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7. Abstract This supporting document details calculations for the proper design of a lifting beam and redesigned lifting lugs for the 241AZ01A decant pump. This design is in accordance with <i>Standard Architectural-Civil Design Criteria, Design Loads for Facilites</i> (DOE-RL 1989) and is safety class three. The design and fabrication is in accordance with American Institute of Steel Construction, <i>Manual of Steel Construction</i>, (AISC, 1989) and the <i>Hanford Hoisting and Rigging Manual</i> (DOE-RL 1993).		
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ANALYSIS OF LIFTING BEAM AND REDESIGNED LIFTING LUGS FOR 241AZ01A DECANT PUMP

November 15, 1994

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Westinghouse Hanford Company
Hanford Operations and Engineering Contractor
for the
U. S. Department of Energy

DESIGN VERIFICATION METHOD

The need for design verification has been reviewed with the method selected as indicated below: (ESR/Work Plan # NA / WP-8D430-331).

<u> X </u>	Independent Review
<u> </u>	Alternate Calculations
<u> </u>	Qualification Testing
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R. B. Pan
Cognizant/Project/Design Manager

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CHECKLIST FOR INDEPENDENT REVIEW

Document Reviewed ANALYSIS OF LIFTING BEAM AND REDESIGNED LIFTING LUG FOR 241AZ01A DECANT PUMP

Author B. L. Coverdell

<u>Yes</u>	<u>No</u>	<u>N/A</u>	
☒	☐	☐	Problem completely defined.
☒	☐	☐	Necessary assumptions explicitly stated and supported.
☒	☐	☐	Computer codes and data files documented.
☒	☐	☐	Data used in calculations explicitly stated in document.
☒	☐	☐	Data checked for consistency with original source information as applicable.
☐	☐	☒	Mathematical derivations checked including dimensional consistency of results.
☒	☐	☐	Models appropriate and used within range of validity or use outside range of established validity justified.
☒	☐	☐	Hand calculations checked for errors.
☐	☐	☒	Code run streams correct and consistent with analysis documentation.
☒	☐	☐	Code output consistent with input and with results reported in analysis documentation.
☒	☐	☐	Acceptability limits on analytical results applicable and supported. Limits checked against sources.
☒	☐	☐	Safety margins consistent with good engineering practices.
☒	☐	☐	Conclusions consistent with analytical results and applicable limits.
☒	☐	☐	Results and conclusions address all points required in the problem statement.

MANDATORY

Software QA Log Number 94-070

J. S. Burgess JS Burgess
Reviewer

11/29/94
Date

ENGINEERING ANALYSIS SOFTWARE REPORT FORM

November 15, 1994
DATE ANALYSIS PERFORMED

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SOFTWARE APPLICATION: COSMOS/M ver 1.70.

B. L. Coverdell 11-30-94
B. L. Coverdell, Equipment Stress Analysis DATE
ANALYST

R. B. Pan 12/1/94
R. B. Pan, Equipment Stress Analysis DATE
MANAGER

DESCRIPTION OF ANALYSIS:

The use of the finite-element (FE) analysis program *COSMOS/M* version 1.70 (SRAC 1994) permitted a quick, easy, and detailed stress analysis of the following components of the 241AZ01A lifting beam: the top lug, the I-beam and gusset plates and bottom lugs. These analysis are called TOP_LUG02, LBEAM02 and BOTTOM_LUG01 respectively. Each of these analysis used triangular elements to determine the stresses developed by the 5,442 kg (6 ton) load.

The FE analysis was performed on a Silicon Graphics Incorporated, Indigo computer. Electronic copies of the input and output files can be found on the attached tape and on the Common File Storage (CFS) in the directory w81423/241az01a.

J. S. Burgess JS Burgess 11/30/94
REVIEWER, DATE

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1.0 INTRODUCTION

This analysis determines whether the 241AZ01A decant pump lifting beam design is adequate given the pump weight of 5,442 kg (12,000 lb). The analysis also determines the adequacy of the redesigned 241AZ01A lifting lugs given that the existing ones are inadequate due to clearance problems.

