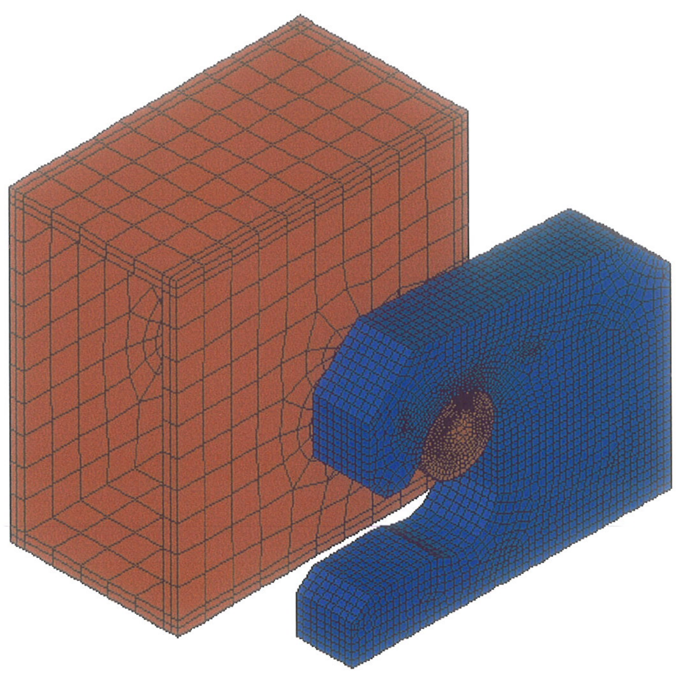


Figure 5. FE Model used for Hook/Pin Interaction Analysis

The model was loaded by constraining the lifting pins and applying a vertical force on the lifting eye.

Two load cases were run:

1. Load Case 1 – this is the case with the lifting beam positioned as close as possible to one side of the frame. The load on the lifting eye is adjusted to give a pin force of 30000 lbs. Because the two pins furthest from the lifting beam must still take some load, the total load applied to the lifting eye is 70 tons, or 10 tons in excess of the crane rating.
2. Load Case 2 – this is the case in which the lifting beam is positioned midway between the end tubes. This case was used *only* to look at the maximum stresses in the center of the side tubes.

All results discussed below are from Load Case 1 unless noted.

Results

Side Tube

The side tube is subjected to bending stresses, and high local stresses in the regions of the pins.

Fig. 6 shows the bending stresses in the horizontal direction . The line A-B is the section across which these stresses are greatest. Linearizing the stress intensity across this section into membrane and bending components shows that the total primary membrane plus bending stress intensity (P_m+P_b) is 9.4 ksi. This is well below the allowable of 15.3 ksi.

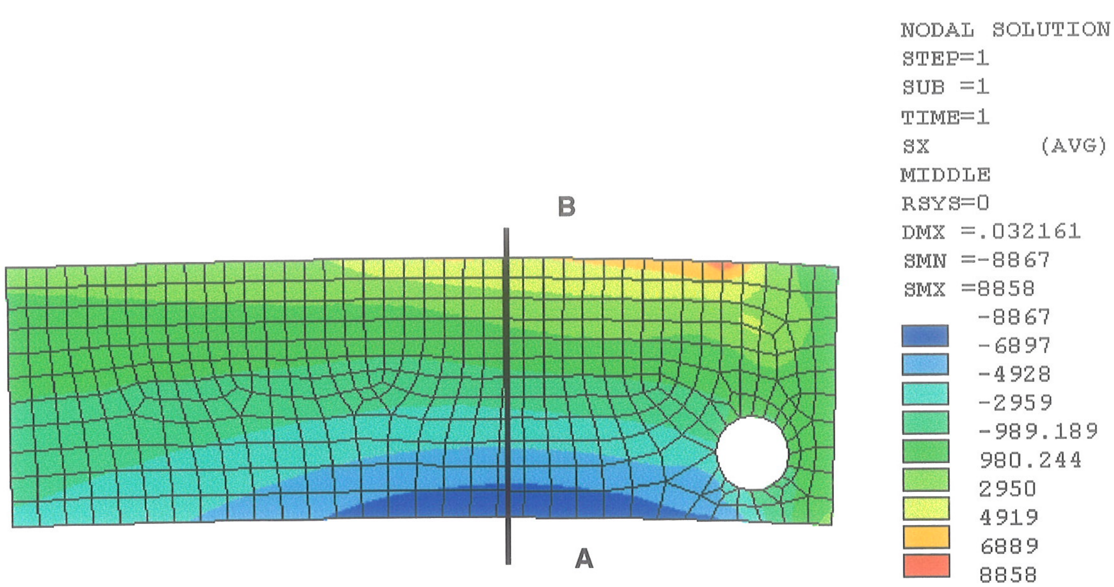


Figure 6. Bending Stresses in Side Beam

The highest stresses in the tube occur at the outer surface on which the pins bear. These stresses exceed 15.3 ksi in a region surrounding the pin hole, as shown in Fig. 7. (In the figure, all regions with stresses in excess of 15.3 ksi are gray). The maximum stress is 36 ksi, and occurs at the edge of the hole. These stresses around the hole (which is already reinforced with a 0.75 inch plate on the inside of the tube) are concentrations, and no upper limit is placed on them, since in statically loaded structures constructed of ductile materials, such concentrations have no mechanical significance. The region below the hole, directly in the load path of the pin, is less than 15.3 ksi. On this basis it can be stated that the pin/tube connection meets the $S_y/3$ criterion.

As an additional check, the nominal bearing stress can be calculated from the pin diameter and the total plate thickness upon which the pin rests. This thickness is 0.5 in (tube wall) + 0.75 in (internal bearing plate). With the pin diameter of 2.75 in, and a total load of 30000 lbs, the nominal bearing stress is $\sigma = 30000/((2.75)(1.25)) = 8.7$ ksi, well below the allowable stress of 15.3 ksi.

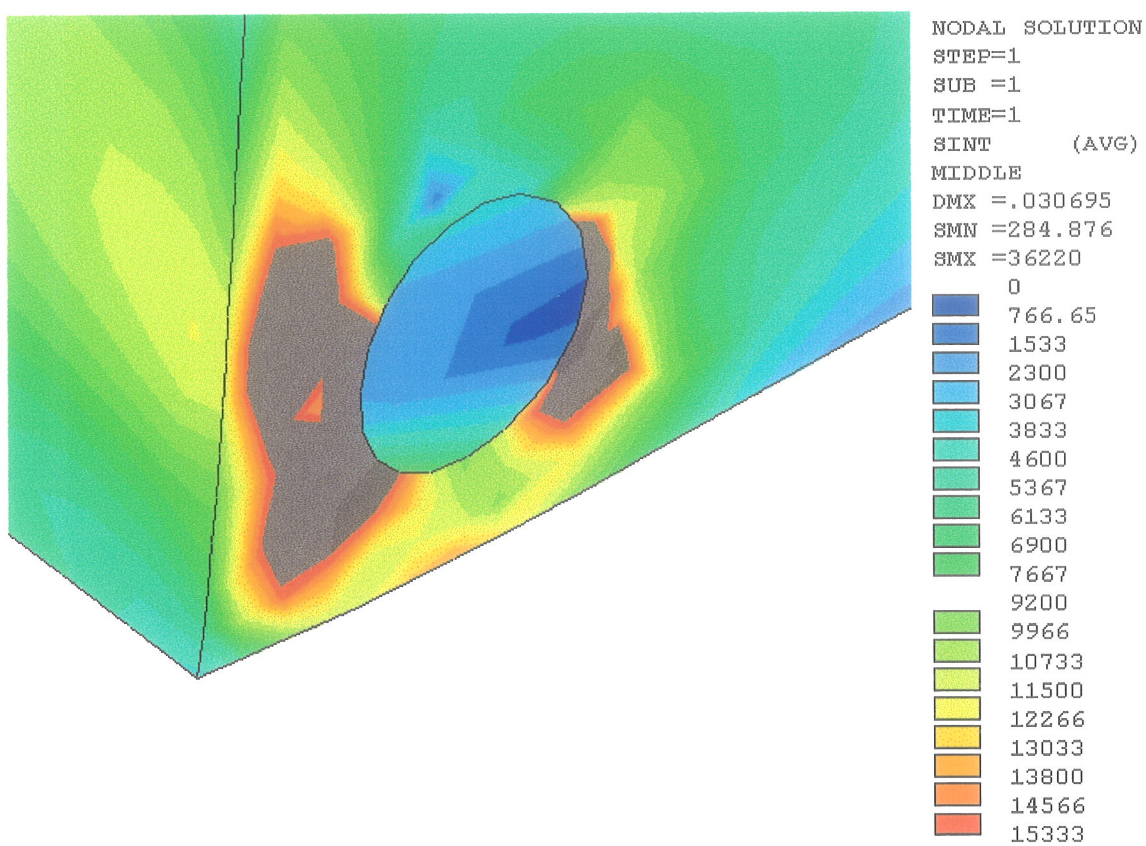


**Figure 7. Stress Intensity in Region of Pin Hole in Tube.
(stresses in excess of 15.3 ksi are gray)**

The possibility exists that the fixture could be used in a way that positions the moving beam at the center of the side tubes. This would put the maximum moment on the tubes. Fig. 8 shows the stresses in the side tube for a lifting force of 60 kips (i.e., rated crane capacity).

