SLSF LOOP HANDLING SYSTEM

Volume I
Structural Analysis

by

H. Ahmed, A. Cowie, and D. Ma

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ABSTRACT

SLSF loop handling system was analyzed for deadweight and postulated dynamic loading conditions, identified in Chapters II and III in Volume I of this report, using a linear elastic static equivalent method of stress analysis. Stress analysis of the loop handling machine is presented in Volume I of this report. Chapter VII in Volume I of this report is a contribution by EG & G Co., who performed the work under ANL supervision. Stress analysis of SLSF-LHS cradle structure is performed in Volume II and Volume III of this report; Volume II consists of computer modeling of LHM-Cradle-transporter system and Volume III consists of stress computations of Cradle and critical attachments per AISC Code guidelines. HFEF is credited with in-depth review of initial phase of work in Volume III of this report. A summary of results with conclusions and recommendations is presented in Chapter XV in volume III of this report. Volume IV of this report contains some relevant backup appendices for the report.

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SLSF Loop Handling System Cradle Analysis
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I. INTRODUCTION

I.1 Summary Description of Loop Handling System

The loop handling system (LHS) provides the capability for handling and transporting sodium loop safety facility (SLSF) in-pile loops at the Idaho National Engineering Laboratory (INEL). Handling of an assembled, sodium filled in-pile loop within a facility is accomplished with the loop handling machine (LHM) in conjunction with the facility crane. Transport of the loop between facilities is accomplished using the transporter system (tractor-trailer rig) with the LHM serving as the container for the loop (See Fig. I.1-1).

The LHM is basically a shielded cylinder with an integral hoist. With all appurtenances, the LHM weighs 45.9 tons. It is approximately 30 feet long and has a nominal main body diameter of 30 inches. Depleted uranium is used as the shielding material for the LHM.

The LHM is completely sealed except for the lower end which is provided with a valve-like shield assembly. This shield valve assembly may be open or closed to permit entry and discharge of the in-pile loop from the shielded cavity in the LHM. An additional sealing flange is added to the gate assembly to permit sealing of the LHM.

Lateral supports to support the in-pile loop in the LHM when the LHM is loaded on the transporter and lowered to the 10° near horizontal position are provided at three locations. Penetrating the solid stainless steel flanges just above the bottom gate assembly and again approximately 12 feet up at the mid-flange are two systems of three motorized screw jacks 120° apart. The third support is achieved at the top of the LHM when the grapple enters a close clearance sleeve in the LHM cavity.

Electrical power for the loop annular linear induction pump (ALIP) and test train instrumentation from selected thermocouples are supplied to the loop through the LHM grapple. Connectors in the lower region of the grapple connect with the loop whenever the grapple is attached to the loop. Other connectors are provided at the top of the grapple for connection to either removable bulkhead fittings or cable spools. The cable spools provide continuous electrical and instrumentation hook-ups whenever the grapple with loop is being raised or lowered. The bulkhead fittings would provide the same hook-ups whenever the loop is fully up and being transported from facility to facility.

Fig. I.1-1 SLSF LOOP HANDLING SYSTEM

During transport between facilities, the LHM rests at approximately a 10° angle in a cradle which is mounted on a special trailer. The cradle pivots near its lower end and is raised and lowered by two double acting hydraulic cylinders. At the rear of the trailer is the hydraulic power pack and hydraulic control console. Space has been provided for future installation of a diesel driven MG-set for ALIP power.

I.2 Loop Handling Machine (Fig. I.2-1)

The LHM performs a double service in that it provides the "package" for shipping radioactive material (a loop containing irradiated fuel), as defined by ERDA Manual 0529, and it is both a hoisted and hoisting device involving critical components as defined by RDT Standard F8-6T. Where similar requirements, but quantitatively different limitations are invoked by these documents, the more restrictive has been used in the design of the LHM.

The main body assembly provides the basic "package" for radioactive shipments as well as the framework for the hoisting system. The main body is essentially a shielded hollow cylinder with a 20 inch inside diameter and an internal length of about 28 feet. It has two sections joined by a bolted flanged connection. The lower section has an outside diameter of 29-3/4 inches and the upper is 27 inches. Depleted uranium, cast and machined into hollow cylinders and welded together, occupies the annular space between the outer and inner stainless steel shells.

Mounted at the lower end of the main body is a non-sealing shield valve assembly. The shield valve gate contains a 24 inch diameter by 2 inch thick piece of depleted uranium which serves as the basic shield component through the bottom of the LHM. The gate is supported by ball-type drawer slides and can be positioned in the valve body by a screw and nut drive system. A commercial valve operator which contains limit and torque switches is used as the prime mover for the shield gate. A recess is provided in the lower face of shield valve body for a seal plate. With the seal plate installed, the LHM becomes a hermetic chamber.

Penetrating the top and bottom flanges of the lower body section are two systems of motorized screw jacks which function as the upper and lower loop supports. The upper loop supports rigidly secure the loop when the LHM is being handled or transported. The lower loop supports close to within 0.032 inch of the loop and provide support only when the loop deflects

sufficiently to contact these, (normally in the transport position). These supports are basically keyed acme screws which are driven by commercial valve operators similar to that used on the shield valve assembly. Each support system consists of three powered screw jacks spaced at 120° intervals around the periphery of the LHM. The upper supports travel approximately 1 inch while the lower supports travel about 7 inches. The valve operators powering the loop support mechanisms contain an override and handwheel for manual operation, if required.

The top of the LHM is closed by a dome assembly which contains a limit switch for directing grapple to the hoist, ALIP power and instrumentation bulkhead fittings. A removable dome cover provides the upper sealing boundary on the LHM. The dome cover must be removed when power and instrument cable spools are attached to the grapple.

Affixed to the side of the lower main body assembly is the grapple hoist mechanism. The grapple hoist provides its hoisting capability through two triple strand roller chains which are routed to the top of the LHM where they are connected to the grapple. At the hoist mechanism, the chains are routed from the grapple under individual idler sprockets which in turn are anchored to load cells external to the hoist housing. The load cells are in turn anchored under the hoist housing with fasteners which incorporate stacked belleville springs to provide a measure of impact protection for the hoist. These same fasteners are adjustable to permit minor compensation for differences in chain lengths. The chains extend from the load cell idlers over the drive sprockets and on to constant torque spring-loaded take-up drums. The drive load line includes spur gears, one-way clutches, disc brake, reverse-locking clutch, speed reducer, slip clutch, and DC motor with integral brake. A manual override of the hoist motor is also provided. A mechanical counter, synchronized to numerically display chain payout (from the grapple up position) in tenths of an inch is connected to the hoist drive system. Included with the grapple position indicator is a shaft encoder which can be used as a signal generator for programming the drive during automated gamma scanning of the loop at HFEF and also for remote readout of position.

The drive assembly and the rectangular tubes which encase the chain from the dome assembly to the hoist were made to stand off several inches from the main body. Clam shell type shielding sections can be added to the LHM shell should additional shielding be warranted. Up to 10 tons of additional shielding could be added with the grapple hoist removed.

Because of the mass and side mounted positions of the grapple hoist the LHM would tilt approximately $2-1/2^\circ$ when suspended by overhead cranes. A removable counter balance has been provided to maintain the LHM in an approximate plumb position.

I.3 Description of Transporter

The transporter system consists of a specially designed heavy duty trailer which carries the cradle, electrical and hydraulic components, jeep dolly, and a heavy duty tractor. This equipment provides the capability for transferring the LHM and contained loop between various facilities at the INEL. The transporter system is illustrated in Fig. I.1-1.

I.3.1 Trailer

The trailer was designed solely to support the LHM handling and transporting operations. The trailer is 58 feet long and 12 feet wide, with a load carrying capacity of 120,000 pounds. Because of the trailer length with the jeep dolly and tractor, the ETR truck door cannot be closed except when the tractor and jeep dolly have been disconnected and removed from the ETR building. A set of four power operating landing legs across the front of the trailer, as well as two sets of four stabilizer legs along each side of the trailer, can be lowered to support the trailer when the tractor and jeep dolly are disconnected. The air-ride suspension for the rear axles of the trailer can be exhausted to lower the rear frame onto the axles to create a rigid platform during LHM loading-unloading operations.

I.3.2 Cradle

The cradle assembly provides strong back support for the LHM during road travel and uprighting operations. Trunnions on the cradle frame pivot in a support structure on the trailer to facilitate uprighting of the LHM. This operation is accomplished using two telescoping cylinders which can be controlled for gentle stops at either end of the stroke. Different hydraulically operated pin mechanisms lock the cradle in either the vertical or near horizontal travel position.

The LHM is secured to the cradle by a T-bar slot device at the top of the cradle, two sets of two hydraulically operated over-center side clamps along the cradle, and a set of four slider type clamps located on the cradle base plate. These locking devices restrain the LHM movement in the vertical, lateral (horizontal) and axial directions. All hydraulically operated clamps are electrically interlocked to prevent out of sequence operations.

Four saddles and a pedestal type structure in the base plate support the LHM while it is in the cradle. The saddles are designed with inserts which may be changed to accommodate diameter changes in the LHM if additional shielding is added to the LHM. The cradle assembly is illustrated in Figure V.3.1.2-2.

I.4 SLSF Hoist System

A hoist system is designed in order to facilitate handling of SLSF an loop in and out of the ETF reactor. A grapple assembly is designed, which will hold the SLSF loop in the vicinity of top flange, and is linked to a chain driven hoist system. The chain storage reel is enclosed in the drive box mounted on the SLSF loop handling machine. The SLSF loop hoist system is shown in Fig. I.4-1.

I.5 Brief Outline of Stress Analysis Work

With assistance from ANL, EG&G Co. carried the initial task of stress analysis reported in Ref. (1) and (2). For acceptance of LHM and cradle structure at HFEF facility ANL (west) additional analysis was generated which was reported in Refs. (3) to (5). Due to the pressures of design schedules, and the pressures of acceptances of SLSF loop at HFEF facility, the design analysis of Refs. (1) to (5) was considered only preliminary. This report attempts a much broader perspective of analysis. Loading conditions and design criteria have been concluded from Refs. (6) to (10). Attempts have been made to use a three dimensional finite element analysis technique wherever it was necessary.

Fig. I.4-1 SLSF Loop Hoist System

The analysis and conformance to the design criteria performed in this report is under the guidelines of SLSF-LHS functional requirements as given in Appendix A (included in Volume V of this report).

Chapter II describes the static loading conditions, while Chapter III describes the dynamic loading conditions used under the guidelines and procedures of Appendix A. Chapter IV provides the design criteria for SLSF loop handling systems. Chapter V gives the engineering considerations in the analysis. Chapter XI provides the stress analysis of SLSF-LHM and Chapter VII provides a thermal analysis of LHM.

II. STATIC LOAD CONDITIONS

II.1 Loop Loads on LHM

When the LHM is in the horizontal position, the loop is supported inside the cask, by the grapple assembly and two sets of support brackets enter the cask through the middle and lower flanges. To estimate the loads on the three supports, it is assumed that the LOOP is a beam carrying the main loop components as concentrated loads and that the supports are simply and free from restraining moments. The Slope-Deflection method of analysis will be used. Refer to Fig. II.1-1.

At Section B the fixed-end moments are

$$F_{BA} = \frac{1}{2 \times 183.72} \left(790 \times 21.3 \times 162.4 \times 205 + 3355 \times 78.8 \times 104.9 \times 262.5 + 5524 \times 165.3 \times 18.4 \times 349 \right)$$

$$= 19.8325 \times 10^4 \text{ in. lbs. (clockwise)}$$

$$F_{BC} = \frac{-439 \times 63 \times 71.6 \times 197.6}{2 \times 134.62} = -1.0799 \times 10^4 \text{ in. lb. (counterclockwise)}$$

$$M_{BA} = F_{BA} + \frac{3EI}{183.7} \theta_B, \quad M_{BC} = F_{BC} + \frac{3EI}{134.6} \theta_B$$

$$M_{BA} + M_{BC} = +18.7526 \times 10^4 + 0.0386 \frac{3EI}{EI} \theta_B = 0$$

$$EI\theta_B = -485.567 \times 10^4$$

$$M_{BA} = 19.8325 \times 10^4 - \frac{3 \times 485.567 \times 10^4}{183.7}$$

$$= 11.9026 \times 10^4 \text{ in. lb.}$$

$$M_{BC} = -1.0799 \times 10^4 - \frac{3 \times 485.567 \times 10^4}{134.6}$$

$$= -11.9026 \times 10^4 \text{ in. lbs.}$$

From left of B

$$\sum M_B = 183.7 R_A - 790 \times 162.4 - 3355 \times 104.9 - 5524 \times 18.4 + 11.9026 \times 10^4 = 0$$

$$R_A = \underline{2489} \text{ lb. (acting up)}$$

From right of B

$$\Sigma M_B = -11.9026 \times 10^4 - 134.6 R_C = 0, R_C = -88.4 \text{ lb. (acting down)}$$

Total weight of loop loads = 9808 lb.

$$R_B - 9808 - 884 + 2489 = 0$$

$$R_B = +8203 \text{ lb. (acting up)}$$

II.2 Loop Handling Machine - Component Weights

Loop Handling Machine (LHM) component weights and center of gravity (CG) locations are shown in Figure II.2-1 and tabulated in Table II.2-1. Major subassemblies such as gate valve, hoist assembly, dome assembly, counterweights, etc., were actually weighed prior to assembly. Weights of flanges, shells and shielding were calculated.

Table II.2-1 shows that the computed LHM weight is within 1.5% of the actual measured weight. This 1250 pound weight difference is attributed to grapple chain, electrical conduit, fasteners, etc., and was spread evenly along the length of the LHM to locate the CG.

The loop weight of 9808 lb shown in Table II.2-1 is an estimated value; the actual weight is 7500 lb. In design calculations a loop weight of 10,000 lb was used because of the possibility that future loop weights would reach the higher value.

II.3 Loop Handling Machine - Lifting Lugs

II.3.1 Vertical Hoist Mode

The Loop Handling Machine (LHM) is lifted by means of a 50 Ton crane attached to the two lifting lugs through shackles and clevises as shown in Fig. II.3.1-1. A separate cable fall is used for each attachment and the individual maximum lifting forces are taken as 25 Tons, each. This value could conceivably be exceeded under certain dynamic conditions, e.g., a sudden stop as the LHM is lowered.

II.3.2 Horizontal Hoist Mode

Figure II.3.1-1 shows the LHM in the normal vertical position for hoisting. A horizontal hoisting load configuration is shown in Fig. II.3.2-1. The latter was originally considered as a means of erecting the LHM prior to vertical hoist, but was not used and is included here for completeness.