The 241AZ01A decant pump lifting beam was separated into two sections for analysis purposes. The top section was divided into three components, which are: the top lifting lug, the I-Beam and gusset plates, and the (4) lower lifting lugs. The lower section is also divided into three components: the connecting pin, the vertical bar, and the lifting hook. For more information on the lifting beam, reference drawing H-2-83761 Rev. 0 (WHC 1994a). A finite-element (FE) analysis was completed on the three components comprising the top section of the lifting beam. Hand calculations were used to determine the adequacy of the lower section of the lifting beam and the redesigned 241AZ01A lifting lugs. For more information on the redesigned 241AZ01A lifting lugs, reference drawing H-2-820775 Rev. 0 (WHC 1994b) and ECN# 704915 (WHC 1994c).

Both the lifting beam and redesigned 241AZ01A lifting lugs are in accordance with *Standard Architectural-Civil Design Criteria, Design Loads for Facilities* (DOE-RL 1993) and are safety class 3. The design and fabrication of each component is in accordance with American Institute of Steel Construction, *Manual of Steel Construction*, (AISC 1989) and the *Hanford Hoisting and Rigging Manual* (DOE-RL 1993).

2.0 SUMMARY OF RESULTS

The calculations in Appendix A determined that the maximum allowable bending, shear, and bearing stresses are 148.8 MPa (21,600 lb/in²), 148.8 MPa (21,600 lb/in²), and 223.2 MPa (32,400 lb/in²) respectively. The maximum allowable stresses were determined from the *Manual of Steel Construction* (AISC 1989). The maximum von Mises stresses for each of the three components comprising the top section of the lifting beam are plotted in Appendix B. The plots show the maximum von Mises stresses are 62.77 MPa (9,110 lb/in²) for the top lifting lug, 26.53 MPa (3,850 lb/in²) for the I-Beam and gusset plates, and 37.90 MPa (5,500 lb/in²) for the lower lifting lugs. The hand calculations in Appendix A determined that the maximum stress in the lower section of the lifting beam is 68.23 MPa (9,903 lb/in²) and occurs in the center of the pin connecting the lower lifting lug and the vertical bar.

Appendix A also contains calculations showing the maximum stress for the redesigned 241AZ01A lifting lug. The maximum stress for this component was determined to be 59.21 MPa (8,594 lb/in²) and occurs due to bending in the 5.08-cm (2-in) diameter pin.

3.0 DISCUSSION

The lifting beam and the redesigned 241AZ01A lifting lugs analyzed in this document are to be used for hoisting the 241AZ01A decant pump. The lifting beam was divided into two sections for analysis purposes. A FE analysis was performed on the three components comprising the top section. These components are:

- top lifting lug
- I-Beam and gusset plates
- lower lifting lug.

DOE-RL 1992 requires a factor of safety of three for all below-the-hook lifting devices. For each of the three FE models, the maximum von Mises stresses were obtained from the stress plots depicted in Appendix B. These maximum stresses then were compared to the allowable stress of 82.68 MPa (12,000 lb/in²). The maximum stresses for the top lug, I-Beam and gusset plates, and lower lifting lug are 62.77 MPa (9,110 lb/in²), 26.53 MPa (3,850 lb/in²) and 37.90 MPa (5,500 lb/in²), respectively.

The lower section of the lifting beam was analyzed using hand calculations. These calculations are shown in Appendix A. The components of the lower section are:

- pin connecting the lower lug and vertical bar
- vertical bar
- lifting hook.

The remaining components of the lifting beam were analyzed with hand calculations (Appendix A). The maximum stress in the lifting beam, found in the pin between the lower lug and the vertical bar, is 68.23 MPa (9,903 lb/in²).

Appendix A also contains calculations showing the maximum stress in the redesigned 241AZ01A lifting lug. The maximum stress is 59.21 MPa (8,594 lb/in²) and was found in the 5.08-cm (2-in.) diameter pin.

4.0 CONCLUSIONS

The calculations in Appendix A and the stress plots in Appendix B show that the lifting beam and the redesigned 241AZ01A lifting lug are adequate as shown in drawing H-2-83761 (WHC 1994a), drawing H-2-820775 (WHC 1994b) and in ECN# 704915 (WHC 1994c).