Along the line A-B in Fig. 8 the primary membrane plus bending stresses are 11.5 ksi, which is less than the allowable of 15.3 ksi.

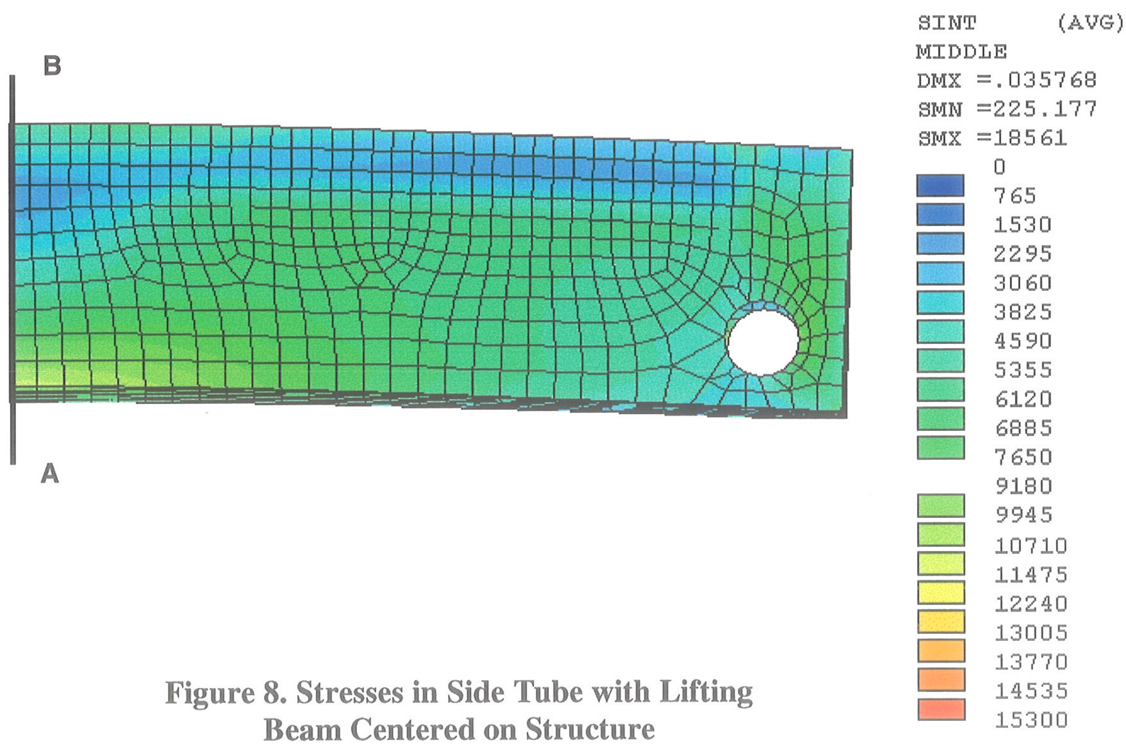


Figure 8. Stresses in Side Tube with Lifting Beam Centered on Structure

End Tube

Fig. 9 shows the stresses in the end tube nearest the lifting beam. No stress exceeds the 15.3 ksi allowable. This is verified by the linearized stresses across line A-B in Fig. 9. $P_m + P_b$ across this section is 11.1 ksi.

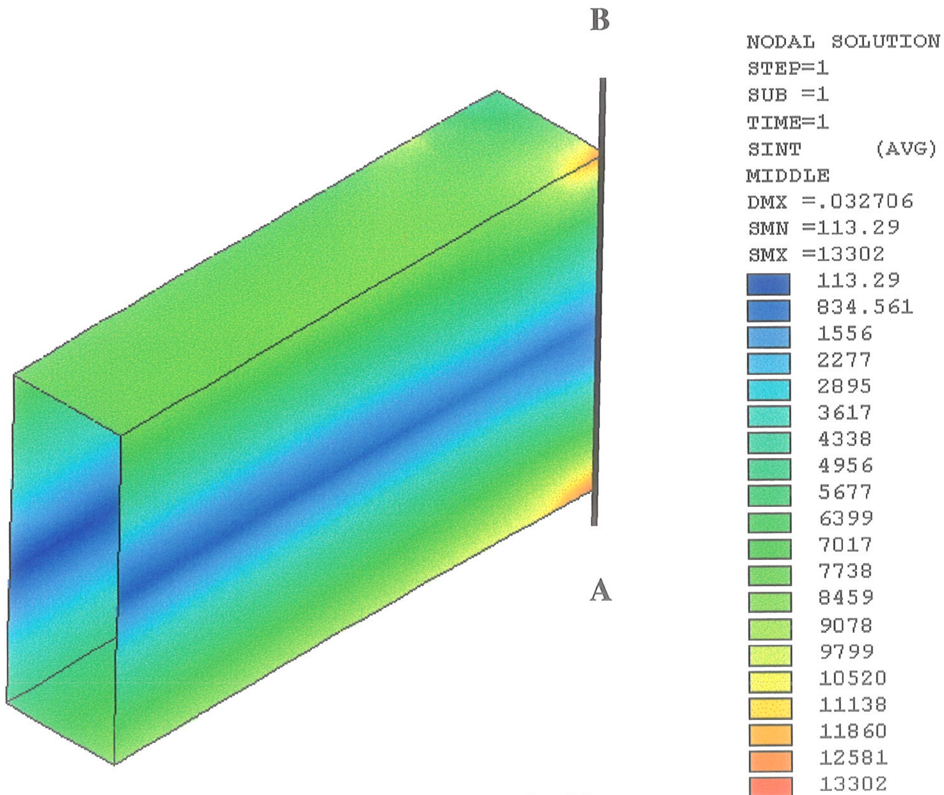


Figure 9. Stresses in End Tube Nearest Lifting Beam

Lifting Beam and Lifting Eye

Fig. 10 shows the stresses in the lifting beam. Stresses in the side plates are largest through sections near the lifting eye, and near the shelf upon which the rack sits. Both of these sections have been reinforced with 0.5 in. plate, giving a total steel thickness of 1 inch.

Fig. 11 shows the two most heavily stressed sections in the lifting beam. Along line A-B, the linearized stress is 9.5 ksi. Along line C-D, the linearized stress is 10.9 ksi. Both of these stresses are less than the allowable stress of 12 ksi.

The lifting eye profile is shown in Fig. 12. For a vertical load of 60000, the nominal stress on the section A-B is $\sigma = 30000/(3(2.88-1.16)) = 5.8$ ksi. The shear along section C-D is $\tau = 30000/(3)(2.64) = 2.8$ ksi. Both of these stresses are well below the normal and shear stress allowables for A36 steel.