Fig. II.2-1 LHM Component Weights and CG's

Table II.2-1. Loop Handling Machine - Component Weights

LHM COMPONENTS	WT (LB)	xCG (in)	yCG (in)	zCG (in)	Drawing Number
Shield Valve Assembly	3560	+ -66/4	-	-	R1030-0006-DE
Mainbody - Lower Half					-0106-DE
+ From Trunnion Pin					
* From Fig. II.2-1					
Flange - Upper	1205	80.3/150.3	-	-	-0106-DE
Flange - Lower	3075	-55.5/14.5	-	-	-0106-DE
Loop Support - Upper	435	80.9/150.9	-	-	-0011-DE
Loop Support - Lower	430	-53.7/16.3	-	-	-0012-DE
Outer Shell 13.8 #/in	1781	13.9/83.9	-	-	-0006-DE
Inner Shell 2.4 #/in	301	13.9/83.9	-	-	-0006-DE
Shield - Upper Section 122.6 #/in	5647	55.3/125.3	-	-	-0006-DE
Shield - Lower Section 216.9 #/in	18003	-9.2/60.8	-	-	-0006-DE
Chain Break Box	305	67.0/137.0	-	-	-0006-DE
Hold Down Strap	100	56.0/126.0	-	-	-0006-DE
Grapple Drive Assembly	6710	-5.0/65.0	-	36.0	-0005-DE
Mainbody - Upper Half					-0103-DE

Table II.2-1. Loop Handling Machine - Component Weights (cont)

LHM COMPONENTS (cont)	WT (LB)	xCG (in)	yCG (in)	zCG (in)	Drawing Number
		+ *			
Flange - Upper	709	271.6/341.6	-	-	-0103-DE
Flange - Lower	454	84.5/154.5	-	-	-0103-DE
Grapple and Sleeve	994	264.6/334.6	-	-	-0010-DE -0102-DD
Outer Shell 18.5 #/in	3394	178.6/248.6	-	-	-0103-DE
Inner Shell 2.3 #/in	422	178.6/248.6	-	-	-0103-DE
Shield 122.6 #/in	22489	178.6/248.6	-	-	-0103-DE
Lifting Lugs (2)	1219	119.4/189.4	+23.9	-	-0103-DE
Counterweights (2)	3160	131.4/201.4	+47.3	-56.7	-0020-DE
Shackles (2)	80	167.9/237.9	+28.5	-	-0004-DE
Hold Down Strap	100	183/253.0	-	-	-0103-DE
Dome Assembly	854	276.1/346.1	-	-	-0009-DE
Chain Conduit	234	160/230.0	-	13.8	-0103-DE
Grapple Drive Mnt'g Angles	620	13.9/83.9	+11.0	8.5	-0106-DE

Table II.2-1. Loop Handling Machine - Component Weights (cont)

LHM COMPONENTS (cont)	WT (LB)	x_{CG} (in)	y_{CG} (in)	z_{CG} (in)	Drawing Number
LHM Subtotals					
Total WT (Computed)	80749				
Total WT (Measured)	82000				
Net Distributed	1251	101.8/171.8			
Net/Total	1.5%				
Loop Components					
Miscellaneous (Upper)	790	243.3/314.3	-	-	
Heater	3355	185.8/255.8	-	-	
Pump	5224	99.3/169.3			
Miscellaneous (Lower)	439	9.3/79.3	-	-	
Loop Subtotals	9808	$\bar{x}_{CG} = 136.5/206.5$	0	0	
LHM and Loop Totals	91808	$\bar{x}_{CG} = 86.9/156.9$	$\bar{y}_{CG} = \sim 0$	$\bar{z}_{CG} = \sim 0$

*Values shown for 50 ton crane (~1.1g)
Actual weight is 91,838 lb

Fig. II.3.1-1 Vertical Lifting (LHM)

Fig. II.3.2-1 Horizontal Lifting (LHM)

II.4 Loop Handling Machine - Cradle Composite

The Loop Handling Machine (LHM) is transported in a cradle which is pivoted to the transporter trailer as shown in Figs. II.4.1-1 through II.4.4-1. The LHM-cradle combination is raised to a vertical position by means of the telescoping hydraulic cylinders shown. Computation of the composite weight and center of gravity (CG) location are given in Table II.4-1. The coordinate system is fixed in the LHM-cradle assembly. The z-axis coincides with the trunnion axis. The x-axis is parallel to and the x-y plane contains the LHM center line.

Load cases include the travel position and three representative lift positions.

II.4.1 Travel Position

During road travel, the LHM-cradle is supported by two pads mounted on the trailer which exert forces normal to the cradle axis as shown in Fig. II.4.1-1. Trunnion force components are required to support the axial weight component caused by the 10° elevation.

II.4.2 10° Lift Position

Figure II.4.2-1 shows the LHM-CRADLE combination as it is lifted clear of the travel support pads by the two hydraulic cylinders. Constant velocity is assumed. However, depending upon the performance of the hydraulic system, an initial peak cylinder force in excess of the 90,000 lb equilibrium value may occur.

II.4.3 46.5° Lift Position

An intermediate lift position has been selected for structural analysis with the hydraulic cylinders in the vertical position as shown in Fig. II.4.3-1. The LHM-cradle elevation angle is 46.5°. Although previous analyses used a value of 41°, results should not differ significantly using 46.5°.

II.4.4 90° Lift Position

Figure II.4.4-1 shows the LHM-cradle in the vertical position just prior to contact with the vertical stops on the trailer.

II.4.5 Transport Loads

The Tie-Down System is required to support a 2g transport load in all directions without yielding. By implication, the LHM cask shell and the cradle assembly will be subjected to a similar loading with the LHM-cradle combination in the 10° Travel Position.

Table II.4-1

Combined LHM (w/Loop) and Cradle
In Cradle Coords: (Origin at Trunnion)

$$W_{CR} = 9282 \text{ (lb)}, \quad \bar{x}_{CR} = 54", \quad \bar{y}_{CR} = -27"$$

$$W_{LHM} = 91808, \quad \bar{x}_{LHM} = 86.9", \quad \bar{y}_{LHM} = -5.25"$$

$$W \bar{x} = W_{CR} \bar{x}_{CR} + W_{LHM} \bar{x}_{LHM} = 8.479 \times 10^6$$

$$\bar{x} = 83.9", \quad W = W_{CR} + W_{LHM} = 101,090 \text{ lb}$$

$$W \bar{y} = W_{CR} \bar{y}_{CR} + W_{LHM} \bar{y}_{LHM} = -7.326 \times 10^5$$

$$\bar{y} = -7.25" \quad \bar{z} \approx 0$$

Fig. 11.4.2-1 10° LIFT POSITION

Fig. II.4.3-1 46.4° LIFT POSITION

II.5 Trailer Loading

The trailer loading lists the vertical loads acting on the trailer when it is supported on the twelve hydraulic jacks. These jacks are inactive when the loop handling system is traveling - (see Fig. II.6-1). They are used in parked position of the trailer when the driving unit is removed. The jacks are shown as circles in Figure II.5-1. In Fig. II.5-2 the loads on the trailer are shown for the LHM in both the horizontal and vertical position.

II.6 Floor Loadings

The loads acting on the floor when the SLSF Handling Machine Transporter is parked in a facility is shown in Fig. II.6-1.

Fig. II.5-1 Trailer Jacks (by circles) and wheel positions (by rectangles) located in the plan view of trailer.

Fig. II.5-2 LHM - CRADLE Loading on Trailer

GROSS LOAD — 196,200 LB
 (193,579) — 38.5 TON LHM
 ((215,579)) — 45 TON LHM AND 5 TON LOOP

Fig. 11.6-1 Trailer Floor Loading

III. DYNAMIC LOADING CONDITIONS

III.1 Dynamic Loading During Travel

During establishment of the loop handling machine and transporter design requirements and conceptual designs, the question arose concerning the magnitude of shock and vibration the SLSF Loop could possibly experience during highway transport at NRTS. In an attempt to answer this question, the referenced material was searched and pertinent data from References (11), (12), (13) and (14) were utilized.

In reviewing the data, the following items were observed:

- (1) The referenced data cover a span of 13 years, 1958 to 1971. The referenced data were collected and compiled during this time span by different researchers who in all probability used different vehicles, loads and data collection and compiling techniques.
- (2) In spite of the 13-year time span and the probably different researchers and techniques, the attached referenced data are in good agreement in that:

- (a) The peak truck transportation vibrations are shown to occur at frequencies of 3 to 5 Hz, 10 to 20 Hz, 40 to 70 Hz and 120 to 140 Hz.

- (b) It can be shown from Ref. (15) that the acceleration experienced by the vehicle tire surface is directly proportional to the square of the speed of the vehicle, so that impact loads due to a given road irregularity increase rapidly with speed. Data in the References (1) to (5) indicated that a 1.0 to 6.0 "G" acceleration occurring at the peak vibration frequencies can be expected during high speed (50 to 70 MPH) operation of the transporter. The data also show that low speed (15 to 20 MPH) operation can result in accelerations of 1.0 to 1.5 "G", also occurring at the peak vibration frequencies.

To answer the question on shock and vibration that the SLSF Loop can expect to see during transportation on the NRTS roads, the following analysis was made as reported in the following chapters and Volume II of this report:

- (1) An analysis of the SLSF Loop and components while loaded in the LHM and in a nearly horizontal position to determine the natural frequency of the SLSF Loop and components. The vibration frequency peaks for the transporter could act as a forcing function and cause destructive vibration of loop

components which are in resonance with the expected transporter vibrations. If resonance is found to be present, damping of the transporter vibrations or changing the loop natural frequency will be required to prevent possible loop component damage.

(2) The SLSF Loop transporter will be operated at slow speeds (15 to 20 MPH) over the NRTS highways, therefore, the loop components could experience a 1.5 "G" acceleration during the peak vibration periods. The design of the loop components, as well as the LHM loop supports are analyzed for this loading to ensure against damage during normal transport conditions.

III.2 Dynamic Loading During Elevation of LHM

For the purpose of lifting the LHM from the cradle from a vertical position, the cradle and LHM together in the tie-down situation is elevated by means of two telescopic cylinders from a new horizontal position to a vertical position. It was considered necessary to analyze the LHM for three loading conditions listed in the following:

- (1) LHM tied down to the cradle structure and is laid on saddle supports at 10° from horizontal position.
- (2) LHM tied down to the cradle structure and is at 46.5° from the horizontal position during elevation. At this position the cradle telescoping cylinders are expected to be vertical.
- (3) LHM tied down to the cradle structure and is vertical.

III.3 Dynamic Loading During Single Failure Mode

Several modes of single failure are expected on the loop handling machine and cradle assembly which we outlined in Refs. 6 and 7. These are as follows:

(A) Electrical Power Failure

Phase: Raising or Lowering Cradle with LHM

Effects on System Operation:

All operations cease. Cradle maintains position at time of power outage. Cradle can be induced to creep downward by manual override of solenoid valve to tank. Creep rate extremely slow because flow control valve (FCV) goes to minimum flow (35 cc/min) with loss of power. Needle valve bypass around FCV being considered.

(B) Guillotine Rupture of Both Cylinder Extended Lines:

Phase: Lowering of Cradled LHM

Effects on System Operation:

Oil spews from cylinder-side hose remnants of the extend lines, which at this phase are discharging oil to tank. Velocity fuses in extend line at each of the cylinders will limit oil escape to 30 gpm per port causing the cradle to settle safely to the down position.

(C) Largest Packing Nut on One Cylinder Blows Completely Free:

Phase: Lowering of Cradled LHM

Effects on System Operation:

Oil spews freely from open annulus. Cradle suddenly lurches and settles rapidly on faulted cylinder side. The undamaged cylinder is simultaneously loaded by the unbalanced load and its resistance causes racking of the cradle. The sudden increase in load on the undamaged cylinder results in a high pressure surge which is attenuated somewhat by the compressibility of the oil, expansion of the flex lines, and gross oil leakage from all packing glands on the undamaged cylinder. The spherical rod end bearings on the cylinders permit cylinder compliance with the rack condition and the cradle continues to lower jerkily until movement is arrested by the crane rigging or by stops at lowered positions. If control action is taken, one cylinder would be able to stop the load in a fundamentally safe position.

(D) Other failure modes that are within the envelope of those above which were identified are:

- (1) Longitudinal rip in one cylinder housing.
- (2) Rupture of any 1" line or loss of one velocity fuse.
- (3) Buckling of remaining cylinder following the loss of other.

Phase: Lowering of Cradled LHM

Effects on System Operation:

Terminates the operation in a fundamentally safe condition.

(E) Instantaneous Failure of a Cylinder PinPhase: Elevating or Lowering of Cradled LHMEffects on System Operation:

This accident condition, though serious, does not appear to be the limiting mishap. If one of the cylinder pins (upper) were to fail instantaneously, load transfer to the remaining cylinder and racking of the cradle structure are expected to occur more quickly than that associated with a hydraulic failure. The damage potential is thus greater and the failed pin event should be considered as the limiting mishap.

III.4 Load Cases on Transporter

Load cases for which cradle structure was analyzed are listed below:

Load Case 1 Cradle with LHM at 10° from horizontal position with both cylinders active. Front row of trailer legs unsupported longitudinally.

Load Case 2 Cradle with LHM at 10° from horizontal position with one cylinder active, the other cylinder inactive. Front row of trailer legs unsupported longitudinally.

Load Case 3 Cradle with LHM at 45.4° from horizontal position with both cylinders active. Front row of trailer legs unsupported longitudinally.

Load Case 4 Cradle with LHM at 45.4° from horizontal position with one cylinder active, the other cylinder inactive. Front row of trailer legs unsupported longitudinally.

Load Case 5 Cradle with LHM at 90° from horizontal position with both cylinders active. Front row of trailer legs unsupported longitudinally.

Load Case 6 Cradle with LHM at 90° from horizontal position with one cylinder active, the other cylinder inactive. Front row of trailer legs unsupported longitudinally.

Load Case 7 Vertical IG load during travel. All rows of trailer legs inactive.

Load Case 8 Lateral IG load during travel. All rows of trailer legs inactive.

Load Case 9 Longitudinal IG load in forward direction during travel. All rows of trailer legs inactive.

Load Case 10 Longitudinal IG load in backward direction during travel. All rows of trailer legs inactive. This load case discarded as improbable with administrative control suggested for non-occurrence.

Load Cases 11 to 16 Same as Load Cases 1 to 6 except that the front row of legs supported longitudinally.

Fig. II.4.4-1 90° LIFT

IV. SLSF HANDLING SYSTEM DESIGN CRITERIA

A schematic of the organization of the design criteria for the SLSF Loop Handling System is provided in Fig. IV-1. This chart was used as a basis during the workscope of stress analysis of LHS.

The analysis performed herein in Volume I and Volume II of this document will be within the general guidelines of AISC specifications although all the detailed aspects of AISC specifications, are not intended. Later, a non-mandatory Volume III of this document may be issued which could include some useful, but not essential, detailed investigations of the dynamic structural behavior of the SLSF loop handling system.

IV.1 Loop Handling Machine

Table IV.1-1 provides a listing of various parts of the LHM, the material, properties including the yields and ultimate strength of the material and a list of reference drawings for the material. For the components of LHM, since the specific nature of dynamic loadings and their effect on the components is not known without a detailed study of each event, it is difficult to make complete assessments using static analysis methods. Based on static analysis, therefore, factor of safety for each calculated stress (lg loading) is reported based on tensile and yield strength or the shear yield strength of the material as it is applicable based upon the loading of each component. If in some situation where the factor of safety does not provide adequate margin for emergency dynamic loadings, if justification of such loading is available, then either a redesign of that particular component is recommended or the administrative procedures is recommended against occurrence of such dynamic loadings.