5.0 REQUIREMENTS

Below is a list of requirements that must be met for this lifting beam to conform to DOE-RL 1992:

- The drawing number, drawing revision number, weight and hoisting capacity must be painted on the lifting beam.
- The lifting beam must be welded and inspected per American Welding Society D1.1 (AWS 1994).
- The lifting beam must load tested to 125% of its rated capacity.

With the above requirements met, the lifting beam is adequate for hoisting of the 241AZ01A decant pump.

The only requirement for the redesigned 241AZ01A lifting lug is that it must be welded and inspected per AWS 1994. The redesigned 241AZ01A lifting lug need not be load tested since it is not considered a below the hook lifting device per DOE-RL 1992. The redesigned 241AZ01A lifting lug, in this case, is considered to be part of the object to be hoisted.

6.0 REFERENCES

- AISC, 1989, *Manual of Steel Construction*, Ninth Edition, American Institute of Steel Construction, Chicago, Illinois.
- AWS, 1994, *Structural Welding Code*, AWS D1.1, American Welding Society, Miami, Florida.
- DOE-RL, 1993, *Standard Architectural-Civil Design Criteria*, Design Loads for Facilities, DOE-RL-SDC 4.1 Revision 12, Westinghouse Hanford Company, Richland, Washington.
- DOE-RL, 1992, *Hanford Site Hoisting and Rigging Manual*, DOE-RL-92-36, U.S. Department of Energy, Richland Operations Office, Richland, Washington.
- SRAC, 1994, *COSMOS/M*, Structural Research and Analysis Corporation, Version 1.70, Santa Monica, California.
- WHC, 1994a, *Lifting Beam*, drawing H-2-83761 Rev. 0, Westinghouse Hanford Company, Richland, Washington.
- WHC, 1994b, *Piping 241-AZ-01A Decant Pump Adapter Flange*, drawing H-2-820775 Rev. 0, Westinghouse Hanford Company, Richland, Washington.
- WHC, 1994c, ECN# 704915, Westinghouse Hanford Company, Richland, Washington.

Appendix A
Calculations

Calculation Cover Sheet

Subject: ANALYSIS OF LIFTING BEAM AND REDESIGNED LIFTING LUG FOR 241AZ01A
DECANT PUMP

Originator: B. L. Coverdell 11-30-94
B. L. Coverdell, Advanced Engineer Date

Checker: J. S. Burgess 11/30/94
J. S. Burgess, Advanced Engineer Date

From the figure on the previous page and given that the maximum load on the pin will be less than or equal to 6000 lbf, the maximum shear and moment can be determined using the following equations. From the maximum shear and moment calculations the maximum shear bending stresses can be determined. The interaction of these two stresses must also be checked. The AISC Manual of Steel Construction was used to determine the allowable stresses. These allowables are shown below.

Allowable stress calculations for A36 CS.

$$F_y = 36000 \cdot \frac{\text{lbf}}{\text{in}^2} \quad \text{The yield stress for A36 CS}$$

$$F_t = .6 \cdot F_y \quad F_t = 21600 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_b = F_t \quad F_b = 21600 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_v = .6 \cdot F_y \quad F_v = 21600 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_{br} = .9 \cdot F_y \quad F_{br} = 32400 \cdot \frac{\text{lbf}}{\text{in}^2}$$

Stress calculations for the pin

$$P = 6000 \cdot \text{lbf} \cdot 1.25$$

The 1.25 is a 25 percent increase in the load to insure that the lifting beam will pass the 125 percent of rated load load test.

$$l = 3 \cdot \text{in}$$

$$d = 1.5 \cdot \text{in}$$

$$a = .25 \cdot \text{in}$$

$$b = 2.5 \cdot \text{in}$$

$$c = a$$

b is the area the force is distributed over. a and c are the distance on either side of the distributed force.

$$\omega = \frac{P}{b}$$

$$\omega = 3000 \cdot \frac{\text{lbf}}{\text{in}}$$

The distributed force.

$$A_c = \frac{\pi \cdot d^2}{4}$$

$$A_c = 1.767 \cdot \text{in}^2$$

Cross-sectional area of pin.

$$S = \frac{\pi \cdot d^3}{32}$$

$$S = 0.331 \cdot \text{in}^3$$

Section modulus of pin.

$$V = \frac{\omega \cdot b}{2 \cdot l} \cdot (2 \cdot a + b)$$

$$V = 3750 \cdot \text{lbf}$$

Maximum shear load on pin.

$$M = V \cdot a + \frac{V}{2 \cdot \omega}$$

$$M = 3281.25 \cdot \text{lbf} \cdot \text{in}$$

Maximum moment on pin

$$f_v = \frac{V}{A_c}$$

$$f_v = 2122.066 \cdot \frac{\text{lbf}}{\text{in}^2}$$

The pin will support the shear loading. OK

$$f_b = \frac{M}{S}$$

$$f_b = 9902.974 \cdot \frac{\text{lbf}}{\text{in}^2}$$

The pin will support the bending load.