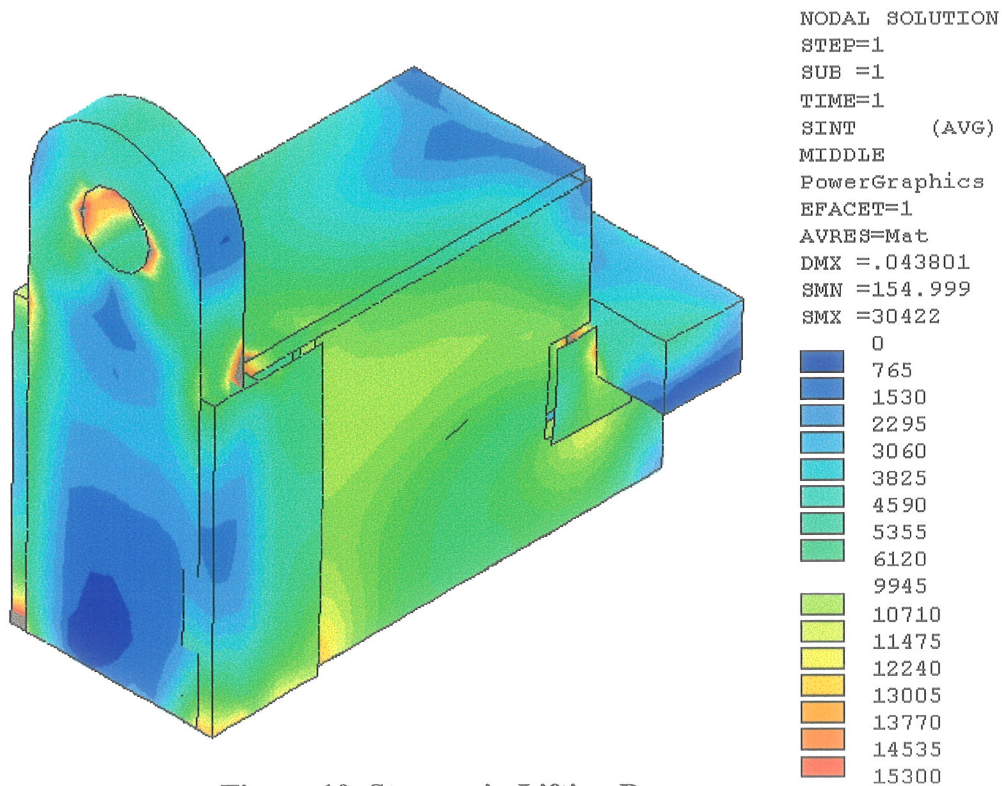


Figure 10. Stresses in Lifting Beam

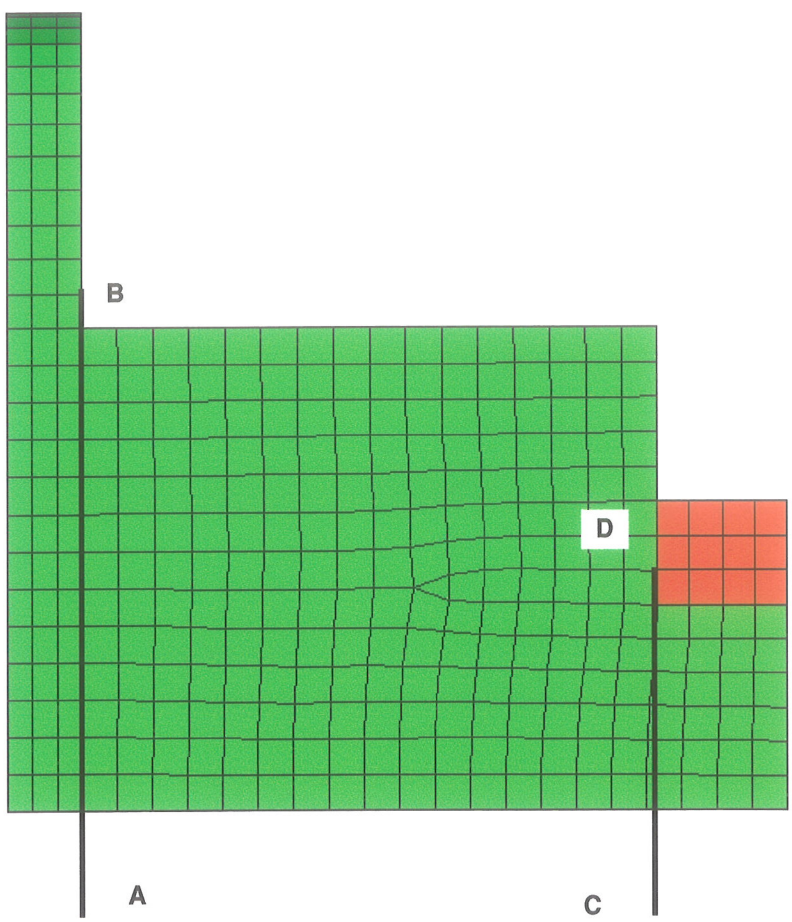


Figure 11. Lines for Stress Linearization

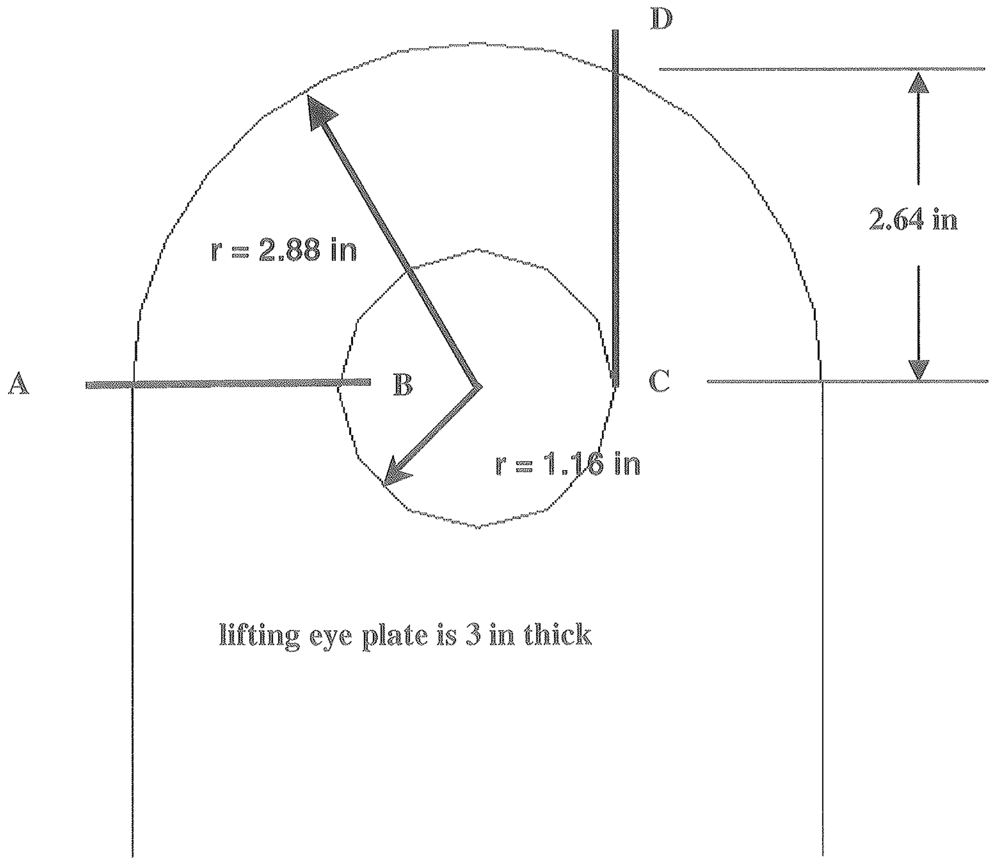


Figure 12. Lifting Eye Dimensions

Rack on Frame

Figure 13 shows the stresses in the rack that attaches to the frame. The stresses are highest near the end, where the rack welds to the end tube. The stresses indicated at this end weld are not accurate, since the finite element model used node-to-surface constraint equations to model the weld. This places all of the reaction forces onto a single line of nodes, with no regard for the actual weld geometry. The stresses in this end of the rack will be considered later in this report.

The stresses in the rack some distance away from the end are accurate, because the line-loading of the edge will redistribute through the rack volume. The line A-B of Fig. 14 is about 1.5 inches from the end of the rack. The linearized stresses across this section are 9.5 ksi, which is below the maximum allowable stress of 12 ksi for A36 steel.