IV.2 LHS Cradle Structure

Design criteria for the SLSF-LHS Cradle Structure is governed by the agreements between SLSF project management and HFEF management who considered application of only limited aspects of AISC specifications in Ref. 5 to 9. In these references, AISC specifications were applied to the Cradle structure welds. Further, AISC specifications were used in determining the cross-section compactness, limiting the calculated stresses to allowables only on a component allowable basis. Treatment of these allowable stresses in meeting the limited aspects of AISC specifications is to be addressed in Volume II of this document.

The cradle structure is composed of a series of box beams. AISC specifications require a comprehensive treatment, as outlined in Appendix B, (included in Volume IV of this report), to establish the structural integrity of the cradle structure. Depending on the limitations of time and funding, all the aspects of AISC specifications as reported in Appendix B will be applied to the cradle structure in Volume III of this document.

Although not generally required by the AISC specifications, determination of zones of high stress concentrations, in the cradle structure by further detailed finite element analysis is likely to produce useful results. Investigations of dynamic behavior under postulated emergency conditions by actual dynamic analysis of structure is expected to produce very useful results also. These investigations are not mandatory and hence are to be part of Volume III of this document depending on time limitations.

IV.3 LHS Trailer

Although SLSF-LHS trailer structure is included in the analysis in Volume II of this document, the structural integrity of this trailer is considered to be the responsibility of its manufacturer.

In Volume II of this document, the trailer is included in the structural modeling only to determine more accurately the effect of dynamic-accidental loading on the cradle structure.

Fig. IV-1 Schematic of Design Criteria and Analysis Work Organization of SLSF-LHS.

Table IV.1-1. Material Properties for LHM Components

Part	Material	S _y (ksi)	(S _S) _y (ksi)	S _{ult} (ksi)	Source of Data
LHM Cask	304 SST	30.	20.	80.	R1030-0103-DE
Cask Bolts	Steel	100.		125.	R1030-0004-PL
LHM Shield	Depleted Uranium	35.		60.	R1030-0100-DE
Grapple	17-R PH Cond. H1100	115.		140.	R1030-0254-DC
Grapple Finger Pivot	17-4 PH Cond. H1150	105.		135.	R1030-0257-DB
Grapple Shackle	4340 Stl. 34-37 R _C	130.		153.	R1030-0241-DB
Grapple Drive Bolts	Steel	100.		125.	R1030-0005-PL
Grapple Bolts	SS 304	50.		90.	R1030-0010-DE
Grapple Drive Shaft 9* (15)	4340 Stl. 35-38 R _C	138.	0.57 S _y	160.	R1030-0168-DB
Grapple Drive Shaft 11* (16)	Steel	138. -156.	0.57 S _y	160.	R1030-0169-DC
Grapple Drive Shaft 14* (37)	4340 Stl. 35-38 R _C	138.	0.57 S _y	160.	R1030-0427-DB
Grapple Drive Shaft 22* (44)	4340 Stl.	138.	0.57 S _y	160.	R1030-0181-DC
Grapple Drive Shaft 28*	4340 Stl.	95.	0.57 S _y	110.	R1030-0139-DB
Woodruff Key Shaft 11* (16)	SAE 2340 35-38 R _C	147.	0.57 S _y	165.	R1030-0014-DE
Chain Reel Shaft		70.	0.57 S _y	82.	R1030-0141-DB
Pulley Block Anchor Rod	4340 Stl.	95.	0.57 S _y	110.	R1030-0155-DB
Loop Support Casting	CI		65.	50.	R1030-0011-PL

Table IV.1-1.(cont) Material Properties for LHM Components

Part	Material	S _y (ksi)	(S _s) _y (ksi)	S _{ult} (ksi)	Source of Data
Lifting Link	4340 Stl.	135.	0.57 S _y	160.	R1030-0103-DE
Lifting Lug	Steel Plate	30.	20.	80.	R1030-0004-DE
Counter Bal. Arm	ASTM A-36	36.	.6 S _y	60.	R1030-0020-DE
Counter Bal. Bolts	ASTM A 320 Grade L7	100.		125.	R1030-0020-PL
T-Slot	304 SST	30.		80.	R1030-0103-DE
T-Bar	Low Carbon Steel	36.	.6 S _{ult}	60.	R1030-0307-DC
Tie-Down Bolts	Steel	100.			
Tie Down Strap	SS	30.	20.	80.	R1030-0004-DE
Welds	SS	30.	20.	80.	
Tie-Down Clamp Bar	Steel	36.	.6 S _y	60.	R1030-0334-DC
Hold-Down Clamps	Steel				R1030-0045-DE
Hold-Down Bolts	Steel	100.		125.	R1030-0045-PL
Trunnion Assembly					R1030-0045-P1
Trunnion Pin	Stress Proof GRD. & Pol. Stl.	100.		125.	R1030-0380-DC
Cradle Frame Assembly	Cor-Ten Steel	50.	.6 S _{ult}	70.	R1030-0303-DE
Base Plate	ASTM A242 Cor-Ten Stl.	50.	.6 S _{ult}	70.	R1030-0046-DE

Table IV.1-1.(cont) Material Properties for LHM Components

Part	Material	S_y (ksi)	$(S_S)_Y$ (ksi)	S_{ult} (ksi)	Source of Data
Compression Spring	Oil Tempered Steel SAE 6150	90.			R1030-0411-DB
Disk Spring	Chrome-Vanadium	350. 420.			
Pinion 15 and Gear 29	{ Allowable	$S_t = 43,000 - 56,000$ psi		AGMA	
	{ Allowable	$S_c = 160,000 - 180,000$ psi		Standards	

*Shaft Numbers on Fig. V.2.4-1

V. ENGINEERING DESIGN INFORMATION

V.1 Hoist Systems

V.1.1 ETR Crane Information

- (1) Cable: McWhite 7/8" diameter "extra improved" 6 x 37 wire rope.
Premium plough steel.
- (2) Eight falls on cable.
- (3) Hooks are 33'6" from floor in full up position.
- (4) Bottom pulley and hook assembly 6100 lbs.
- (5) Maximum hoisting speed 25 ft/min.
- (6) Traverse speeds: Trolley - 55 ft/min.
Bridge - 80 ft/min.
- (7) Size of main bridge beam: 2'4" wide, 5 ft 1-7/8" deep
5/16" side plates
1" top plate
7/8" bottom plate.
- (8) Beam span: 85'6".
- (9) Weight of machine plus loop: 50 tons.
- (10) Weight of trolley: 31,400 lbs.

V.1.2 Crane Hooks and Shackles

Information on crane hooks and shackles is available at EG&G Co. It is believed that these components have been purchased for the required capacity of handling the LHM.

V.1.3 Lifting Links

A lifting link is pivotally connected to each of the two lifting lugs as shown in Fig. V.2.6-1. The links are also pivotally connected to the shackles that engage the crane hooks when the LHM is being lifted.

V.1.4 Spreader Bar

At ETR, EG&G Co. will handle the loop handling machine directly through crane hooks and shackles. At HFEF (hot fuel examination facility) the LHM would be connected to the crane indirectly, a spreader bar connects the LHM to the crane hooks and shackles. HFEF facility engineers have designed this spreader for LHM handling in their facility. It is believed that this spreader bar has been adequately designed by HFEF facility engineers.

V.2 Loop Handling Machine (LHM)

V.2.1 Functional Requirements

LHM functional requirements are outlined in detail along with the functional requirements of entire loop handling system in Appendix A (included in Volume IV of this report).

V.2.2 Loop Handling Machine Cask

The LHM cask is made in two sections (upper and lower halves), each having appropriate flanges for joining the two sections and mounting the end assemblies (see Fig. V.2.2-1). The lower section has a 1/2 in. outer shell, the upper section has a 3/4 in. outer shell and each has a 1/8 in. inner cylindrical shell made of 304 stainless steel. These shells are welded to the flanges as detailed in Fig. V.2.2-2. A cylindrical uranium shield of variable cross section is mounted between the inner and outer stainless steel shells as indicated in Fig. V.2.2.2. The uranium shield is spigotted to the flanges, i.e., it is not welded or fastened to the flanges and is free to move longitudinally with about 1/16 in. clearance. Allowing for radiusing at the spigot joint, the bearing length is on the order of 1/2 in., suggesting that these joints will transmit shear but very little bending moment between the flanges and the shield.

The LHM is supported in the cradle by four saddles as shown in Fig. V.2.2-1. In addition, the LHM is clamped to the base plate with four wedge action clamps and is strapped to saddles #2 and #4 as shown.

The four saddles act directly on the outer shell. Ideally, when the LHM-cradle is horizontal, the shield will act as two simply supported beams loading the cask in shear at the flanges. However, relative bending deflection and local distortion at the saddle bearings may allow the outer shell and shield to come into contact so that the shield may be partially supported at these points (check manufacturing tolerances).

At saddle #4 two internal stiffening rings are welded to the outer shell in the shield step to minimize outer shell distortion.

FIG. V.2.2-2 LIM CASK WELDS AND SHIELDING

V.2.2.1 Lifting Lug Loads

The lifting lugs are welded to the outer shell of the upper portion of the LHM cask. Stress intensification due to shell bending in the region immediately outside the weld line should be checked, particularly near the upper corners, which are not connected to a flange. This problem is addressed in Section VI.2.2.

V.2.2.2 Saddle Loads

The 2g transport loads (II.4.5) are applied to the LHM outer shell through the saddle contact areas. In some instances these saddle pressures may be concentrated because of partial saddle contact. Since the saddle loads are indeterminate, the LHM-Cradle System must be analyzed as one structure.

V.2.3 Grapple

The grapple, shown in Fig. V.2.3-1 is a mechanical device that provides means for connecting the loop to the hoisting system. It consists of a main body (1) that is connected to the hosting chains by two diametrically opposite shackles, (2) a member, (3) that can be moved vertically relative to the main body, and six fingers (4) that can rotate about pivots (5). After contact between the loop (6) and the main body on line AB, the grapple fingers are brought into load-bearing positions by means of a manually operated gear-driven mechanism that moves member (3) downward to make contact with the fingers at C and D.

The drawing number for the shackle, (2) is R1030-0241-DB, for the pivot, (5) it is R1030-0275-DB and for the finger, (4), it is R1030-0254-DC. All cap screws are stainless steel with a tensile yield point of 50,000 psi.

FIG. V.2.3-1 GRAPPLE

V.2.4 Grapple Drive

This drive includes all of the mechanical components that connect the drive motor to the hoisting chains and are utilized in raising and lowering the grapple and the loop. The Grapple Drive drawings are arranged in three main subassemblies, namely;

(a) The Motor Drive Assembly includes the drive motor, the right angle gear drive, the torque limiter, the Harmonic Drive gear reducer, and an output sprocket.

(b) The Gear Drive Assembly consists of all gears, sprockets, clutches, and shafts that serve as a speed reducer and connect the motor drive sprocket to the sprockets that control the motion of the chain attached to the grapple.

(c) The Loop Location Counter Assembly consists of a box containing small bevel and spur gears that are driven by a sprocket from the gear drive and serve to drive an Encoder and a Counter that indicate the position of the Loop.

Figure V.2.4-1 shows assemblies (a) and (b) and the chains and their attachment to the Grapple. Assembly (a) is shown separately in Fig. V.2.4-2 and assembly (c) on Fig. V.2.4-3. The Chain Drum Spring Motor that keeps the chain from becoming slack on the drum is shown on Fig. V.2.4-4. The torque on the drum shaft is produced by two negator or constant-load springs. A parts list of all parts shown in Fig. V.2.4-1 follows.

FIGURE V.2.4-1 GRAPPLE DRIVE ASSEMBLY

$$\text{INPUT TORQUE TO HARMONIC DRIVE (ASSUME 70\% EFFICIENCY)} = \frac{15,924}{140 \times 0.7} = 162 \text{ IN. LBS.}$$

FIGURE V.2.4-2 MOTOR DRIVE ASSEMBLY

FIGURE V.2.4-3 LOOP LOCATION COUNTER ASSEMBLY

FIG. V.2.4-4 CHAIN DRUM SPRING MOTOR

Parts List

1. 7-1/2 H.P., D.C. electric motor with 105 ft-#brake, variable speed. Reliance Electric Company.
2. Flexible Coupling - Ajax Cat. No. 6M-73.
3. Right Angle Drive - Dalton Model RA100 Cat. No. 768.
4. Torque Limiter - Model TT2X, torque capacity 180 in.-#, Helland Research and Engineering, Inc.
5. Harmonic Drive (custom made for SLSF Project) output torque 80,000-#, overhung load 7300#, USM Corporation, Gear System Division.
6. and 8. Sprocket - Drawing No. R1030-0189-DB.
7. and 19. 1" Pitch Triple Strand Chain - average ultimate strength 40,500 lbs. Rex. Chainbelt, Inc.
9. Shaft - Drawing No. R1030-0168-DB.
10. Ball Bearing, Bearing No. 221-M, Marlin-Rockwell, Cat. No. 60.
11. Shaft - Drawing No. R1030-0169-DC.
12. Reverse-Locking Clutch, Model RL-160A/2.500, rated capacity 26,500 in.-#, Cat. No. 4102, Formsprag Company
13. Ball Bearing, Bearing No. 315-R, Marlin Rockwell, Cat. No. 60.
14. Shaft - Drawing No. R1030-0527-DB.
15. Gear Pinion - Drawing No. R1030-0175-DB.
16. Pillow Block - Drawing No. R1030-0176-DB.
17. Over-running Clutch - Torque capacity 4000 ft-#, Model No. 700, Formsprag Cat. No. 2106.
18. Brake Assembly - Drawing No. R1030-0079-DE.
20. Sprocket-Idler, Drawing No. R1030-0235-DB.
21. Pillow Block - Drawing No. R1030-0178-DB.
22. Shaft - Drawing No. R1030-0181-DC.
23. Sprocket-Drive, Drawing No. R1030-0179-DB.
24. Sprocket-Idler, Drawing No. R1030-0159-DB.
25. Load Cell - Capacity 20,000# Type T2P1 - BLH Electronics, Inc., Cat. No. 401-4.
26. Pulley Block Anchor Rod, Drawing No. R1030-0155-DB.
27. Chain Storage Reel, Drawing No. R1030-0143-DC.
28. Chain Drum Shaft Drawing No. R1030-0139-DB.
29. Gear - Drwing No. R1030-0180-DB.

V.2.5 Grapple Drive Brackets

The grapple drive assembly is mounted on the LHM as shown in Figure V.2.5-1. The mounting brackets are welded and gusseted T-section rails as shown in Figure V.2.5-2. The grapple drive is fastened to the rails by means of 13 5/8-11 UNC bolts, yield strength = 100,000#/bolt, on each side. The bolts are spaced on 6 in. centers (72 in.) and the bolt pattern is centered in the 77 1/4 in. assembly length.

If the grapple drive assembly is sufficiently rigid to absorb internal bearing loads and motor torques without significant distortion, the fasteners and weld should be checked for the vertical hoisting position when the assembly dead weight and chain tension cause application of shears and moments to the brackets. Maximum total chain tension is set at 11,000 lb. by means of load cells.