Check interaction of the two stresses.

$$\frac{f_b}{F_b} + \frac{f_v}{F_v} = 0.557$$

Since the resultant is less than or equal to 1, a 1.375 in. diameter pin is adequate. OK

5.0 VERTICAL BAR STRESS CALCULATIONS

Determine the stresses in the bar given that each bar must support 6000 lbf. Refer to drawing H-2-83761.

5.1 LUG SHEAR TEAR OUT

$$d := 1.5625 \cdot \text{in} \quad w := 2.5 \cdot \text{in} \quad R := 1.5 \cdot \text{in} \quad \text{offset} := .5 \cdot \text{in}$$

$$A_c := R + \text{offset} - \frac{d}{2} \cdot w \cdot 2 \quad A_c = 6.094 \cdot \text{in}^2$$

$$f_v := \frac{P}{A_c} \quad f_v = 1230.769 \cdot \frac{\text{lbf}}{\text{in}^2}$$

Since the actual shear stress is less than allowable, the lug is adequate to resist shear tear out. OK

5.2 BEARING FAILURE

$$A_{br} := w \cdot \frac{d}{2} \quad A_{br} = 1.953 \cdot \text{in}^2$$

Area under bearing stress.

$$f_{br} := \frac{P}{A_{br}} \quad f_{br} = 3840 \cdot \frac{\text{lbf}}{\text{in}^2}$$

Since actual bearing is less than the allowable bearing, the bar will adequately support the load. OK

5.3 TENSION FAILURE

This section of calculations determines the remaining stressable area. It then calculates the tensile stress this area is under. Each side of the bar has a small but significant section of area removed.

$$a := 1.5 \cdot \text{in}$$

$$b := .25 \cdot \text{in}$$

$$c := 2 \cdot \sqrt{2 \cdot b \cdot R - b^2}$$

$$c = 1.658 \cdot \text{in}$$

$$\frac{c}{2} = 0.829 \cdot \text{in}$$

$$A_s := \int_{-\frac{c}{2}}^{\frac{c}{2}} \sqrt{x^2 + a^2} - 1.125 \cdot \text{in} \, dx$$

Integral defining the area between the two curves.

$$A_s = 0.743 \cdot \text{in}^2$$

Area due to one removed section (see the sketch above).

$$A_h := d \cdot w$$

$$A_h = 3.906 \cdot \text{in}^2$$

Cross-sectional area of the hole.

$$A_t := \pi \cdot R^2 - 2 \cdot A_s - A_h \quad A_t = 1.676 \cdot \text{in}^2$$

Total tensile area.

$$f_t := \frac{P}{A_t}$$

$$f_t = 4475.604 \cdot \frac{\text{lbf}}{\text{in}^2}$$

The actual tensile stress is less than allowable the bar will adequately support the tensile load. OK

5.4 TENSION FAILURE AT TOP OF BAR THREADS

Threads must be added to the vertical bar in order to attach a lifting hook. The reduction in cross-section area will cause an increase in stress. Determine this stress.

$$A_{\text{thread}} := \frac{\pi}{4} \cdot (2.26 \cdot \text{in})^2$$

$$A_{\text{thread}} = 4.011 \cdot \text{in}^2$$

Cross-sectional area removed due to the lifting hook threads.

$$A_{\text{bar}} := \frac{\pi}{4} \cdot (3 \cdot \text{in})^2$$

$$A_{\text{bar}} = 7.069 \cdot \text{in}^2$$

Total cross-sectional area of the 3 in. diameter bar.

$$A_t := A_{\text{bar}} - A_{\text{thread}}$$

$$A_t = 3.057 \cdot \text{in}^2$$

$$f_t := \frac{P}{A_t}$$

$$f_t = 2453.318 \cdot \frac{\text{lbf}}{\text{in}^2}$$

OK

5.5 THREAD STRESS CALCULATIONS

Threads matching the hook design from drawing H-2-99569 Rev. 4 are used on the vertical bar, therefore, the threads on the vertical bar are adequate.