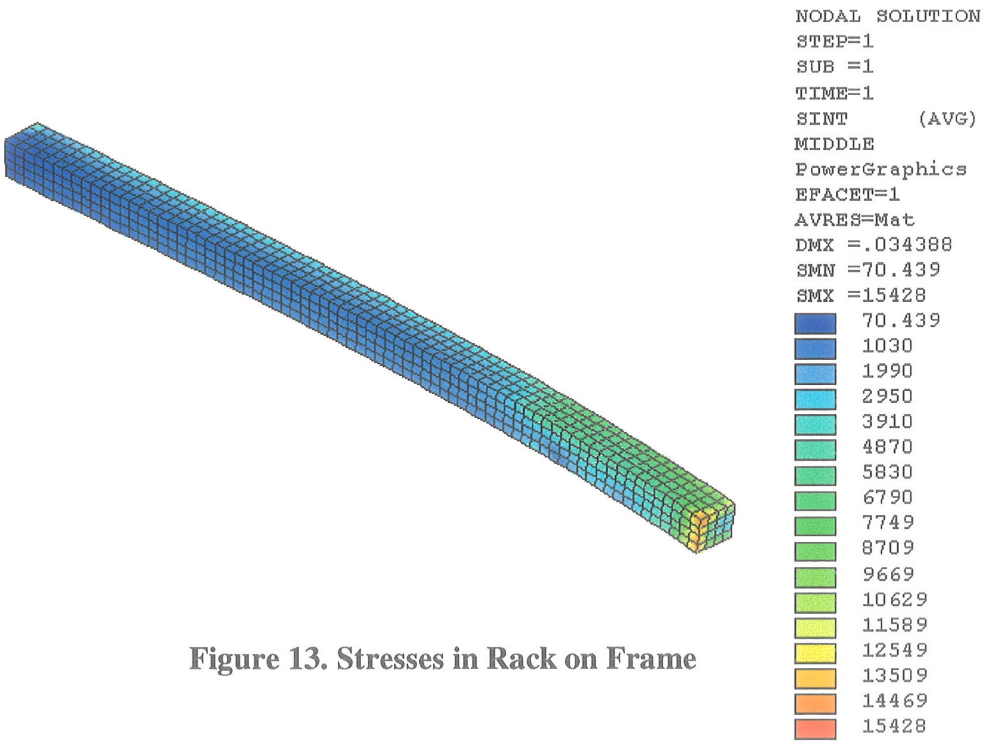


Figure 13. Stresses in Rack on Frame

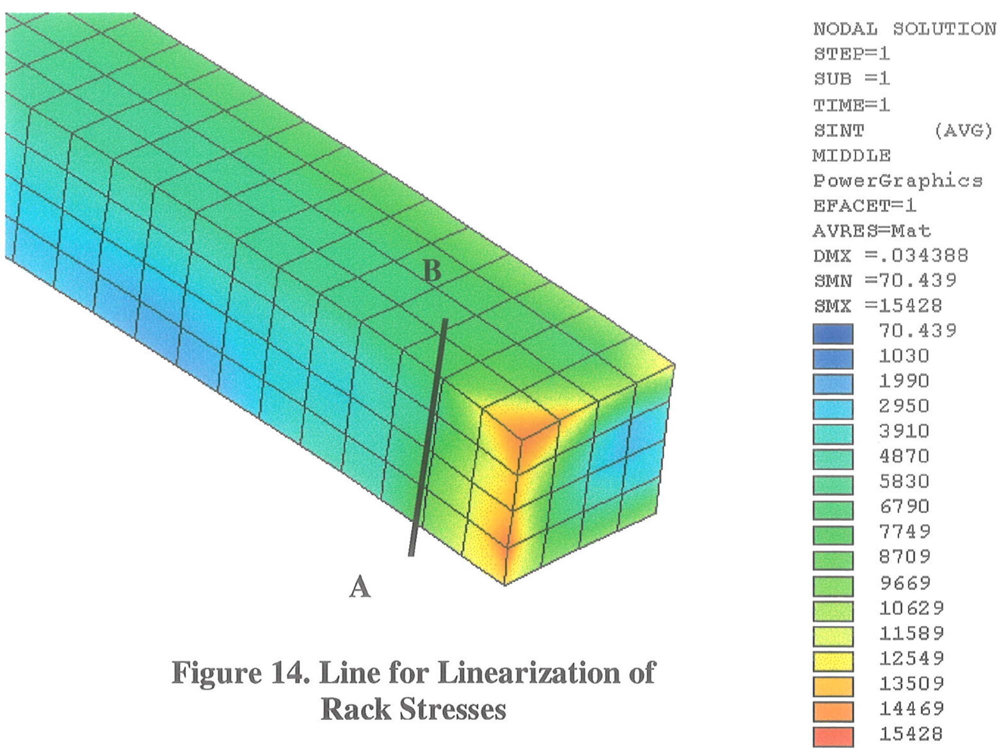


Figure 14. Line for Linearization of Rack Stresses

Rack on Lifting Beam

The stresses in the rack on the lifting beam are shown in Fig. 15. The two concentrations occur at the notch in the lifting beam plates against which the rack rests. Because this is a concentration (and unrealistic because of both line-contact, and mesh coarseness), the stress linearization section is chosen one element away (about 0.65 in) from the front vertical surface (line A-B of Fig. 15). The linearized stresses on this section are 9.2 ksi, below the 12 ksi allowable for A36 material.

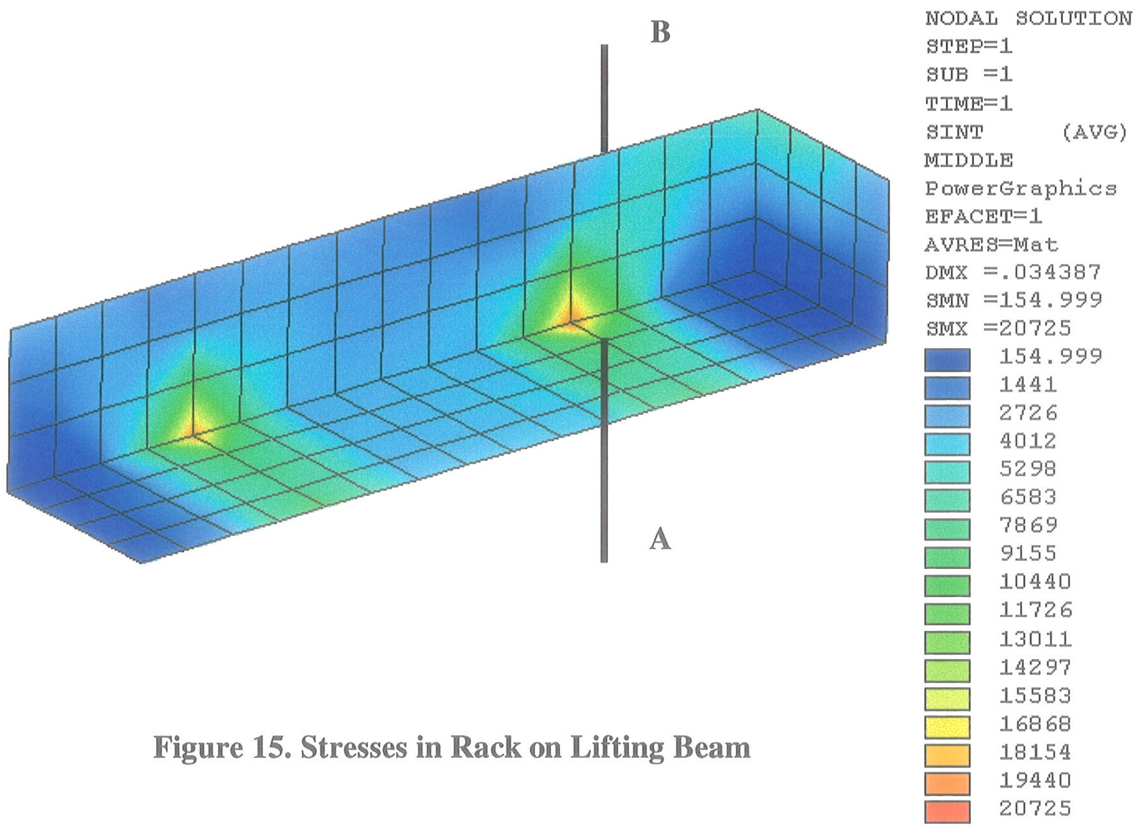


Figure 15. Stresses in Rack on Lifting Beam

Pins

The pins are sized according to the worst case engagement with the horn module hooks. Clearances of the parts show that if the frame is placed off center such that the hooks on the module on one side touch the tube surface, the clearance between tube and hook on the other side can be as large as 1.125 inches. This is shown in Fig. 16, the finite element model of the hook/pin/tube interaction. The deformation of the pin and hook moves the center of the hook force on the pin inward from the center of the hook depth. The effective lever arm implied by the deformation is calculated from the resulting moment at the pin/tube interface, and is 1.687 inches.

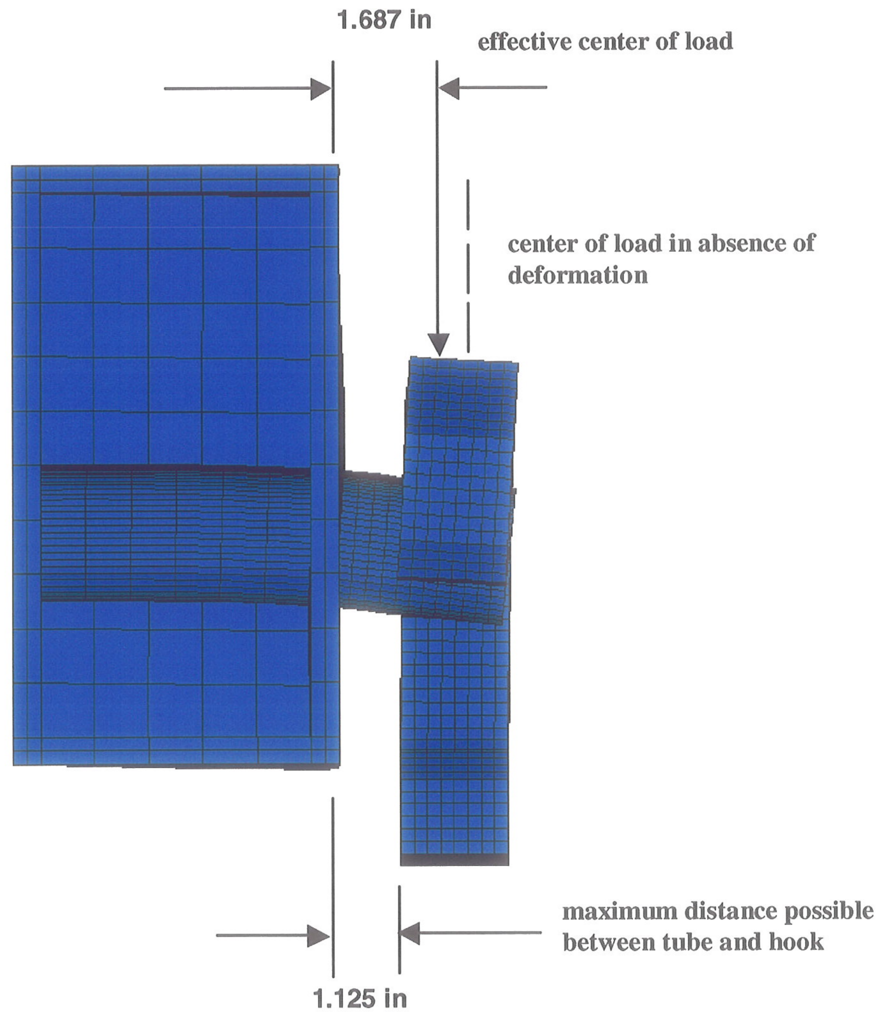


Figure 16. Deformation of Pin under Maximum Hook/Tube Clearance

The pin diameter is 2.75 inches. The moment of inertia for bending is $I = \pi D^4/64 = 2.8$ in^4. With a force of 30000 lbs, and a lever arm of 1.687 in., the applied moment is 50600 lb-in. The bending stress in the pin is then $\sigma_b = Mc/I = 50600(1.375)/2.8 = 24.8$ psi. This is well below the allowable of 29 ksi for the 4340 cold drawn material.