V.2.6 Lifting Lug Assembly

The lifting lug assembly shown in Figure V.2.6-1-2 is a welded structure made of 5/8 inc. stainless steel plate. When the LHM is lifted in the vertical position, half of the weight (45,804#), is applied to each of the two lugs by means of the clevis, shackle and pin shown in Fig. V.2.6-1. A single hoist failure mode in which the total LHM weight (91,808#) is applied to one lug at a 19° angle is also of interest. As shown on Fig. V.2.6-2 both the 4-1/2 inc. stainless steel pins are welded to the three webs. The inner two webs are part of a closed box which extends over most of the lug length. (The lower 20 in. of the lug is an open gusset structure.) The outer bay is open on two faces to allow the lifting link to rotate through 90° during erection and hoisting.

V.2.7 Counterbalance

To counteract the weight of the grapple drive assembly and insure that the LHM will hang vertically when suspended from the lifting lugs a counter-balance assembly is bolted at two places to each of the two lifting lugs. The means of attachment is shown on Figures V.2.7-1 and V.2.7-2. Cross section A-A on Fig. V.2.7-1 is shown on Fig. V.2.7-3.

V.2.8 T-Slot Attachment

The T-Slot is welded to the LHM cask as shown in Fig. V.2.8-1; it engages a matching T-Bar (Section V.3.1.5) that is bolted to the cradle.

FIG. V.2.5-1 GRAPPLE DRIVE MOUNTING BRACKETS

Fig. V.2.6-1 Lifting Lug - Pin and Lifting Link Locations

V.2.6 - 2 LIFTING LUG

FIG. V.2.7-1 COUNTERBALANCE ASSEMBLY

(VIEW 1)

FIG.V.2.7-2 COUNTERBALANCE ASSEMBLY

(VIEW 2)

SECTION A-A

MAT'L. ASTM A-36

 $S_Y = 36,000 \text{ PSI}$

FIG. V. 2.7-3 CROSS SECTION A-A

IN FIG. V.2.7-1 (VIEW 1)

FIG. V.2.8-1 T-SLOT ATTACHMENT TO LHM CASK

V.2.9 Loop Support Mounting Detail

When the loop is inside the cask it is supported at two locations by two sets of three mechanisms similar to that shown in Fig. V.2.9-1. The supports are arranged 120° apart around the cask. Figure V.2.9-2 shows the detail of the upper support pad.

FIG. V.2.9.-1 LOOP SUPPORT MOUNTING DETAIL

Fig.V.2.9-2 Upper Loop Support Pad

V.3 Transporter

V.3.1 Cradle Assembly

V.3.1.1 Functional Requirements

The functional requirements of the entire loop handling system including those of the cradle assembly are given in Appendix A (included in Volume IV of this report).

V.3.1.2 Cradle Frame

The cradle frame is shown in Fig. V.3.1.2-1 with the LHM placed and supported by the four saddles and base plate.

Figure V.3.1.2-2 shows the cradle frame with its two longitudinal beams and five cross beams. The cross sections of the longitudinal beams at sections A-A, B-B, and C-C are shown in Fig. V.3.1.2-3. Cross beams 1, 2, 3, and 4 shown in Figs. V.3.1.2-4 to V.3.1.2-11. The fifth cross beam, shown in Fig. V.3.1.2-12, is the cradle pivot support; it carries a bearing for the trunnion pin which permits the cradle to rotate relative to the trailer.

The weld connections are shown in Figs. V.3.1.2-13 to V.3.1.2-15. The end plate-base plate bolt connections in Fig. V.3.1.2-16. Base plate is treated in Section V.3.1.4.

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Fig.V.3.1.2-1 LHM ON CRADLE SADDLES

Fig. V.3.1.2-2 CRADLE FRAME

MTL. ASTM A 242
 $S_y = 50,000 \text{ PSI}$

BF = WIDTH OF FLANGE

TW = THICKNESS OF WEB

DW = DEPTH OF FLANGE

Fig. V.3.1.2-3 CROSS SECTIONS OF LONGITUDINAL CRADLE BEAMS

FIG. V.3.1.2-5 CROSS BEAM NO. 1 (TOP)

FIG. V.3.1.2-8 CROSS BEAM NO. 3 (FRONT)

FIG. V3.1.2-9 CROSS BEAM NO. 3 (TOP)

FIG. V.3.1.2-10 CROSS BEAM NO. 4 (FRONT)

FIGURE V.3.1.2-12

FIG. V.3.1.2-13

WELD CONNECTIONS

CROSS BEAM NO. 4 TO LONGITUDINAL BEAM

ALL WELDS ARE 1/4" FILLET

CROSS BEAM NO. 1 TO LONGITUDINAL BEAM

WELD SIZE

5/16"

1/4"

3/8"

WELD AREA

$$A = .312 \times .707 \times 3 = .66 \text{ IN}^2$$

$$A = .25 \times .707 \times 7 = 1.23 \text{ IN}^2$$

$$A = .375 \times .707 \times 7 = 1.85 \text{ IN}^2$$

FIG. V.3.1.2-15 WELD CONNECTION - END PLATE TO LONGITUDINAL BEAM

FIG. V.3.1.2-16 BOLT CONNECTION - END PLATE TO BASE PLATE

V.3.1.3 Saddles

The LHM is supported on the cradle by the four saddle assemblies shown in Figures V.3.1.3-1 through V.3.1.3-4. Figure V.3.1.3-5 shows a *typical saddle cross section which is welded to the cradle crossbeam as shown*. In normal operation, the load is carried to the crossbeam through the steel contact block and the rubber block shown. The retaining assembly is normally not in metal to metal contact.

V.3.1.4 Base Plate

This plate is attached to the cradle and the LHM rests on it when in the vertical position. The hole in the center of the plate permits the loop to be lowered into the reactor. (See Figures V.3.1.4-1 and V.3.1.4-2).

V.3.1.5 T-Bar

The T-Bar, which is fastened to the cradle assembly, engages the T-Slot on the LHM (Section V.2.8). Because of dimensional variations, the fit between the engaging members may be tight or loose. The T-Bar is a back-up for the tie-down and base plate clamps during erection. (See Figure V.3.1.5-1)

FIG.V.3.1.3-1 SADDLE NO. 1 WITH T-BAR

FIG.V3.1.3-2 SADDLE NO. 2 WITH TIE-DOWN MECHANISM

FIG. V.3.1.3-3 SADDLE NO. 3

FIG.V3.1.3-4 SADDLE NO. 4 WITH TIE-DOWN MECHANISM

NOTE : SADDLE MOVES $1/32''$
RADIALLY AND CIRCUMFERENTIALLY
TO DEFORM RUBBER LAYER

FIG. V.3.1.3-5 RADIAL SECTION OF SADDLE

FIG. 3.1.4-1 PLAN VIEW OF BASE PLATE

$\frac{5}{8}$ - 11 UNC 28 THD. THRU $\frac{5}{8}$ R. ONLY EQUALLY SPACED
ON $25 \frac{1}{2}$ DIA. B.C. (6 HOLES)

FIG. V.3.1.5-1 T-BAR ATTACHMENT TO CRADLE

V.3.1.6 Tie-Down Mechanisms

At saddles #2 and "4, the LHM is strapped to the cradle by means of tie-down straps. Hooks on the straps are engaged by the tie-down bar A which is part of the tie-down mechanism shown in Fig. V.3.1.6-1. Clamping is accomplished by rotating the eccentric shaft D.

The tie-down straps are detailed in Fig. V.3.1.6-2.

V.3.1.7 Horizontal Hold-Down Clamps

As shown in Figures V.3.1.7-1, V.3.1.7-2, and V.3.1.7-3 each of the four Hold-Down Clamps consists of a body that is clamped to the base plate and containing a groove in which a sliding member can be pushed, by means of a hydraulic cylinder, into firm contact with a flange on the end of the LHM; the clamps are approximately equally spaced circumferential around the LHM.

FIG.V3.1.6-1 TIE-DOWN MECHANISM

FIG. V.3.1.6-2 TIE-DOWN STRAP DETAIL

FIG. V.3.1.7-1 HORIZONTAL HOLD-DOWN CLAMP ASSEMBLY

FIG. V.3.1.7-2 BODY OF HORIZONTAL HOLD-DOWN

FIG. V.3.1.7-3 SLIDE OF HORIZONTAL HOLD-DOWN

V.3.2 Trunnion Pins

The trunnion pins, co-acting with containing surfaces, provide means for rotary motion of the cradle relative to the transporter. In Fig. V.3.2-1 the 5 in. diameter floating pin is free to rotate in both the containing member T attached to the transporter, which is fixed, and the containing member C attached to the cradle, which is free to rotate. To minimize friction the pin is coated with a dry lubricant (molybdenum disulphide). The mechanism associated with the 1 in. diameter rod is inoperative.

Cradle Bore	5.011/5.008	Drawing No. R1030-0303-DE (Sh. 7)
Pin Dia.	5.002/4.998	Drawing No. R1030-0380-DC
Trunnion Bore	5.008/5.005	Drawing No. R1030-0293-DD

V.3.3 Trunnion Assembly

The trunnion assembly, illustrated in Figure V.3.3-1, is mounted on the trailer and provides a pivotal support for the cradle, which occupies the space between the members M in Figure V.3.3.1 and pivots on two trunnion pins (Fig. V.3.2-1) about the centerline of the bearings B. Each of the members M is stiffened against sidewise loads by two struts S.

FIG.V.3.2-1 TRUNNION PIN ASSEMBLY

FIG. N.3.3-1 TRUNNION ASSEMBLY

V.3.4 Elevating Cylinders

V.3.4.1 Cylinders

The (two) Elevating Cylinders are five-stage devices as indicated in Figure V.3.4.1-1.

V.3.4.2 Cylinder Pins

The Elevating Cylinder Pin and Bearing (same top and bottom) are detailed in this section. The self-aligning bearings, which are attached to both ends of the cylinder, permit a misalignment of 6 degrees between the plane of the cylinder ends and the attachments to the cradle and the trailer. (See Figure V.3.4.2-1)

V.3.4.3 Attachment Brackets

The attachment Bracket mounting and structural details are shown in this section. (See Figures V.3.4.3-1, V.3.4.3-2, V.3.4.3-3 and V.3.4.3-4)

V.3.4.4 Elevating Cylinders

Two hydraulic elevating cylinders, one of which is shown in Figure V.3.4-1, are used to raise and lower the LHM-cradle combination between the horizontal travel position and the vertical hoist position. The upper end of the cylinder is pivotally connected to the cradle while the lower end is pivotally connected to the trailer.

FIG.VB.4-1 ELEVATING CYLINDER

STAGE	ROD DIA.	BORE DIA.	STROKE	SQ. AREA ROD	SQ. AREA BORE	SQ. AREA DIFF.	RATIO	PRESS. P.S.I.
1	9 3/4"	10 1/2"	22.0"	74.66"	86.59"	11.93"	7.26	1089
2	8 1/4"	9"	23.9"	53.46"	63.62"	10.16"	6.26	939
3	7"	7 1/2"	26.5"	38.49"	44.13"	5.69"	7.79	1168.50
4	6"	6 1/2"	28.1"	28.27"	33.18"	4.91"	6.75	1012.50
5	5"	5 1/2"	29.8"	19.64"	23.76"	4.12"	5.76	864

FIG.V.3.4.1-1 DETAILS OF ELEVATING CYLINDER

FIG. V.3.4.2-1 CYLINDER PIN

FIG. V.3.4.3-1 CYLINDER BRACKET - UPPER MOUNTING

FIG. V.3.4.3-2 CYLINDER BRACKET - LOWER MOUNTING

FIG. V.3.4.3-4 UPPER BRACKET ATTACHMENT

VI. STRUCTURAL ANALYSIS OF THE LHM AND ITS COMPONENTS

Structural analysis of the SLSF loop handling machine was performed for static as well as emergency hoist (when one cable is assumed to fail) loading conditions. Analysis is performed for components where significant loading effects are expected.

VI.1 Stress Analysis of Hoist System

Stress analysis of the hoist system was carried out for normal hoist, emergency hoist, and for a sudden impact type loading. No attempt was made to analyze certain components of the hoist system such as hooks, shackles and spreader bars, since they are not within the scope of this report. The hoist system analysis in the following sections is limited to crane cables and lifting links.

VI.1.1 Analysis of Acceleration and Impact

To accomplish the analysis of accelerations and impact under single failure mode operation would require a careful study of crane accelerations during handling sequence. It is assumed that administrative procedures are incorporated to operate the crane at an extremely low speed such that significant accelerations are not transmitted to the entire hoist system and only one directional motion at a time is considered.

VI.1.2 Stress Analysis of Crane Cables

Derivation of Dynamic Load Equation

W = Load on Hoist (lbs)

P = Dynamic Load (lbs)

δ_{st} = Static Deflection (in.)

δ = Dynamic Deflection (in.)

K = Stiffness of Cable

$$\delta = \frac{P}{K}, \quad \delta_{st} = \frac{W}{K}$$

$$\text{Strain Energy} = \frac{1}{2} P \delta - \frac{1}{2} W \delta_{st} = \frac{P^2}{2K} - \frac{W^2}{2K}$$

Fig. VI.1.2-1 Cable Loading

$$\left\{ \begin{array}{l} \text{Kinetic energy in weight} = \frac{1}{2} \frac{W}{g} V^2 \\ \text{Work done during drop} = W(\delta - \delta_{st}) \\ \quad = \frac{WP}{K} - \frac{W^2}{K} \end{array} \right.$$

Equating K.E. + work to strain energy

$$\frac{1}{2} \frac{W}{g} V^2 + \frac{WP}{K} - \frac{W^2}{K} = \frac{P^2}{2K} - \frac{W^2}{2K}$$

$$\frac{KW}{g} V^2 + 2WP - 2W^2 + W^2 = P^2$$

$$P^2 - 2WP + (W^2 - \frac{KW}{g} V^2) = 0$$

$$P = \frac{2W + \sqrt{4W^2 - 4W^2 + \frac{4KW}{g} V^2}}{2}$$

$$P = W + \sqrt{\frac{KW}{g} V^2} = W \left[1 + \sqrt{\frac{KV^2}{gW}} \right]$$

For steel cable K (approx) = $\frac{AE}{L}$

Sudden Stoppage of Hoist

$$P = W \left[1 + \sqrt{\frac{KV^2}{gW}} \right] \quad \text{(Refer to Section V.1.1 for data on crane cables)}$$

P = dynamic load on cables (lbs.)

W = static load on cables (lbs.)

K = stiffness of cable system (lbs./in.)

V = maximum hoisting speed (in./sec.)