6.0 LIFTING HOOK CALCULATIONS

The 15 ton lifting hook design was obtained from a Westinghouse Hanford drawing (H-2-99569). Therefore, the hook is adequate for use in this design.

7.0 WELD CALCULATIONS

The welds connecting the top lug and bottom lugs to the I-beam are a prequalified complete penetration welds per AISC 1992, page 4-164. OK.

8.0 DETERMINE THE ADEQUACY OF THE LUGS

Determine if the lifting lug shown below is adequate to support the 12,000 lb load of the 241AZ01A decant pump. The analysis must conform to the *Hanford Hoisting and Rigging Manual* (a factor of safety of 3 must be used). The *AISC Manual of Steel Construction* was used to determine the allowables for such things as bearing stress. The plate is 1/2" thick and the pin is 2" bar. For further information reference drawing H-2-820775.

Allowable Stress Calculations for A36 CS

$$F_y = 36000 \cdot \frac{\text{lbf}}{\text{in}^2}$$

Yield stress for A36 CS.

$$F_t = \frac{F_y}{3}$$

$$F_t = 12000 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_t = \frac{F_y}{3}$$

$$F_t = 12000 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_b = F_t$$

$$F_b = 12000 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_v = \frac{F_y}{3}$$

$$F_v = 12000 \cdot \frac{\text{lbf}}{\text{in}^2}$$

$$F_{br} = .9 \cdot F_y$$

$$F_{br} = 32400 \cdot \frac{\text{lbf}}{\text{in}^2}$$

AISC Manual of Steel Construction

8.1 SHEAR TEAROUT

Determine if the lifting lug will fail due to shear tearout as shown in the sketch below. Since there are two lugs assume that one lug can carry 2/3 of the load to be conservative.

$$P := 12000 \cdot \text{lbf}$$

$$P_{\text{lug}} := \frac{\frac{2}{3} P}{2}$$

$$P_{\text{lug}} = 9000 \cdot \text{lbf}$$

Each lug is constructed of two plates, assume that one plate can carry 2/3 of the lug load

$$P_{\text{plate}} := \frac{\frac{P_{\text{lug}}}{2}}{2}$$

$$P_{\text{plate}} = 6750 \cdot \text{lbf}$$

$$A_v := 2 \cdot 1 \cdot \text{in}^2$$

$$A_v = 2 \cdot \text{in}^2$$

The minimum area in shear.

$$f_v := \frac{P_{\text{plate}}}{A_v}$$

$$f_v = 3375 \cdot \frac{\text{lbf}}{\text{in}^2}$$

This calculation shows that shear tearout is not a factor in this design. OK

8.2 BEARING FAILURE

Determine the bearing stress on the 2 1/16" diameter hole.

$$t := 0.5 \cdot \text{in}$$

Plate thickness.

$$A_{\text{br}} := 2.0625 \cdot \text{in} \cdot t$$

Projected area of hole in bearing.

$$f_{\text{br}} := \frac{P_{\text{plate}}}{A_{\text{br}}}$$

The actual bearing stress is less than the allowable bearing stress. OK

8.3 TENSION FAILURE

Given the tension failure shown below and the sketch on the previous page, determine the tension stress.

$$A_t := 2.1 \cdot \text{in} \cdot t$$

$$A_t = 1 \cdot \text{in}^2$$

Area under tensile loading.

$$f_t := \frac{P_{\text{plate}}}{A_t}$$

$$f_t = 6750 \cdot \frac{\text{lbf}}{\text{in}^2}$$

The actual tensile loading is less than the allowable. OK

9.0 PIN STRESS CALCULATIONS

Given the 2" diameter pin shown in the sketch on page A-1, determine if it is adequate to support the load on the lug.

$$l := 3 \cdot \text{in}$$

$$d := 2 \cdot \text{in}$$

Obtained from the sketch on page A-1 and the drawing H-2-820775.