Nominal shear is $\tau = P/A = 30000/(\pi D^2/4) = 5$ ksi, well below the allowable of 14.5 ksi.

<u>Weld Between End Tube and Side Tube</u>

The weld between the end tube and side tube is a 3/8 in partial-penetration groove weld, The most highly stressed region is the vertical weld at the outer wall of the end tube, as shown in Fig. 9. This is because the 0.75 in plate which reinforces the end of the side tube works to stiffen the area substantially, causing the pin to transmit most of its load through this outer wall weld.

The stresses are analyzed by considering the forces on the nodes that comprise the weld. These are shown in Fig. 17. Nodes 3110 and 3115 are chosen for force summation and stress calculation. The length of the weld is taken as the distance between nodes 3110 and 3115, plus one half of the distances between nodes 3115 and 3116, and one half the distance between nodes 3110 and 3113.

The width of the weld is 0.375 inches. The length of weld is 1.86 inches. Then, the stress area is 0.698 in^2.

The Z-direction forces produce normal tensile stresses in the weld and base metal. Summing the Z-forces for nodes 3110 and 3115 give a total of 10600 lbs, for a total normal in both base metal and weld of 15.1 ksi. This is less than the allowable base metal stress of 15.3 ksi. It is about 8% greater than the allowable normal stress on the weld metal of 14 ksi. This difference is not significant, particularly in light of the conservative assumptions made in both working stresses and loads throughout the design process.

The X and Y direction forces produce shear on the weld and base metal. This shear force tends to concentrate at nodes 3110 and 3115, since these are closest to the load path from the pin through the 0.75 in plate in the end of the side tube. Taking the vector sum of the X and Y forces on these nodes gives a total shear force of 1222 lbs, for a shear stress of 1.3 ksi. This is far less than the allowable shear stress in the base metal of 7.65 ksi, and far less than the weld metal shear allowable of 14 ksi.

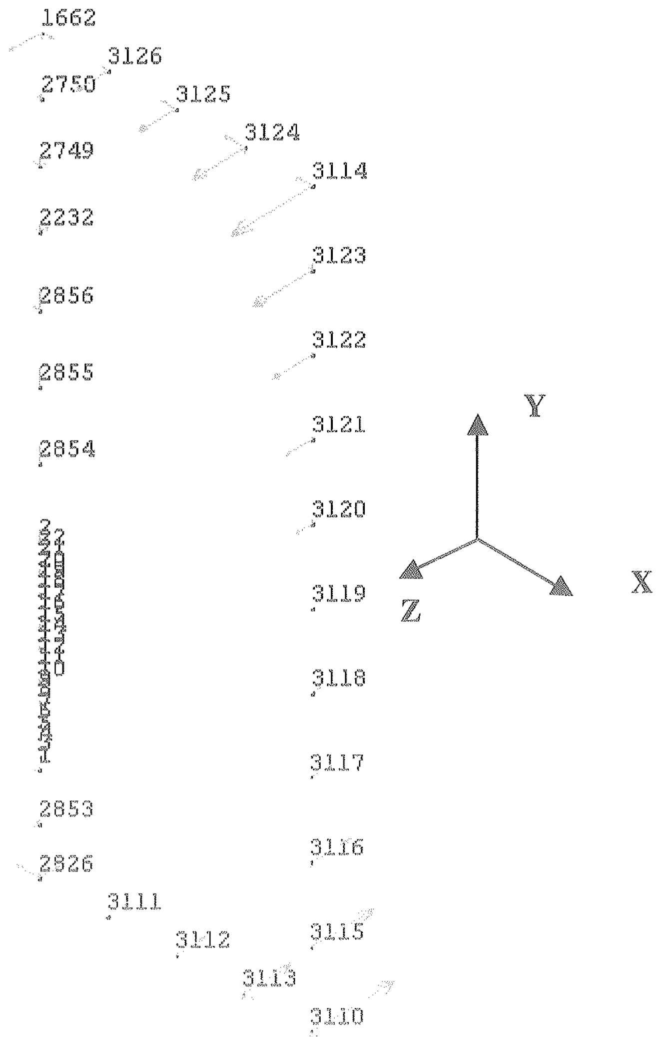


Figure 17. Nodes and Forces on Weld Between End Tube and Side Tube

Welds Between Rack and Side Tube

Fig. 18 shows the welds which attach the rack to the side tube. The most heavily stressed portion of weld is the 3-sided end weld along lines E-B, E-F, and F-D.

Fig. 19 shows the nodes and forces in the end weld. For each line, the forces for the appropriate set of nodes were summed. Because these are fillet welds, all weld stresses are considered to be shear. The base metal is subjected to both shear and normal stresses.

Table II summarizes the forces, stress areas, and stresses for each line of weld in Fig. 19.

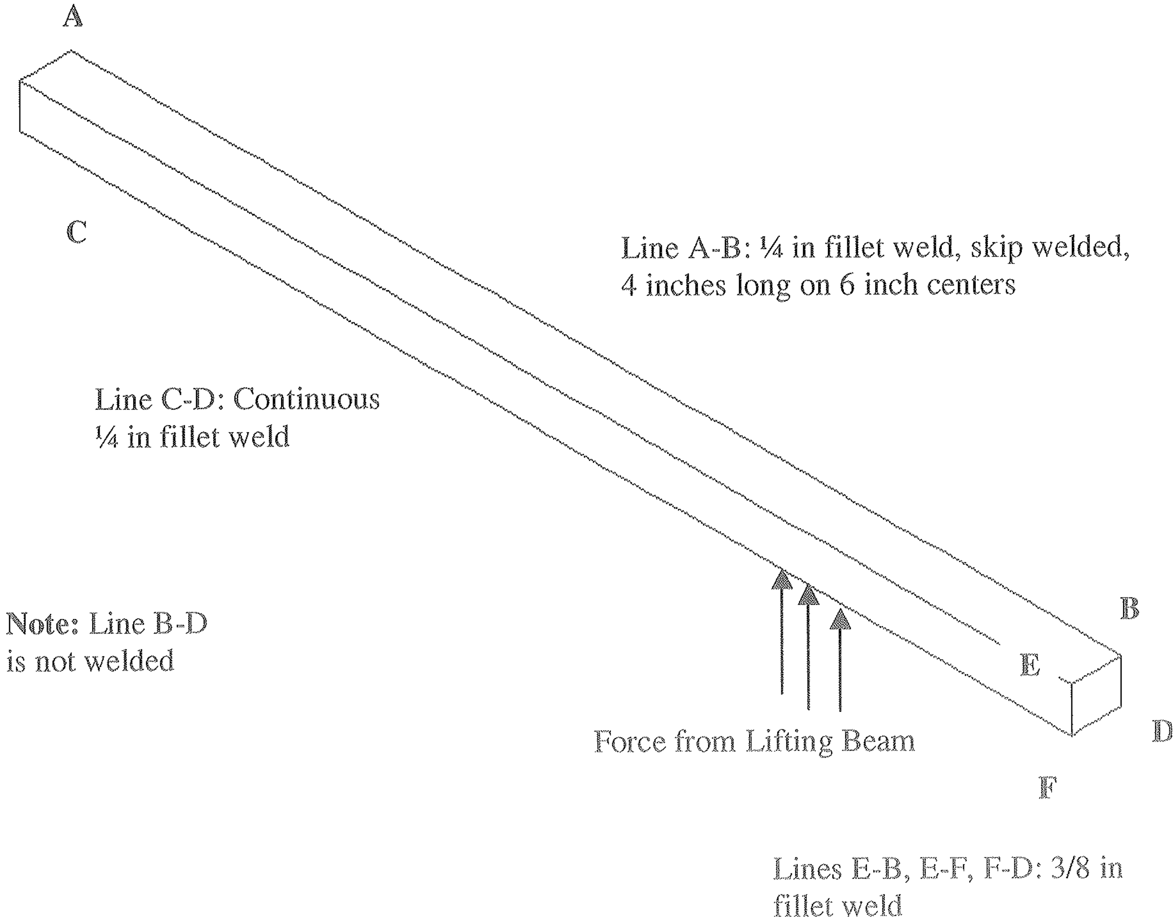


Figure 18. Welds Between Rack and Frame

Table II. Stresses on Weld Between Rack and End Tube

Weld	Weld Length (in)	Weld Stress Area (in^2)	Base Metal Stress Area (in^2)	Weld Shear Stress (ksi)	Base Metal Shear Stress (ksi)	Base Metal Normal Stress (ksi)
B-E	2.625	0.696	0.984	5.3	2.9	2.4
E-F	2.875	0.762	1.078	8.1	5.4	1.8
F-D	2.125	0.563	0.797	10.8	6.8	3.5