For 6 - 37, 7/8" dia., premium plow steel cable

$$E = 30 \times 10^6 \text{ psi}$$

$$A = \text{total wire area} = 0.2703 \text{ in}^2$$

L = 120 in. (estimated minimum length when attached to machine)

$$K = \frac{AE}{L} = \frac{0.2703 \times 30 \times 10^6}{120} = 67,575 \text{ lbs/in.}$$

For 8 falls - 4 on each pulley block

$$K = 8 \times 67,575 = 540,600 \text{ lbs/in.}$$

$$V = 25 \text{ ft/min} = \frac{25 \times 12}{60} = 5 \text{ in/sec.}$$

W = weight of machine + loop + 2 pulley blocks

$$= 91,808 + 2 \times 6100 = 104,008 \text{ lb.}$$

$$\sqrt{\frac{KV^2}{gW}} = \sqrt{\frac{540,600 \times 25}{386 \times 104,008}} = 0.58$$

$$P = 104,008 \times 1.58 = \underline{\underline{164,332 \text{ lb}}}$$

$$\text{Cable capacity} = 8 \times 30.5 \times 2000$$

$$= 488,000 \text{ \#}$$

$$\text{Factor of safety} = \frac{488,000}{164,332} = 2.97$$

Fig. VI.1.2-2
LHM During Cable Breakage

Breakage of Cables to one Lifting Link

From Fig. VI.1.2-2 if the cables to the left-hand lifting link fail, the instantaneous value of the load in the right-hand cables (F_0) due to the LHM alone can be obtained from

$$F_0 a = I_{CG} \alpha \quad (\text{VI.1.1})$$

$$W - F_0 = \frac{W}{g} a \alpha \quad (\text{VI.1.2})$$

Substituting for α from (VI.1.1) in (VI.1.2)

$$W - F_0 = \frac{W}{g} a \frac{F_0 a}{I_{CG}} \quad \text{from which}$$

$$F_0 = \frac{W}{1 + \frac{W}{g} \frac{a^2}{I_{CG}}}$$

The moment of inertia I_{CG} is that of the LHM alone - it does not include the effect of the pulley blocks. From Section 2-11 the value of I_{CG} was found to be $2.213 \times 10^6 \text{ lb-sec}^2\text{-in.}$ From Fig. VI.1.2-2, $a^2 = 28.4 \text{ in}^2 = 808.8 \text{ in}^2$. Then

$$\frac{W}{g} \cdot \frac{a^2}{I_{CG}} = \frac{91,808 \times 808.8}{386 \times 2.213 \times 10^6} = 0.0869$$

$$F_0 = \frac{91,808}{1.0869} = 84,468 \text{ lb.}$$

Adding to this the weight of the pulley blocks

$$F_0^1 = 84,468 + 2 \times 6100 = 96,668 \text{ lb}$$

After the failure of the left-hand cables, the LHM will swing down and assume the position shown in the right hand view in Fig. VI.1.2-2. Assuming that the swinging takes place about point O as a fixed axis, the center of gravity will move down a distance $S = 4.75$ in as shown in the figure. The angular velocity of the LHM in the lower position can be obtained from the energy relation

$$WS = \frac{1}{2} I_0 \omega^2 \quad (\text{VI.1.3})$$

$$I_0 = I_{CG} + \frac{W}{g} b^2$$

$$I_{CG} = 2.213 \times 10^6 = \frac{W}{g} K^2 = \frac{91808}{386} K^2$$

$$K^2 = 9304 \text{ in}^2, \quad b^2 = 7612 \text{ in}^2$$

Then

$$I_0 = \frac{W}{g} [9304 + 7612] = 16916 \frac{W}{g}$$

From (VI.1.3)

$$\omega^2 = \frac{2W \times 4.75 \times g}{W \times 16916} = 5.62 \times 10^{-4} g$$

The centrifugal force F_c acting downward on the right-hand cables is

$$\begin{aligned} F_c &= \frac{W}{g} \cdot b \cdot \omega^2 = \frac{91808 \times 87.25 \times 5.62 \times 10^{-4} g}{g} \\ &= 4502 \text{ lb} \end{aligned}$$

The total load on the cables will be

$$\begin{aligned} F &= 4502 + 91808 + 2 \times 6100 \\ &= 108,510 \text{ lb} \end{aligned}$$

The capacity of the four cables on the right-hand side is

$$4 \times 30.5 \times 2000 = 244,000 \text{ lb}$$

Then

$$SF = \frac{244,000}{108,510} = 2.25$$

VI.1.3 STRESS ANALYSIS OF LIFTING LINKS

EMERGENCY HOIST

A scale drawing of a lifting link and an insert showing the manner in which it is loaded during an emergency hoist (single failure mode) is shown in Fig. VI.1.3-2.

EFFECTS OF HORIZONTAL LOAD W_H .

$$W_V = 100,000 \cos 19^\circ = 94,552 \text{ lb.}$$

$$W_H = 100,000 \sin 19^\circ = 32,557 \text{ lb}$$

Taking moments about P_1 .

$$P_2 \times 8.484 - 32,557 \times 21.344 = 0$$

$$P_2 = 81907 \text{ lb}$$

$$P_1 = 81907 + 32,557 = 114,464 \text{ lb}$$

Bending moment at section AA (in terms of W_H)

$$\begin{aligned} M_{AA} &= 32,557 \times (26 - 7.03) \\ &= 617,606 \text{ in. lb} \end{aligned}$$

$$\begin{aligned} \frac{I}{c} &= \frac{1}{6} \times 9.5 \times 4^3 \\ &= 25.33 \text{ in}^3 \end{aligned}$$

Fig. VI.1.3-1 Cross-Section AA of Fig. VI.1.3-2

Fig. VI.1.3-2 Lifting Link

$$\sigma_B = \frac{617,606}{25.33} = 24,382 \text{ psi}$$

On account of the contact between the corner of the spacer and the link at point E, it is probable that the stress is much greater than the above value.

Even if yielding occurs, however, it will be localized and of small consequence as far as failure of the link is concerned.

It is unlikely that the spacer will fill all the space between the link and the lifting lug; consequently, bending at Section BB may occur.

Bending moment at Section BB (in terms of P_2)

$$\begin{aligned} M_{BB} &= 81,907 \times 3.828 \\ &= 313,540 \text{ in. lb} \end{aligned}$$

$$\begin{aligned} \frac{I}{c} &= 2 \times \frac{1}{6} \times 3.094 \times 4^2 \\ &= 16.50 \text{ in}^3 \end{aligned}$$

Fig. VI.1.3-3 Cross-Section BB of Fig. VI.1.3-2

$$\sigma_B = \frac{313,540}{16.5} = 19,002 \text{ psi}$$

EFFECTS OF VERTICAL LOAD W_V .

Under this load the link, at each end, is best described as a strip loaded in tension by means of a central pin. For this state no elementary "strength of materials" formula is applicable; an exact solution utilizing a biharmonic stress function and the theory of elasticity has been obtained. ("The Stress

Distribution in a Strip Loaded in Tension by Means of a Central Pin" by P. S. Theocaris. Trans. ASME, Vol. 78 (1956) Applied Mechanics Section.) This solution, however, regards the strip as infinitely long. The influence of the hole on the stress distribution extends to a distance equal to 0.75 times the width of the strip from the center of the hole, whereas on the lifting link this distance is 0.5 times the width of the strip. In addition the pin is assumed to be a close fit in the hole in the mathematical solution while on the lifting link there is a clearance of approximately 1/16 in. between the pin and the hole.

Experimental data based on tests with strain gages on aluminum plates and photoelastic experiments have yielded results that are compatible with one another and also with the theoretical values obtained by Theocaris. Curves summarizing these results and showing the effects of variations in plate length and pin clearance (which increases the stress) are contained in "Photo Elasticity" volume 2, chapter 6, by Max Frocht. In all cases the maximum tensile stress occurs inside the holes on Sections BB and CC; the results are in the form of stress concentration factors that increase the nominal stress at the rim of the hole.

Data for this shape.

At Section CC (small hole)

$$\frac{d}{D} = \frac{2.75}{7.5} = 0.367$$

$$\frac{H}{D} = 0.5$$

Net area through hole

$$= (7.5 - 2.75)4 = 19 \text{ in}^2$$

Fig. VI.1.3-4 Cross-Section CC of Fig. VI.1.3-2

$$\sigma_{\text{nom}} = \frac{94,552}{19} = 4976 \text{ psi}$$

From theoretical results $K_t = 3.2$ (no clearance)From experiments $K_t = 3.6$ (clearance)

$$\sigma_{\text{max}} = 4976 \times 3.6 = 17,914 \text{ psi}$$

At section BB (large hole)

$$\frac{d}{D} = \frac{4.5625}{10.75} = 0.424$$

$$\frac{H}{D} = 0.5$$

Net area through hole

$$= (10.75 - 4.5625)4 = 24.75 \text{ in}^2$$

$$\sigma_{\text{nom}} = \frac{94,552}{24.75} = 3820 \text{ psi}$$

From theoretical results $K_t = 2.9$ (no clearance)From experiments $K_t = 3.3$ (clearance)

$$\sigma_{\text{max}} = 3820 \times 3.3 = 12,606 \text{ psi}$$

At cross section BB this stress will be in the same direction as the tensile bending stress produced by W_H . Then the resultant stress will be

$$\sigma_r = 12,606 + 19,002 = 31,608 \text{ psi}$$

The yield point of the link material has a minimum value of 135,000 psi.

Then

$$SF = \frac{135,000}{31,608} = 4.27$$

NORMAL HOIST

For this case

$$W_V = 50,000 \text{ lb}$$

$$W_H = 0$$

At section CC (small hole)

$$\sigma_{\max} = 17,914 \times \frac{50,000}{94,552} = 9473 \text{ psi}$$

At section BB (large hole)

$$\sigma_{\max} = 12,606 \times \frac{50,000}{94,552} = 6666 \text{ psi}$$

$$SF = \frac{135,000}{9473} = 14.25$$

VI.2.1 Design Criteria and Allowable Stresses

The design criteria and allowable stresses for SLSF-LHM are as identified in Chapter IV.

VI.2.2 Stress Analysis of Loop Handling Machine Cask

The Loop Handling Machine (LHM) Cask is described in considerable detail in Section V.2.2. The cask welds and assembly fasteners are analyzed in Section VI.2.9, below. Stress levels in the outer, weight supporting, shell are considered below.

VI.2.2.1 Analysis of Stress at Lifting Lug-LHM Cask Interface Due to Lifting Loads

Stresses in the outer shell of the LHM Cask due to lifting loads applied by the lifting lugs were examined by means of the finite element computer program SAP-IV using thin plate/shell element (element #6). Both the normal and single failure lifting modes were considered.

Weights for the two loading cases and pertinent dimensions are found in Table II.2-1 and Fig. VI.2.2.2-1(b). The lifting lugs are essentially closed, hollow boxes made of 0.625 in. thick stainless steel plates. This analysis neglects the three open webs which connect the closed box to the bolting flange.

The tractions applied to the cask shell by the lugs are calculated in Section VI.2.2.1 and the modelling techniques and results of the computer analysis are given in Section VI.2.2.2.

From the weld pattern in Fig. VI. 2.2.1.-1

$$I_{yy} = 22\left(\frac{1}{12} \times \frac{3}{8} \times 3^3\right) + 4 \times \frac{3}{8} \times 3 (25^2 + 20^2 + 15^2 + 10^2 + 5^2) \\ + 2 \times \frac{3}{8} \times 15.825 \times 28^2$$

$$= 15,511 \text{ in}^4$$

$$I_{xx} = 22 \times \frac{3}{8} \times 3 \times 7.73^2 + 2 \times \frac{1}{12} \times \frac{3}{8} \times 15.825^3$$

$$= 1725 \text{ in}^4$$

$$J = 15,511 + 1725 = 17,236 \text{ in}^4$$

$$\text{Total weld area } A_w = \frac{3}{8} (22 \times 3 + 2 \times 15.825)$$

$$= 36.6 \text{ in}^2$$

Secondary (torsional) shear stress distributions on weld pattern assuming continuous welds on sides.

$$a = 7.73 \text{ in.}, \quad b = 28 \text{ in.}$$

on sides

$$\tau = \frac{TS}{J} = \frac{T}{J} \frac{a}{\cos \theta}$$

$$\tau_x = \tau \cos \theta = \frac{T}{J} a \text{ (constant)}$$

$$\tau_y = \tau \sin \theta = \frac{\tau_x}{S} = \frac{T}{J} X \text{ (proportional to } X \text{)}$$

WELD PATTERN

FIG. VI.2.2.1-1 LIFTING LUG - LHM CASK INTERFACE

on top and bottom

$$\tau = \frac{T}{J} S = \frac{Tb}{J \sin \theta}$$

$$\tau_x = \tau \cos \theta = \frac{\tau y}{S} = \frac{TSy}{JS}$$

$$= \frac{Ty}{J} \text{ (proportional to } y)$$

$$\tau_y = \tau \sin \theta = \frac{\tau b}{S} = \frac{T}{J} S \frac{b}{S} = \frac{T}{J} b \text{ (constant)}$$

From Fig. VI. 2.2.1.-2 forces and moments on plane of welds:

Normal Loading - Both Lugs

$$F_x = 50,000 - 1219 - 3160 = +45621 \# \uparrow$$

$$F_y = 0$$

$$F_z = 0$$

$$M_x = 0$$

$$M_y = 50,000 \times 16 - 1219 \times 11.5 - 3160 \times 34.9 \\ = +675,698 \text{ in lbs (ccw)}$$

$$M_z = T = -3160 \times 51 = -161,160 \text{ in lbs torque (cw)}$$

Single Failure Mode - Loaded Lug

$$F_x = 94,550 - 1152 - 2988 = +90,410 \# \uparrow$$

$$F_y = 0$$

$$F_z = 32,550 - 1029 - 397 = +31,124 \# \rightarrow$$

$$M_x = 1029 \times 51 = 52,479 \text{ in lbs (cw from bottom)}$$

FIG. VI.2.2.1.-2 LIFTING LUG AND LOAD DIRECTIONS

$$\begin{aligned}
 M_y &= 94,550 \times 16 - 32,550 \times 34.45 - 1152 \times 11.5 \\
 &\quad - 397 \times 15.55 - 2988 \times 34.9 - 1029 \times 3.55 \\
 &= + 264,097 \text{ in lb (ccw)}
 \end{aligned}$$

$$M_z = T = - 2988 \times 51 = - 152,388 \text{ in lbs torque (cw)}$$

Single Failure Mode - Unloaded Lug (no lifting force)

(refer to Fig. VI. 2.2.1.-3)

$$F_x = - 1152 - 2988 = - 4140 \# \quad \downarrow$$

$$F_y = 0$$

$$F_z = + 1029 + 397 = +1426 \# \quad \leftarrow$$

$$M_x = 1029 \times 51 = 52,479 \# \text{ (ccw from bottom)}$$

$$\begin{aligned}
 M_y &= + 1152 \times 11.5 + 2988 \times 34.9 - 397 \times 15.55 \\
 &\quad - 1029 \times 3.55 \\
 &= + 107,703 \text{ in lbs (ccw)}
 \end{aligned}$$

$$M_z = T = - 2988 \times 51 = - 152,388 \text{ in lbs torque (cw)}$$

Normal Loading - Both Lugs

$$\sigma_z = \frac{M_y X}{I_{yy}} = \frac{6.757 \times 10^5}{0.15511 \times 10^5} X = 43.6X$$

$$\text{Primary shear stress, } \tau_p = \frac{F_x}{A_w} = \frac{45621}{36.6} = 1246 \text{ psi}$$

x component of secondary shear stress

$$\tau_s^x = \frac{T}{J} a = \frac{1.6116 \times 10^5}{0.1724 \times 10^5} \times 7.73 = 72 \text{ psi (on sides)}$$

$$\tau_s^x = \frac{T}{J} y = 9.35y \text{ (on top and bottom)}$$

y component of secondary shear stress

Fig. VI.2.2-1-3 Single Failure Position

$$\tau_S^y = \frac{T}{J}x = 9.35x \text{ (on sides)}$$

$$\tau_S^y = \frac{T}{J}b = 9.35 \times 28 = 262 \text{ psi (on top and bottom)}$$