$$A_c := \frac{\pi \cdot d^2}{4}$$

$$A_c = 3.142 \cdot \text{in}^2$$

Cross-sectional area of the pin.

$$S := \frac{\pi \cdot d^3}{32}$$

$$S = 0.785 \cdot \text{in}^3$$

Section modulus of the pin.

$$V := \frac{P_{\text{lug}}}{2}$$

$$V = 4500 \cdot \text{lbf}$$

Maximum shear load on the pin.

$$M := \frac{P_{\text{lug}} \cdot l}{4}$$

$$M = 6750 \cdot \text{in} \cdot \text{lbf}$$

Maximum moment on the pin.

$$f_v := \frac{V}{A_c}$$

$$f_v = 1432.394 \cdot \frac{\text{lbf}}{\text{in}^2}$$

The actual shear load is less than the allowable. OK

$$f_b := \frac{M}{S}$$

$$f_b = 8594.367 \cdot \frac{\text{lbf}}{\text{in}^2}$$

The actual moment is less than the allowable. OK

Check interaction of the shear and moment stresses.

$$\frac{f_b}{F_b} + \frac{f_v}{F_v} = 0.836$$

Since the interaction of the two stresses is less than or equal to 1 the pin is adequate per AISC. OK

10.0 WELD CALCULATIONS

10.1 1/8" PIN WELD

Due to the configuration of the lug the only possible load the welds between the pin and the plate may see is a tensile load. This could only happen if the hook were to slip to one end of the pin wrapping around the plate and pulling on the pin. The weld is a 1/8" fillet weld.

$$A_w := \pi \cdot d$$

$$A_w = 6.283 \cdot \text{in} \quad \text{Linear area of the weld.}$$

$$f_r := \frac{P_{\text{lug}}}{A_w}$$

$$f_r = 1432.394 \cdot \frac{\text{lbf}}{\text{in}} \quad \text{Linear force on weld.}$$

$$f_a := 36000 \cdot \frac{\text{lbf}}{\text{in}^2} \quad \text{Allowable force on weld.}$$

$$w_r := \frac{f_r}{.707 \cdot f_a}$$

$$w_r = 0.056 \cdot \text{in} \quad \text{Required weld size.}$$

$$w_a := .125 \cdot \text{in}$$

$$FS := \frac{w_a}{w_r} \quad FS = 2.221$$

The actual weld size is less than the required weld size. OK

10.2 3/16" LUG ATTACHMENT WELD

Determine the stress in the rectangular weld shown below if it is treated as a line.

$$t_{\text{plate}} := .5 \cdot \text{in} \quad \text{Thickness of plate.}$$

$$w_{\text{plate}} := 8 \cdot \text{in}$$

$$A_w := 2 \cdot t_{\text{plate}} + w_{\text{plate}}$$

$$A_w = 17 \cdot \text{in} \quad \text{Linear area of the weld.}$$

$$f_r := \frac{P_{\text{lug}}}{A_w}$$

$$f_r = 529.412 \cdot \frac{\text{lbf}}{\text{in}} \quad \text{Linear force on weld.}$$

$$f_a := 36000 \cdot \frac{\text{lbf}}{\text{in}^2} \quad \text{Allowable force on weld.}$$

$$w_r := \frac{f_r}{.707 \cdot f_a} \quad w_r = 0.021 \cdot \text{in} \quad \text{Required weld size. size.}$$

$$w_a := .1875 \cdot \text{in}$$

$$w_a = 0.187 \cdot \text{in}$$

The actual weld size.

$$FS := \frac{w_a}{w_r}$$

$$FS = 9.014$$

The actual weld size is less than the required weld size. OK

Appendix B

von Mises stress plots of the lifting beam
components: Top Lifting Lug, I-Beam and Gusset Plate,
and Lower Lifting Lug

Lin STRESS Lc=1

Von Mises

9.11E+03

7.97E+03

6.84E+03

5.70E+03

4.56E+03

3.42E+03

2.29E+03

E+03

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COSMOS/M

Version : V1.70

TOP_LUG02

11-16-94

WHC-SB-WM-DA-179
Rev. 0

L1n STRESS Lc=1

Von Mises

COSMOS/M

Version : V1.70

BOTTOM LUG01

11-18-94

8-4