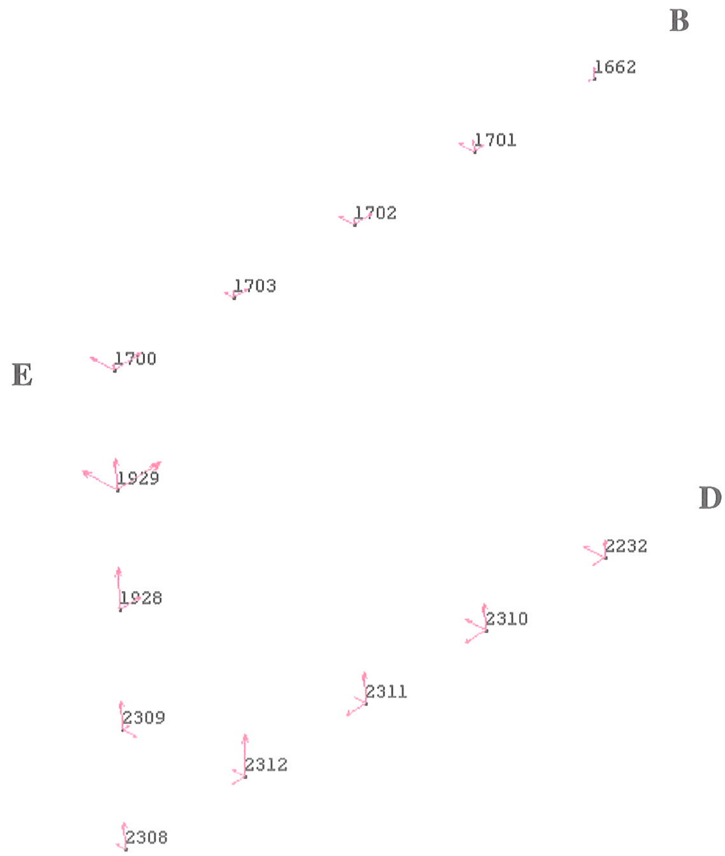


Figure 19. Nodal Forces on Welds along Lines E-B, E-F, F-D between Rack and End Tube

The welds between the rack and the side tube exhibit the expected concentration of force in the region of the lifting beam, as shown in Fig. 20.

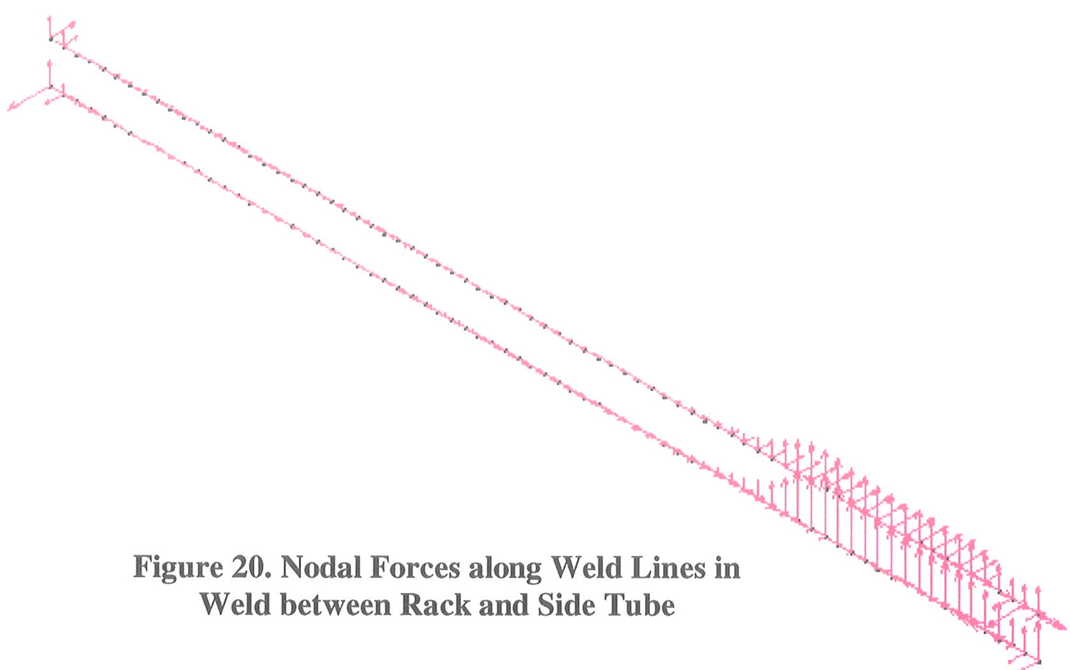


Figure 20. Nodal Forces along Weld Lines in Weld between Rack and Side Tube

The welds are analyzed by summing the forces on the nodes in the region of the lifting beam. These weld lines and corresponding nodal forces area shown in Fig. 21. The weld on these lines is a ¼ in fillet weld, so weld stresses are considered to be shear. Base metal stresses are both shear and normal stresses

Table III summarizes the stresses in the top and bottom rack-to-side tube welds.

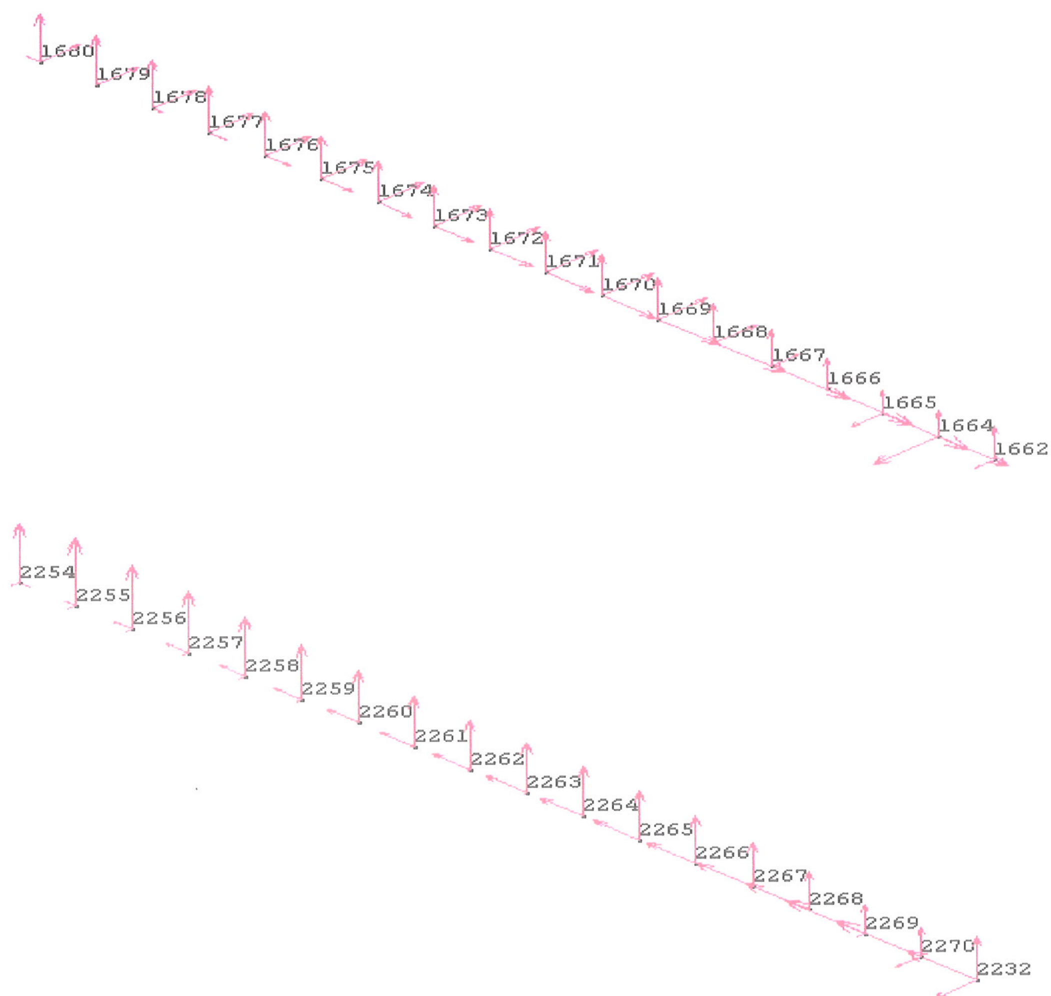


Figure 21. Nodal Forces on Top and Bottom Weld Lines between Rack and Side Tube

Table III. Summary of Stresses in Top and Bottom Welds between Rack and Side Tube

Weld	Weld Length (in)	Weld Stress Area (in^2)	Base Metal Stress Area (in^2)	Weld Shear Stress (ksi)	Base Metal Shear Stress (ksi)	Base Metal Normal Stress (ksi)
A-B	8	1.414	2	8.8	5.7	2.6
C-D	12	2.121	3	7.1	5.0	0.5

As the table shows, all stresses are well below the allowable stresses for the weld and tube.