Shear Stresses

At A and C

$$\tau_x = \tau_P + \tau_S^x = 1246 + 723.8 \text{ psi}$$

$$\tau_y = \tau_S^y = 262 \text{ psi}$$

At B and D

$$\tau_x = \tau_P + \tau_S^x = 1246 - 721.74 \text{ psi}$$

$$\tau_y = \tau_S^y = 262 \text{ psi}$$

Normal Stresses (bending only)

$$\sigma_z = 43.6 \times 28 = 1221 \text{ psi}$$

Tension at C and D

Compression at A and B

(See Fig. VI. 2.2.1.-4)

Single Failure Mode-Loaded Lug

$$(\sigma_{z1})_{\max} = \frac{M_y b}{I_{yy}} = \frac{2.641 \times 10^5 \times 28}{0.1551 \times 10^5} = 477 \text{ psi (tension at C and D, compression at A and B)}$$

$$(\sigma_{z2})_{\max} = \frac{M_x a}{I_{xx}} = \frac{5.248 \times 10^4 \times 7.73}{0.1725 \times 10^4} = 235 \text{ psi (tension at A and C, compression at B and D)}$$

$$(\sigma_{z3})_{\max} = \frac{F_z}{A_W} = \frac{31124}{36.6} = 850 \text{ psi (tension at A, B, C, and D)}$$

FIG. VI.2.2.1.-4 NORMAL LOADING - BOTH LUGS

Primary shear stress $\tau_P = \frac{F}{A_W} = \frac{90,410}{36.6} = 2470 \text{ psi}$

$$\tau_S^x = \frac{T}{J} a = \frac{1.524 \times 10^5}{0.1724 \times 10^5} \times 7.73 = 68 \text{ psi (on sides)}$$

$$\tau_S^x = \frac{T}{J} y = 8.84 y \text{ psi (on top and bottom)}$$

$$\tau_S^y = \frac{T}{J} x = 8.84 x \text{ psi (on sides)}$$

$$\tau_S^y = \frac{T}{J} b = 8.84 \times 28 = 248 \text{ psi (on top and bottom)}$$

Shear Stresses

at A and C

$$\tau_x = \tau_P + \tau_S^x = 2470 + 68 = 2538 \text{ psi}$$

$$\tau_y = \tau_S^y = 248 \text{ psi}$$

at B and D

$$\tau_x = \tau_P + \tau_S^x = 2470 - 68 = 2402 \text{ psi}$$

$$\tau_y = \tau_S^y = 248 \text{ psi}$$

Normal Stresses

$$\text{At A, } 850 + 235 - 477 = + 608 \text{ psi}$$

$$\text{At B, } 850 - 477 - 235 = +138 \text{ psi}$$

$$\text{At C, } 850 + 235 + 477 = +1562 \text{ psi}$$

$$\text{At D, } 850 + 477 - 235 = +1092 \text{ psi}$$

(See Fig. VI. 2.2.1.-5)

Single Failure Mode - Unloaded Lug (no lifting force)

$$(\sigma_{zl})_{\max} = \frac{M_y b}{I_{yy}} = \frac{1.077 \times 10^5}{0.1551 \times 10^5} \times 28 = 194 \text{ psi (tension at A and B, compression at C and D)}$$

STRESS DISTRIBUTIONS

FIG. VI.2.2.1.-5 SINGLE FAILURE MODE - LOADED LUG

$$(\sigma_{z2})_{\max} = \frac{M a}{I_{xx}} = \frac{52.48 \times 10^3 \times 7.73}{1.725 \times 10^3} = 235 \text{ psi (tension at B and D)} \\ \text{compression at A and C)}$$

$$(\sigma_{z3}) = \frac{F z}{A_W} = \frac{1426}{36.6} = 39 \text{ psi (tension at A, B, C, and D)}$$

$$\text{Primary shear stress } \tau_P = \frac{F x}{A_W} = \frac{-4140}{36.6} = -113 \text{ psi}$$

$$\tau_S^x = \frac{T}{J} a = \frac{1.524 \times 10^5}{0.1724 \times 10^5} \times 7.73 = 68 \text{ psi (on sides)}$$

$$\tau_S^x = \frac{T}{J} y = 8.8y \text{ (on top and bottom)}$$

$$\tau_S^y = \frac{T}{J} x = 8.8x \text{ (on sides)}$$

$$\tau_S^y = \frac{T}{J} b = 8.8 \times 28 = 246 \text{ psi (on top and bottom)}$$

Shear Stresses

At B and D

$$\tau_x = \tau_P + \tau_S^x = -113 + 68 = -45 \text{ psi}$$

$$\tau_y = \tau_S^y = 246 \text{ psi}$$

at A and C

$$\tau_x = \tau_P + \tau_S^x = -113 - 68 = -181 \text{ psi}$$

$$\tau_y = \tau_S^y = 246 \text{ psi}$$

Normal Stresses

$$\text{At A, } 39 - 194 - 235 = -390 \text{ psi}$$

$$\text{At B, } 39 + 194 + 235 = +468 \text{ psi}$$

$$\text{At C, } 39 - 194 - 235 = -390 \text{ psi}$$

$$\text{At D, } 39 - 194 + 235 = +80 \text{ psi}$$

(See Fig. VI.2.2.1.-6)

FIG. VI.2.2.1.-6 SINGLE FAILURE MODE - UNLOADED LUG

VI. 2.2.2 Finite Element Analysis of LHM Shell Due to Lifting Loads

The LHM cask was modelled as a cylinder with radius = 13.5", thickness = 0.75", length = 112" and fixed ends. The cylinder was divided into a grid of rectangular elements by nodes located at $y = 0", 6", 11", 16", 19.65", 23", 28", 33", 38", 43", 48", 53", 58", 63", 68", 73", 76.35", 80", 85", 91", 98", 105, 112"$; $\theta = 0^\circ, 13^\circ, 25^\circ, 36.87^\circ, 48^\circ, 60^\circ, 72^\circ, 84^\circ, 96^\circ, 108^\circ, 120^\circ, 132^\circ, 143.13^\circ, 155^\circ, 168^\circ, 180^\circ, 192^\circ, 205^\circ, 216.87^\circ, 228^\circ, 240^\circ, 252^\circ, 264^\circ, 276^\circ, 288^\circ, 300^\circ, 312^\circ, 323.13^\circ, 335^\circ, 347^\circ$. The lug/cask intersections were represented by the lines $y = 19.65"$: $0^\circ < \theta < 36.87^\circ, 143.13^\circ < \theta < 216.87^\circ, 323.13^\circ < \theta < 360^\circ$; $19.65" < y < 76.35"$: $\theta = 36.87^\circ, 143.13^\circ, 216.87^\circ, 323.13^\circ$; and $y = 76.35"$: $0^\circ < \theta < 36.87^\circ, 143.13^\circ < \theta < 216.87^\circ, 323.13^\circ < \theta < 360^\circ$. The tractions calculated in the previous section were converted to equivalent line loads by multiplying by the 3/8" weld width. After proper consideration for differences in coordinate systems, this yields the line loads listed in Table VI. 2.2.2-1 where the meaning of sides (1) through (8) are as shown in Fig. VI. 2.2.2-1. The loads along sides (1), (2), (5), (6) were applied in the model by dividing halfway between each pair of neighboring nodes, integrating each force component along the lengths so formed, and applying the resultant as a concentrated force at the node. Along sides (3), (4), (7), (8), the discontinuous weld pattern means that the line loads are developed only along a 3" length, symmetrically surrounding each node. The integrals of the force components were therefore carried out over this 3" length and, again, applied as point forces at the nodes. The concentrated loads so derived are listed in Table VI. 2.2.2-2.

The SAP-IV analysis was carried out with the model and loads described. The fixed ends were achieved by deleting from the analysis all 6 degrees of freedom from the nodes along the cylinder edges. The stress resultants and moments in the ten most highly stressed elements are shown in Tables VI. 2.2.2-3. These are converted to stress components and principle stresses in Table VI. 2.2.2-4.

Fig. VI.2.2.2-1(a) LHM Cask Model for SAP Analysis

5/8" Front Bulk'hd	1,2,6,5,1
S.S.: Neat Side	1,3,5,14,33,45,53,63
	61,51,41,21,1
Far Side	2,4,6,8,34,46,64,62,52,42,22,2
L.W.'s Rib	3,4,24,23,3
Upper Rib	5,6,26,32,31,25,14,5
Upper Rib (D.L.)	31,32,44,43,31
Top Rib	7,8,34,33,15,7
Top Rib (D.L.)	33,36,46,45,33
Midl Bulk'hd	21,22,24,26,25,23,21
Rear Bulk'hd	41,42,44,46,45,43,41
Gusset	47,48,66,65,55,47
Stiffener #1:	51,53,56,55,61
Stiffener #2:	55,56,54,52,55

Welds:	1,2 3/8V	61,62 1/2A
(cut out	One 3,4	62,64 1/2A
one side	Side 5,6 3/8 V	65,66 1/2A
only)	Only 11,12,13(PIN)3/8A	47,65 1/4A
	21,22 3/8V 1/4A	41,45 3/8A
	23,24 1/4A	42,46 3/8A
	25,26 1/4A	47,48 1/4A
	41,42 3/8 1/4A	1,61 3/8V3-5
	43,44 1/4A	2,62 3/8V3-5
	45,46 1/4A	3,23 3/8 3-6
	45,46 1/4A	4,24 3/8 3-6
	51,52 5/16A	5,31,43 3/8A3-6
	1,5 1/4A	6,32,44 3/8A3-6
	2,8 1/4A	15,33,45 3/8A
	21,25 1/4A	8,34,46 3/8A
	22,26 1/4A	
	51,53 3/8A	
	52,54 3/8A	
	55,56 1/4A	

Mode			
01	75.9	11.3	-7.6
2	"	"	+7.6
3	73.6	16.4	-7.6
4	"	"	+7.6
5	69.0	26.0	-7.6
6	"	"	+7.6
7	66.6	31.0	-7.6
8	"	"	+7.6
11	59.0	16.4	0
2	"	26.0	0
3	"	31.0	0
4	52.6	26.0	-7.6
5	"	31.0	-7.6
21	41.4	11.3	-7.6
2	"	"	+
3	"	16.4	-
4	"	"	+
5	"	26.0	-
6	"	"	+
31	36.8	26.0	-
2	36.8	26.0	+
3	38.0	11.0	-
4	"	"	+
41	20.2	11.3	-
2	"	"	+
3	"	22.4	-
4	"	"	+
5	"	23.6	-
6	"	"	+
7	"	13.5	0
8	"	23.6	0
51	10.0	11.3	-7.6
2	"	"	+
3	"	18.2	-
4	"	"	+
5	"	13.5	0
6	"	19.2	0
61	0	11.3	-7.6
2	"	"	+
3	"	15.2	-
4	"	"	+
5	"	13.5	0
6	"	16.3	0

Fig. VI.2.2.2-1(b) LHM Cask Model for SAP Analysis

Fig. VI.2.2.2-1(c) LHM Cask Model for SAP Analysis

Fig. VI.2.2.2-1(d) LHM Cask Model for SAP Analysis

Table VI. 2.2.2-3. Line Loads Representing Traction Applied
To LHM Cask by Lug

Load Case	Side	F_x	F_y	$-z$ (lb/in)
Normal Load	1	98.25	-3.33x	-457.875
	2	-98.25	-3.33x	457.875
	3	-3.466 (y-48)	494.25	16.151 (y-48)
	4	-3.466 (y-48)	440.25	16.151 (y-48)
	5	98.25	-3.33x	457.875
	6	-98.25	-3.33x	-457.875
	7	-3.466 (y-48)	494.25	-16.151 (y-48)
	8	-3.466 (y-48)	440.25	-16.151 (y-48)
Single Failure	1	93.00	-3.148x	-497.625 + 10.88x
	2	-93.00	-3.148x	-139.875 + 10.88x
	3	-3.28 (y-48)	951.75	-406.875 + 6.31 (y-48)
	4	-3.28 (y-48)	900.75	-230.625 + 6.31 (y-48)
	5	92.25	3.148x	-58.125 - 10.88x
	6	-92.25	3.148x	87.375 - 10.88x
	7	-3.254 (y-48)	-67.875	-73.5 + 2.566 (y-48)
	8	-3.254 (y-48)	-16.875	102.75 + 2.566 (y-48)

Table VI. 2.2.2-2 Concentrated Loads Used to Model Line Loads

Node #	y	θ	Normal Node			Single Failure		
			F_x	F_y	F_z	F_x	F_y	F_z
121	19.65	0	275.79	0	1285.26	258.95	0	-163.16
122		13	280.26	26.91	1306.09	263.14	25.41	-253.63
123		25	260.02	49.22	1211.77	244.14	46.48	-314.47
124		36.87	117.65	29.94	548.31	110.47	28.22	-167.33
133		143.13	117.65	29.94	-548.31	111.37	28.22	-693.63
134		155	260.02	49.22	-1211.77	246.12	46.48	-1477.61
135		168	280.26	26.91	-1306.09	265.28	25.41	-1507.30
136		180	275.79	0	-1285.26	261.05	0	-1396.83
137		192	280.26	-26.91	-1306.09	265.28	-25.41	-1331.65
138		205	260.02	-49.22	-1211.77	246.12	-46.48	-1156.32
139		216.87	117.65	-29.94	-548.31	111.37	-28.22	-498.18
148		323.13	117.65	-29.94	548.31	110.47	-25.41	28.12
149		335	260.02	-49.22	1211.77	244.14	-46.48	6.81
150		347	280.26	-26.91	1306.09	263.14	-25.41	-77.97
154	23	36.87	259.95	1482.75	1211.33	244.05	-203.63	-412.95
163		143.13	259.95	1482.75	-1311.33	246.00	2855.25	-1693.88
169		216.87	259.95	1320.75	-1211.33	246.00	2702.25	-1165.13
178		323.13	259.95	1320.75	1211.33	244.05	-50.63	115.8
184	28	36.87	207.96	1482.75	969.06	195.24	-203.63	-374.46
193		143.13	207.96	1482.75	-969.06	196.80	2855.25	-1593.23
199		216.87	207.96	1320.75	-969.06	196.80	2702.25	-1064.48
208		323.13	207.96	1320.75	969.06	195.24	-50.63	154.29
214	33	36.87	155.97	1482.75	726.80	146.43	-203.63	-335.97
223		143.13	155.97	1482.75	-726.80	147.60	2855.25	-1500.08
229		216.87	155.97	1320.75	-726.80	147.60	2702.25	-971.33

Table VI. 2.2.2-2 Concentrated Loads Used to Model Line Loads (cont'd)

Node #	y	θ	Normal Node			Single Failure		
			F_x	F_y	F_z	F_x	F_y	F_z
238	33	323.13	155.97	1320.75	726.80	146.43	-50.63	192.78
244	38	36.87	103.98	1482.75	484.53	97.62	-203.63	-297.48
253		143.13	103.98	1482.75	-484.53	98.40	2855.25	-1406.93
259		216.87	103.98	1320.75	-484.53	98.40	2702.25	-878.18
268		323.13	103.98	1320.75	484.53	97.62	-50.63	231.27
274	43	36.87	51.99	1482.75	242.27	48.81	-203.63	-258.99
283		143.13	51.99	1482.75	-242.27	49.20	2855.25	-1310.78
289		216.87	51.99	1320.75	-242.27	49.20	2702.25	-782.03
298		323.13	51.99	1320.75	242.27	48.81	-50.63	269.76
304	48	36.87	0	1482.75	0	0	-203.63	-220.5
313		143.13	0	1482.75	0	0	2855.25	-1220.63
319		216.87	0	1320.75	0	0	2702.25	-691.88
328		323.13	0	1320.75	0	0	-50.63	308.25
334	53	36.87	-51.99	1482.75	-242.27	-48.81	-203.63	-182.01
343		143.13	-51.99	1482.75	242.27	-49.20	2855.25	-1127.48
349		216.87	-51.99	1320.75	242.27	-49.20	2702.25	-598.75
358		323.13	-51.99	1320.75	-242.27	-48.81	-50.63	346.74
364	58	36.87	-103.98	1482.75	-484.53	-97.62	-203.63	-143.52
373		143.13	-103.98	1482.75	484.53	-98.40	2855.25	-1034.33
379		216.87	-103.98	1320.75	484.53	-98.40	2702.25	-505.58
388		323.13	-103.98	1320.75	-484.53	-97.62	-50.63	385.23
394	63	36.87	-155.97	1482.75	-726.80	-146.43	-203.63	-105.03
403		143.13	-155.97	1482.75	726.80	-147.60	2855.25	-941.18
409		216.87	-155.97	1320.75	726.80	-147.60	2702.25	-412.43

Table VI. 2.2.2-2 Concentrated Loads Used to Model Line Loads (cont'd)

Node #	y	θ	Normal Node			Single Failure		
			F_x	F_y	F_z	F_x	F_y	F_z
418	63	323.13	-155.97	1320.75	-726.80	-146.43	-50.63	423.72
424	68	36.87	-207.96	1482.75	-969.06	-195.24	-203.63	-66.54
433		143.13	-207.96	1482.75	969.06	-196.80	2855.25	-848.03
439		216.87	-207.96	1320.75	969.06	-196.80	2702.25	-319.28
448		323.13	-207.96	1320.75	-969.06	-195.24	-50.63	462.21
454	73	36.87	-259.95	1482.75	-1211.33	-244.05	-203.63	-28.05
463		143.13	-259.95	1482.75	1211.33	-246.00	2855.25	-754.88
469		216.87	-259.95	1320.75	1211.33	-246.00	2702.25	-226.13
478		323.13	-259.95	1320.75	-1211.33	-244.05	-50.63	500.7
481	76.35	0	-275.79	0	-1285.26	-258.95	0	245.26
482		13	-280.26	26.91	-1306.09	-263.14	25.41	161.41
483		25	-260.02	49.22	-1211.77	-244.14	46.48	70.60
484		36.87	-117.65	29.94	-548.31	-110.47	28.22	6.90
493		143.13	-117.65	29.94	+548.31	-111.37	28.22	-265.23
494		155	-260.02	49.22	+1211.77	-246.12	46.48	-530.82
495		168	-280.26	26.91	+1306.09	-265.28	25.41	-486.82
496		180	-275.79	0	+1285.26	-261.05	0	-392.63
497		192	-280.26	-26.91	+1306.09	-265.28	-25.41	-311.16
498		205	-260.02	-49.22	+1211.77	-246.12	-46.48	-209.54
499		216.87	-117.65	-29.94	+548.31	-111.37	-28.22	-69.77
508		323.13	-117.65	-29.94	-548.31	-110.47	-25.41	202.36
509		335	-206.02	-49.22	-1211.77	-244.14	-46.48	391.88
510		347	-280.26	-26.91	-1306.09	-263.14	-25.41	337.07

Table VI. 2.2.2-3 Force Resultants and Moments in High Stress Elements

(a) Normal Lifting Mode								
Element #	Centroid Location		Force Resultants			Moments		
	y	θ	S_{xx}	S_{yy}	S_{xy}	M_{xx}	M_{yy}	M_{xy}
495	78.175"	174°	-980.6	-1817	-92.99	-149.6	-134.8	6.841
465	74.675"	174°	-1057	-1836	-84.44	-125.6	-126.9	40.23
481	78.175"	6.5°	-924.3	-1797	89.54	-141.7	-129.6	-10.71
453	74.675"	30.935°	-770.9	-1443	240.3	-206.1	-122.8	-80.95
452	74.675"	19°	-958.2	-1700	225.1	-167.8	-134	-44.211
451	74.675"	6.5°	-1042	-1819	86.92	-118.1	-122	-7.048
463	74.675"	149.065°	-766.9	-1436	-239.1	-201.8	-120.4	79.4
464	74.675"	161.5°	-960.8	-1705	-228.4	-161.8	-131.6	43.45
496	78.175"	186°	-922.1	-1703	-259.3	-125.5	-121.5	3.139
123	21.325"	30.935°	-790.6	1433	151.3	161.9	108.4	-83.77
(b) Single Failure Mode								
133	21.325"	149.065°	901.2	1580	142.7	267.7	144.8	91.06
134	21.325"	161.5	1108	1506	-61.92	183	143.8	54.81
222	35.5"	137.565°	324.1	1304	630.3	272	95.41	3.565
192	30.5"	137.565°	352.4	1360	679.5	259.5	88.71	-1.298
101	.825"	126°	271	526.5	1496	-82.61	-42.74	-59.62
130	21.325"	114°	258.1	346.5	1226	-173.1	-72.09	-38.85
252	40.5"	137.565°	292.2	1191	563.1	268	90.56	5.654
162	25.5"	137.565°	535.7	1301	712.3	217.3	82.26	-4.57
223	35.5"	149.065°	137.2	1448	-78.55	329.5	113.3	-0.1889
8	3"	90°	353.4	1974	1038	2.365	44.18	1.857

Table VI. 2.2.2-4 Stress Components and Principle Stress in High Stress Elements

Element #	<u>(a) Normal Lifting Mode</u>									
	<u>Outside Surface</u>					<u>Inside Surface</u>				
	σ_x	σ_y	σ_{xy}	σ_1	σ_2	σ_x	σ_y	σ_{xy}	σ_1	σ_2
495	-2849	-3860	-51	-2847	-3863	341	-985	-197	370	-1014
465	-2748	-3801	-3	-2748	-3801	-70	-1095	-222	-24	-1141
481	-2744	-3779	5	-2744	-3779	279	-1014	234	320	-1055
453	-3226	-3235	-543	-2687	-3773	1170	-614	1184	1761	-1204
452	-3068	-3696	-171	-3024	-3740	513	-838	772	863	-1188
451	-2649	-3726	40	-2647	-3728	-130	-1124	190	-95	-1160
463	-3175	-3199	528	-2659	-3715	1130	-631	-1166	1711	-1211
464	-3007	-3678	159	-2971	-3713	445	-870	-768	798	-1223
496	-2568	-3566	-312	-2479	-3656	109	-974	-379	229	-1093
123	2781	3067	-692	3631	2218	-673	754	1095	1348	-1267
	<u>(b) Single Failure Mode</u>									
133	4057	3651	1162	5033	2675	-1654	563	-781	811	-1901
134	3430	3542	502	3991	2981	-475	474	-667	818	-819
222	3333	2757	878	3969	2120	-2469	721	803	912	-2659
192	3238	2759	892	3922	2075	-2298	867	920	1115	-2546
101	-520	244	1359	1274	-1550	1243	1160	2631	3833	-1430
130	-1502	-307	1220	454	-2263	2190	1231	2049	3815	-394
252	3249	2554	811	3784	2019	-2469	622	691	770	-2617
162	3032	2612	901	3747	1897	-1604	857	998	1211	-1958
223	3697	3140	-107	3717	3120	-3331	723	-103	726	-3334
8	496	3103	1403	3715	-116	446	2160	1364	2914	-308

The shell is made of 304 stainless steel (annealed) with a yield strength of $\sigma_u = 30000$ psi.

Results obtained for the Normal Loading Case show that the highest principal stress was:

$$\begin{aligned}\sigma_{\max} &= 3863 \text{ psi (lg)} \\ SF &= \frac{30000}{3863} = 7.7 \text{ (lg)}\end{aligned}$$

The Single Failure Loading Case has somewhat higher stresses at the upper (lifting) lug as expected. The maximum principal stress is

$$\sigma_{\max} = 5033 \text{ psi (lg)}$$

yielding a safety factor

$$SF = \frac{30000}{5033} = 5.9$$

This factor must account, not only for the approximate nature of the analysis, but also, for increased load factor resulting from the dynamics of a single load path failure.

VI.2.2.3 LHM Shell Stresses Due to Loop Supports

In Section II.1 the maximum downward reaction on the loop supports was found to be 8203 lb; weight of the support assembly with actuator is 145 lb. Consequently the total load acting on the support is 8348 lb. It will be assumed that all of this load is carried on one loop support pad as shown in Figures V.2.9.-1 and V.2.9-2, and applied to the shell through a welded attachment with a diameter equal to the minimum dimension of the pad contact, namely 3 inches.

BIJLAARD'S SOLUTION

Radial load $P = 8203 + 145 = 8348$ lb (Sections II.1 & VI.2.10)

Vessel radius $R_m = 8.32$ in $K_n = 1$ (Fig. V.2.9.2)

Attachment radius $r_o = 1.5$ in $K_b = 1$ (Fig. V.2.9.1)

Vessel thickness $T = 0.625$ in

$$\gamma = \frac{R_m}{T} = 13.31, \quad \beta = (0.875) \frac{r_o}{R_m} = 0.158$$

From Fig. VI.2.2.3-3, $\frac{N_\phi}{P/R_m} = 2.0$ (rough estimate using γ and β)

From Fig. VI.2.2.3-1, $\frac{M_\phi}{P} = 0.12$

$$\frac{P}{R_m T} = 1605, \quad \frac{6P}{T^2} = 128,225$$

Circumferential Stresses σ_ϕ

$$K_n \frac{N_\phi}{P/R_m} \cdot \frac{P}{R_m T} = 1 \times 2 \times 1605 = + 3210 \text{ psi (membrane)}$$

$$K_b \frac{M_\phi}{P} \cdot \frac{6P}{T^2} = 1 \times 0.12 \times 128,225 = \pm 15,387 \text{ (bending)}$$

$$\sigma_\phi^1 = + 3210 + 15,387 = + 18,597 \text{ psi (tension at outside of shell)}$$

$$\sigma_\phi^2 = + 3212 - 15,387 = - 12,177 \text{ psi (compression at inside of shell)}$$

Fig. VI.2.2.3-1

Fig. VI.2.2.3-2

-Membrane force $N_\phi / (P/R_m)$ due to an external radial load P on a circular cylinder

Fig. VI.2.2.3-3

Longitudinal Stresses σ_x

From Fig. VI.2.2.3-4, $\frac{Nx}{P/R_m} = 2.4$

From Fig. 2.2.3-2, $\frac{Mx}{P} = 0.093$

$$K_n \frac{N_x}{P/R_m} \cdot \frac{P}{R_m T} = 1 \times 2.4 \times 1605 = + 3852 \text{ psi (membrane)}$$

$$K_b \frac{M_x}{P} \cdot \frac{6P}{T^2} = 1 \times 0.093 \times 128,225 = \pm 11,925 \text{ (bending)}$$

$$\sigma_x^1 = + 3852 + 11,925 = + 15,777 \text{ psi (tension at outside of shell)}$$

$$\sigma_x^2 = + 3852 - 11,925 = - 8073 \text{ psi (compression at inside of shell)}$$

Maximum tensile stress $\sigma_t = + 18,597 \text{ psi}$ (at outside of shell)

Maximum compressive stress $\sigma_c = -12,177 \text{ psi}$ (at inside of shell)

Maximum shear stress = $\frac{18,597}{2} = 9298 \text{ psi}$

Fig. VI.2.2.3-4

VI.2.3 STRESS ANALYSIS OF GRAPPLE

The grapple is a mechanical device that provides means for connecting the chains controlled by the grapple drive to the loop. The grapple drive assembly contains all of the mechanical components required to complete the grapppling operation. After contact between the grapple and the loop, the grapple fingers are brought into a load-bearing position by means of a manually operated gear-driven mechanism. When the loop is suspended from the grapple, the load-carrying components are the shackles, the bolts connecting parts of the housing, and the fingers.

SHACKLE 2

Refer to Fig. V.2.3-1

$$\text{Shear area of } \frac{5}{16} \text{ " pin} = 6 \times \frac{\pi}{4} \times \frac{25}{256} = 0.460 \text{ in}^2$$

$$\text{Load on shackle} = \frac{10000 + 760}{2} = 5380\#$$

$$\text{Direct shear stress in pin} = \frac{5380}{0.46} = \underline{\underline{11696 \text{ psi}}}$$

Chain rating 43000 lbs. Shear stress in pin at rated load

$$\tau = \frac{43000}{0.46} = 93478 \text{ psi}$$

$$\text{Factor of safety} = \frac{93478}{11696} = 8$$

GRAPPLE FINGER (Fig. VI.2.3.-1)

$$\text{Total load on 6 grapple fingers} = 10000\#$$

$$\text{Load/finger} = \frac{10000}{6} \approx 1700\# = W$$

Shear stress on Section C-C

$$\tau = 1.5 \times \frac{1700}{3 \times \frac{3}{4}} = 1133 \text{ psi}$$

Tensile stress on Section A-A

$$\sigma = \frac{1700}{3 \times \frac{1}{2}} = 1133 \text{ psi}$$

SECTION B-B
AND SECTION C-C (APPROX.)

FIG. VI. 2.3-1 GRAPPLE FINGER

The forces at B and D and the reactions at the pivot E are statically indeterminate. It is conceivable that the finger will not make contact with the supporting members at B and D simultaneously. Consequently, the applied load W may be resisted by either a force at B and none at D or a force at D and none at B. These two cases are treated separately below:

Case 1. Assuming $P = 0$ and all reactions at B and E.

$$W \tan 25^\circ \times 4 = F \cos 30^\circ \times 1.25$$

$$F \cos 30^\circ = \frac{1700 \times \tan 25^\circ \times 4}{1.25} = 2537\#$$

$$R = 2537 + 793 = 3330\#$$

This force is carried by the 1/4" pin at E.