Weld Between Lifting Eye and Lifting Beam

The lifting eye is welded to the side beam plates along all four edges with a 3/8 inch fillet weld. This is shown in Fig. 22. The total weld stress area is $(0.707)(0.375)(40) = 10.6\ in^2$. The fillet weld shear stress is then $\tau = 60000/10.6 = 5.7$ ksi, well below the allowable for both base metal shear stress and weld metal shear stress.

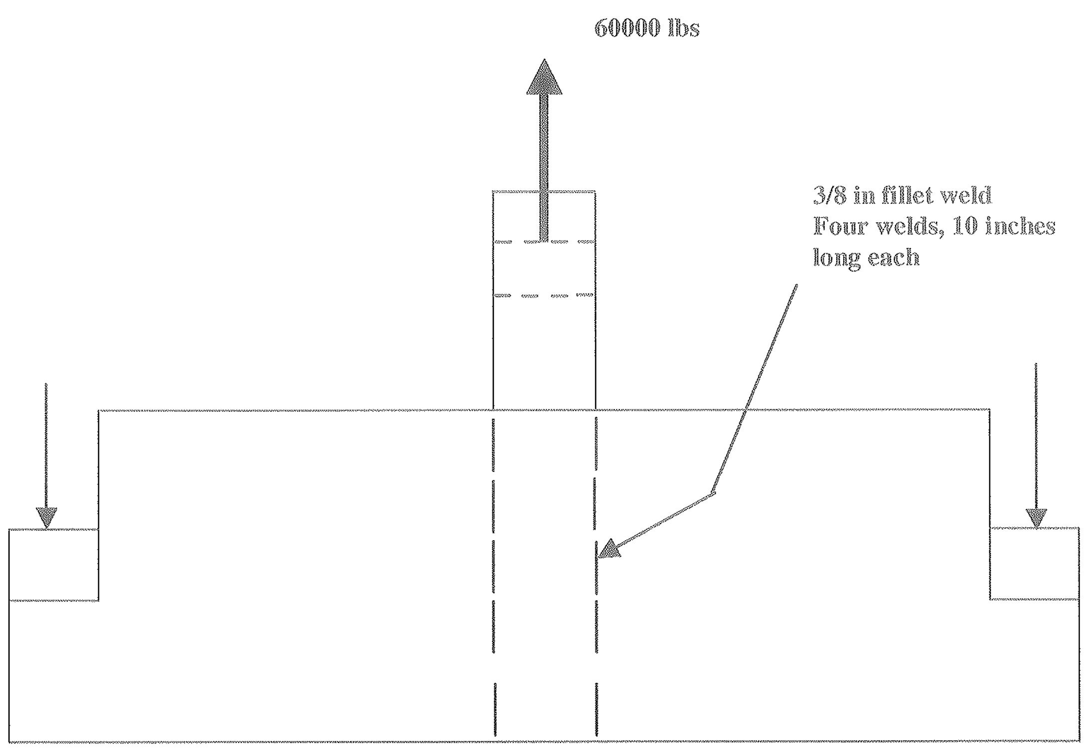


Figure 22. Welds Between Lifting Eye and Lifting Beam

March 12, 2002

Appendix A

Test Fixture Calculations

The basic test fixture geometry is shown in Drwg # 8875.000-ME-406690. There are two primary considerations for the fixture.

1. Strength of the 6x6x0.5 A500 Grade B steel tube from which the test load is suspended on the pin
2. Effects of center-drilling a ½ x 13 hole, 2 inches deep, in the center of the pin for the purpose of attaching a sling restraint.

The first concern is addressed by the model shown in Fig. 1.

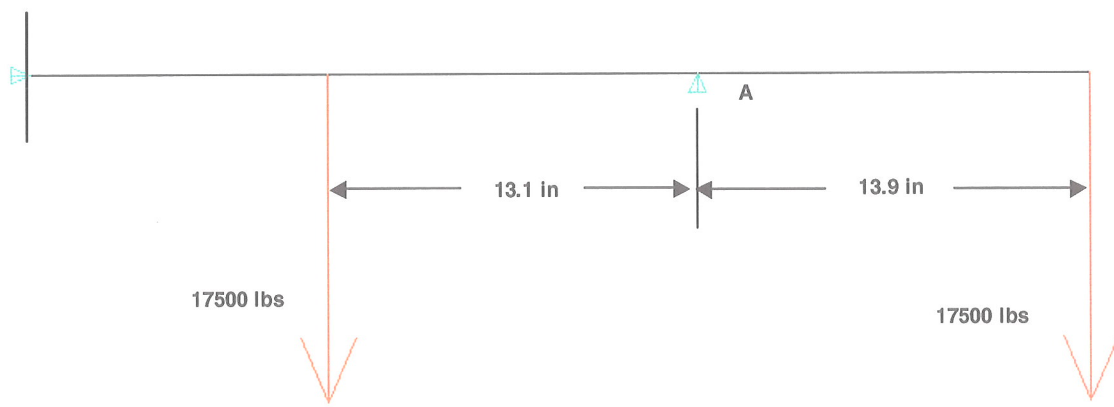


Figure 1. 6x6x0.5 Steel Tube under Load of 35000 lbs/pin (one-half symmetric)

The following are given:

Section Modulus = 16.8 in^3
Yield Stress = 46 ksi

Using a maximum load of 35000 lbs/pin, the moment about point A is

$$M = Pl = 17500(13.9)$$
$$M = 243250 \text{ lb-in}$$

The maximum bending stress in the beam is

$$\sigma = M/S = 243250/16.8$$
$$\sigma = 14500 \text{ psi}$$

The maximum allowable stress is taken as $S_y/3$ = 15333 psi. The bending stress of 14500 psi is less than the allowable stress of 15333 psi.

The second consideration, that of the effects of drilling a hole in the 4140 cold-drawn pin, was addressed by the FE model of Fig. 2.

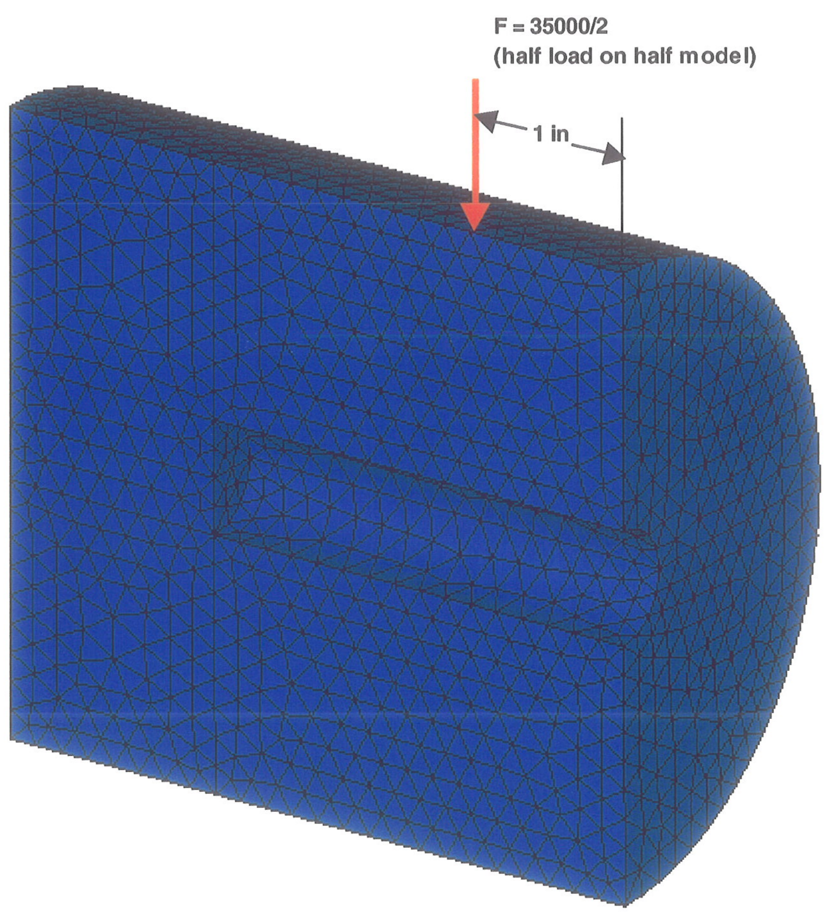


Figure 2. Finite Element Model of Pin w/Hole

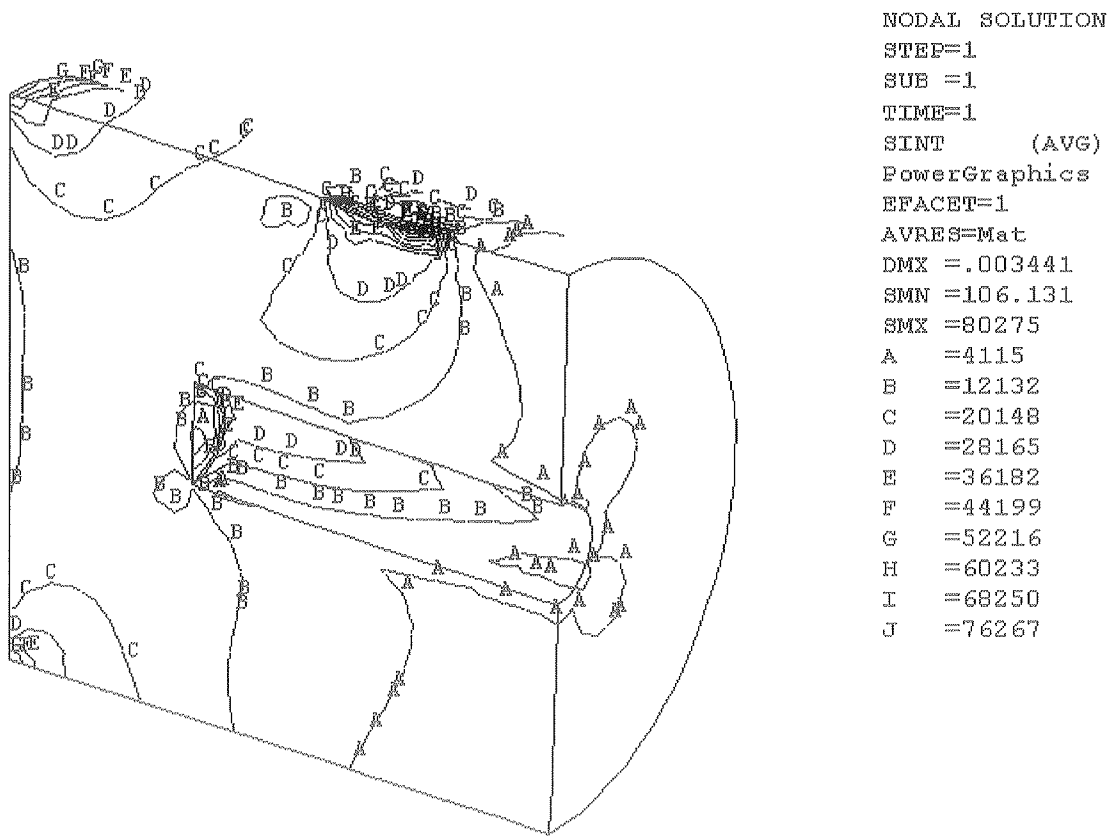


Figure 3. Stress Intensity in Pin w/Hole

Fig. 3 shows the stress intensity in the loaded pin. Stress concentrations are apparent at the base of the hole. However, the stress away from the base (contour D) is 28 ksi or less. This compares with an allowable stress intensity for primary stress of 29 ksi.

Fig. 4 shows the path used to evaluate the primary membrane plus bending stress intensity at the base of the pin. From ANSYS, across section A-B:

$P_m = 13.5$ ksi
$P_b = 19.4$ ksi
$P_m + P_b = 23.5$ ksi

The allowable pin stress is 29 ksi; therefore, the pin, with the hole, is adequate.

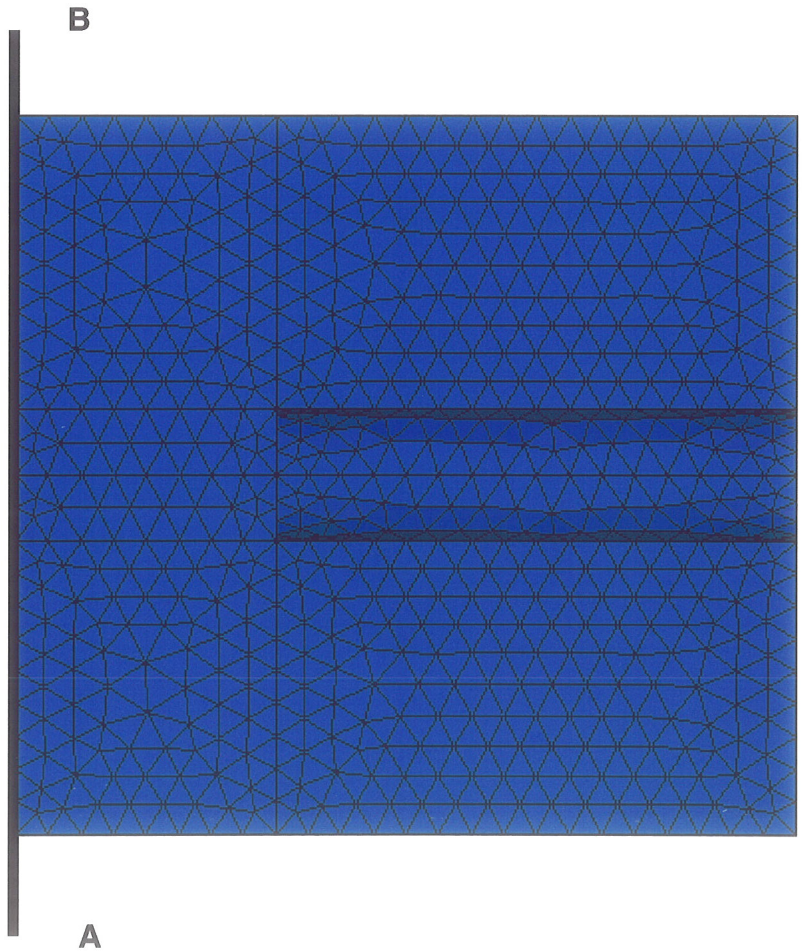


Figure 4. Path for Primary Stress Evaluation

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Batavia, IL. 60510

Date: March 18, 2002

We hereby certify that Fermi Lab's 35 Ton Anchor Shackle was proof tested to two (2) times the testing capacity of 40 tons.

JOHN SAKASH COMPANY

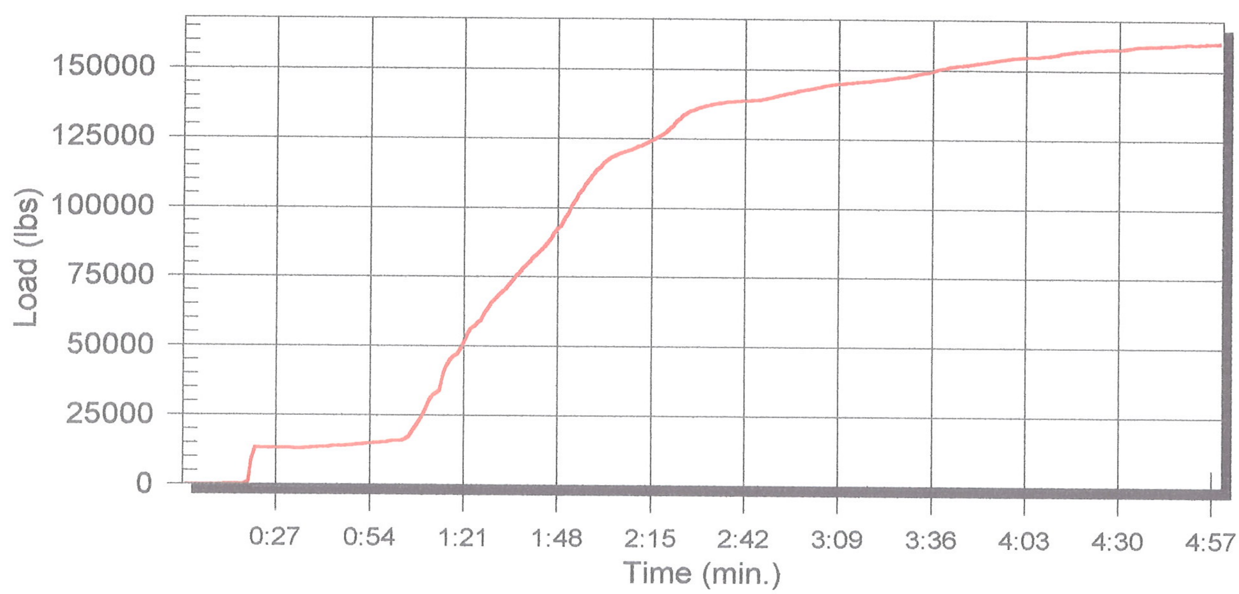
BY: ______________________

TEST FOR CROSBY 35 TON SHACKLE
#G-209-1018650
SHOWN ON DWG ME-406392

Certificate of Testing

John Sakash Company, Inc.

Customer:	FERMI LAB	Test Number:	1
		Test Date:	03/19/02
Serial No.:		Description:	SHACKLE
Order No.:	500258	Test Method:	PROOF = 2.00xWLL
Invoice No.:	PRN19288/DEBRA	WLL:	80,000 lbs.
Size:	2 in.	Peak Load:	160,040 lbs.
Length:	0 ft.	Test Duration:	5.0 Minutes



Test Results
- [x] Acceptable
- [ ] Not Acceptable

Conducted by: Hubert Staley

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