3330 Direct shear stress on 1/4" pin at E,

$$\tau = \frac{3330}{2 \times \frac{\pi}{4} \times \frac{1}{16}} = \underline{\underline{33919 \text{ psi}}}$$

$$F \sin 30^\circ = F \cos 30^\circ \tan 30^\circ = 2537 \times \tan 30^\circ = 1464\#$$

$$F_E = 1464 + 1700 = 3164\#$$

This force is carried by a bearing stress in the 7/8" diameter finger pivot. Projected area of pivot = $\frac{0.875 \times 3}{2} = 1.313 \text{ in}^2$

$$\text{Bearing stress} = \frac{3164}{1.313} = \underline{\underline{2411 \text{ psi}}}$$

Case 2. Assuming $F = 0$ and all reactions at D and E.

$$P \times 3 = 793 \times 4, \quad P = \frac{793 \times 4}{3} = 1057\#$$

$$R = 1057 - 793 = 264\#$$

This force is carried by a bearing stress in the 7/8" pivot.

$$\sigma = \frac{264}{2 \times 1.313} = 100 \text{ psi}$$

$$\text{Bearing stress} = \frac{1700}{1.313} = 1295 \text{ psi}$$

The direct shear stress of 33919 psi on the 1/4" pin obtained in Case (1) assumes that all of the grappling reaction is taken at B. Actually, the pin is pressed into a hole of diameter 0.2493-0.2501 in the pivot, but is carried in a hole of diameter 0.253-0.254 in the retainer plate. This gives a minimum clearance of $0.2530 - 0.2501 = 0.0029$ " and a maximum clearance of $0.2540 - 0.2493 = 0.0047$ " between the pin and the hole in the retainer plate. It is claimed that the pin is not expected to carry any load when the loop is being lifted. The actual stress in the pin will depend on the clearances between the finger and the body of the grapple, and in no case will the shear stress exceed 33919 psi.

12 bolts connecting upper body to the middle body.

$$\frac{1}{2} - 13 \text{ NC} \quad \text{Stress area} = 0.1419 \text{ in}^2$$

Stress concentration factor = 3 (estimated)

$$\sigma = \frac{10760 \times 3}{12 \times 0.1419} = 18957 \text{ psi}$$

Bolts (8 of) connecting middle body 10 to lower body 9

$$\frac{5}{8} - 11 \text{ NC} \quad \text{Stress area} = 0.226 \text{ in}^2$$

Stress concentration factor = 3

$$\sigma = \frac{10760 \times 3}{8 \times 0.226} = 17854 \text{ psi}$$

These bolts are made from stainless steel with a yield point estimated to be about 30000 psi when not heat treated. If threads are rolled, the resulting cold working could raise the yield point considerably.

VI.2.4 Stress Analysis of Grapple Drive

This analysis covers the Gear Drive Assembly and other items such as the chain storage reel, spring drum, and sprocket pulley shafts, the pulley block anchor rod to the load cell, and the Belleville springs. Neither the Motor Drive Assembly nor the Loop Location Counter Assembly are analyzed.

VI.2.4.1 Gear Drive Assembly (Refer to Fig. V.2.4-1 for part numbers)

Fig. VI.2.4.1-1 Intermediate Shaft 11

$$\text{Torque on shaft - from pinion 15} = \frac{31847}{2} = 15924 \text{ in lbs}$$

$$\text{P load on sprocket} = \frac{15924 \times 2}{9.567} + 50 = 3379 \text{ #}$$

$$R_2 = \frac{3379 \times 4.5}{3} = 5069\#, \quad R_1 = \frac{3379 \times 7.5}{3} = 8448\#$$

$$\text{Rated radial load of bearing @ } 33 \frac{1}{3} \text{ RPM} = 28880\#$$

At Section A-A, $M = 3379 \times 1.4 = 4731$ in lbs

$T = 15924$ in lbs

$$\frac{I}{c} = \frac{\pi}{64 \times 1.5} \left[3^4 - 0.75^4 \right] = 2.64 \text{ in}^3$$

$$\frac{J}{R} = 2 \times 2.64 = 5.28 \text{ in}^3$$

$$\frac{D}{d} = \frac{3.485}{3} = 1.16$$

Fig. VI.2.4.1-2 Section A-A, Shaft II

$$\begin{aligned} \sigma_{\max} &= \left(\frac{M}{I/c} \right) K_f = \frac{4731}{2.64} \times 3.8 \\ &= 6810 \text{ psi} \end{aligned}$$

$$\frac{r}{d} = \frac{0.015}{3} = 0.005$$

$$K_f = 3.8$$

$$\begin{aligned} \tau_{\max} &= \left(\frac{T}{J/R} \right) K_{fs} = \frac{15924}{5.28} \times 2 \\ &= 6032 \text{ psi} \end{aligned}$$

$$K_{fs} = 2.0 \text{ (est.)}$$

$$\text{Combined } \tau' = \sqrt{\left(\frac{6810}{2} \right)^2 + 6032^2} = \underline{\underline{6927}} \text{ psi}$$

$$\text{Principal } \sigma_1 = \frac{6810}{2} + 6927 = \underline{\underline{10332}} \text{ psi}$$

Using concentration factor for combined bending and torsion in keyseat.

$$\frac{M}{T} = \frac{4731}{15924} = 0.297, \quad K_{fs} = 2.4 \text{ (in keyseat fillet)}$$

$$K_f = 3.13 \text{ (at surface)}$$

$$\text{Nominal } \sigma = \frac{M}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M} \right)^2} \right] = 4042 \text{ psi}$$

$$\sigma_{\max} = 4042 \times 3.13 = \underline{\underline{12652}} \text{ psi}$$

$$\tau_{\max} = 4042 \times 2.4 = \underline{\underline{9701}} \text{ psi}$$

At section B-B, $M = 3379 \times 3.4 = 11489$ in lbs

$T = 15924$ in lbs

$$\frac{I}{c} = \frac{\pi}{64 \times 1.31} \left[2.625^4 - 0.75^4 \right] = 1.768 \text{ in}^3$$

$$\frac{J}{R} = 2 \times 1.768 = 3.536 \text{ in}^3$$

$$\frac{D}{d} = \frac{3.488}{2.626} = 1.327$$

Fig. VI.2.4.1-3

Section B-B, Shaft II

$$\sigma_{\max} = \left(\frac{M}{I/c} \right) K_f = \frac{11489}{1.768} \times 3.6 = 23393 \text{ psi} \quad \frac{r}{d} = \frac{1}{32 \times 2.626}$$

$$\tau_{\max} = \left(\frac{T}{J/R} \right) K_{fs} = \frac{15924}{3.536} \times 2 = 9006 \text{ psi} \quad = 0.0119$$

$$\text{Combined } \tau = \sqrt{\left(\frac{23393}{2} \right)^2 + 9006^2} = 14762 \text{ psi}$$

$$K_f = 3.6$$

$$K_{fs} = 2 \text{ (est.)}$$

$$\text{Principal } \sigma_1 = \frac{23393}{2} + 14762 = 26458 \text{ psi}$$

In keyway to Shaft 11A

$$\frac{T}{M} = \frac{15924}{11489} = 1.386 \quad \frac{M}{T} = 0.721$$

$$\text{Nominal } \sigma = \frac{M}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M} \right)^2} \right]$$

$$= 8802 \text{ psi}$$

$$K_f = 2.75 \text{ (at surface of shaft)}$$

$$K_{fs} = 2.60 \text{ (in fillet)}$$

$$\sigma_{\max} = 8802 \times 2.75 = \underline{\underline{24206 \text{ psi}}}$$

$$\tau_{\max} = 8802 \times 2.06 = \underline{\underline{18132 \text{ psi}}}$$

Key (sprocket to shaft 11)

$\frac{3}{4}$ " SQUARE

Torque = 15924 in lbs

Shaft radius 1.5"

$$\text{Shear stress in Key} = \tau = \frac{15924}{1.5 \times 3 \times \frac{3}{4}} = \underline{\underline{4718 \text{ psi}}}$$

$$\text{Bearing stress on Key} = \sigma = 2 \times 4718 = \underline{\underline{9436 \text{ psi}}}$$

Key (shaft 11 to shaft 11A)

Shaft radius = 1.31"

$\frac{5}{8}$ " SQUARE

$$\text{Shear stress in Key} = \frac{15924}{1.31 \times 0.625 \times 4} = \underline{\underline{4862 \text{ psi}}}$$

$$\text{Bearing stress on Key} = 2 \times 4862 = \underline{\underline{9724 \text{ psi}}}$$

Stress Summary

The maximum normal stress of 26450 psi occurs at section B-B, and the maximum shear stress of 18132 psi occurs in the keyway to shaft 11A. The shaft material is SAE 4340, hardened to 35-38 R_c with a yield point in tension of 138000 psi to 156000 psi. The yield point in shear is assumed to range from 0.57×138000 psi (78660 psi) to 0.57×156000 (88920 psi).

The minimum factor of safety based on the normal stress is therefore

$$\frac{138000}{26450} = 5.2 \text{ and } \frac{78660}{18132} = 4.34 \text{ based on the shear stress. The stresses}$$

in the keys provide higher factors of safety even for low carbon steel.

Gear Drive Assembly. Shaft 11A.

$$T = 15924 \text{ in lbs}$$

$$\text{At hollow section, } J = \frac{\pi}{32} \left[4.134^4 - 2.627^4 \right] = 24 \text{ in}^4$$

$$\frac{J}{R} = \frac{24 \times 2}{4.134} = 11.61 \text{ in}^3$$

$$\tau_{\max} = \left(\frac{T}{J/R} \right) K_{fs} = \frac{15924}{11.61} \times 2 = 2734 \text{ psi}$$

$$\frac{D}{d} = \frac{4.625}{4.134} = 1.12$$

$$\frac{r}{d} = \frac{1}{32 \times 4.134} = 0.0075$$

$$K_{fs} = 2 \text{ (est.)}$$

Fig. VI.2.4.1-4 Shaft 11A on Fig. VI.2.4.1-1

$$\text{At section A-A } \frac{J}{R} = \frac{\pi 2}{32 \times 2.5} [2.5^4 - 0.75^4] = 3.04 \text{ in}^3$$

$$\frac{D}{d} = \frac{3.3125}{2.5} = 1.325, \quad \frac{r}{d} = \frac{0.015}{2.5} = 0.006, \quad K_{fs} = 2 \text{ (est.)}$$

$$\tau_{\max} = \frac{15924}{3.04} \times 2 = 10476 \text{ psi}$$

$$\text{At section B-B, } \frac{J}{R} = \frac{\pi d^3}{16} = \frac{\pi \times 2.5^3}{16} = 3.06 \text{ in}^3$$

$$K_{fs} = 3 \text{ (est.) } \tau_{\max} = \frac{15924}{3.06} \times 3 = 15612 \text{ psi}$$

$$\text{In key } \tau = \frac{15924}{2 \times 1.375 \times 0.312 \times 1.25} = 14847 \text{ psi}$$

$$\text{Bearing } \sigma = 2 \times 14847 = 29694 \text{ psi}$$

Stress Summary

The estimated maximum shear stress in the shaft is 15612 psi at the Woodruff keyways. The material is the same as for shaft 11 and therefore the factor of safety is

$$\frac{78660}{15612} = 5.03$$

The Woodruff key material is SAE 2340 hardened to 35-38 R_c with a yield point of $0.57 \times 147000 = 83790$ psi. The factor of safety based on the keys is

$$\frac{83790}{14847} = 5.64$$

Gear Drive Assembly, Pinion Shaft 14

Fig. VI.2.4.1-5 Pinion Shaft 14

At section C-C, $M_V = 2823 \times 4 = 11292 \text{ in lbs}$

$M_H = 727 \times 4 = 2908 \text{ in lbs}$

$M_R = \sqrt{11292^2 + 2908^2} = 11660 \text{ in lbs}$

$T = 15924 \text{ in lbs} \quad \frac{M}{T} = 0.732$

$\frac{J}{R} = \frac{\pi \times 2.5^3}{16} = 3.06 \text{ in}^3, \quad K_f = 2.75, \quad K_{fs} = 2.06$
(for Keyways)

$\sigma_{\text{nominal}} = \frac{M}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M} \right)^2} \right] = 10260 \text{ psi}$

$\sigma_{\text{max}} = 10260 \times 2.75 = 28216 \text{ psi}$

$\tau_{\text{max}} = 10260 \times 2.06 = 21136 \text{ psi}$

At left end $T = 15924 \text{ in lbs} \quad K_{fs} = 3$

$\tau_{\text{max}} = \frac{15924}{3.06} \times 3 = 15612 \text{ psi}$

Keys at left end (2 of)

Shaft radius = 1.25"

$\tau = \frac{15924}{2 \times 1.25 \times \frac{1}{2} \times 2.125} = \underline{\underline{5995 \text{ psi}}}$

$\sigma \text{ (bearing)} = 2 \times 5994 = \underline{\underline{11990 \text{ psi}}}$

Key in pinion

Shaft radius = 1.25

$\tau = \frac{15924}{1.25 \times 0.625 \times 4} = \underline{\underline{5096 \text{ psi}}}$

$\sigma \text{ (bearing)} = 2 \times 5096 = \underline{\underline{10,191 \text{ psi}}}$

Keys at right end (2 of)

Shaft radius = 1.25"

$$\tau = \frac{15924}{2 \times 1.25 \times 0.625 \times 2.25} = \underline{\underline{4529 \text{ psi}}}$$

$$\sigma \text{ (bearing)} = 2 \times 4529 = \underline{\underline{9058 \text{ psi}}}$$

$$\text{Load on ball bearing 13} = \sqrt{2823^2 + 727^2} = 2915 \#$$

$$\text{Bearing capacity} = 26350 \#$$

$$\text{Load on pillow block 16} = \sqrt{2510^2 + 646^2} = 2592 \#$$

$$\text{Pillow block capacity} = 23126 \#$$

Pinion Shaft 14 Stress Summary

The maximum normal stress in the shaft is 28216 psi and the maximum shear stress is 21136 psi. The material is the same as for shaft 15 and therefore the factor of safety based on normal stress is

$$\frac{138000}{28216} = 4.89$$

and based on shear stress is

$$\frac{78660}{21136} = 3.72$$

The stresses in the Keys provide higher factors of safety even for low carbon steel.

Gear Drive Assembly Drive Shaft 22.

At section A-A (6" from left bearing)

$$M_V = 7573 \times 6 - 5550 \times 1.5 = 37113 \text{ in lbs}$$

$$M_H = 533 \times 6 = 3198 \text{ in lbs}$$

$$M_R = \sqrt{37113^2 + 3198^2} = 37250 \text{ in lbs}$$

$$\frac{I}{c} = \frac{\pi d^3}{32} = \frac{\pi \times 3.1875^3}{32} = 3.179 \text{ in}^3, \frac{J}{R} = 6.358 \text{ in}^3$$

$$\text{Nominal } \sigma = \frac{37250}{3.179} = 11718 \text{ psi}$$

Fig. VI.2.4.1-6 Shaft 22

Fig. VI.2.4.1-7 Section A-A, Shaft 22

$$\frac{D}{d} = 1.25, \frac{r}{d} = \frac{0.125}{3.1875} = 0.039$$

$$q = 1, K_t = 2.17 = K_f \text{ (Peterson - Fig. 78)}$$

$$3.1875 = d$$

$$\sigma_{\max} = 11718 \times 2.17 = 25428 \text{ psi}$$

$$\text{Torque } T = 15924 \text{ in lbs}$$

$$\text{Nominal } \tau = \frac{15924}{6.358} = 2505 \text{ psi}$$

$$K_{fs} = 1.65 \text{ (Peterson - Fig. 79)}, \tau_{\max} = 2505 \times 1.65 = 4133 \text{ psi}$$

$$\text{Combined } \tau = \sqrt{\left(\frac{25428}{2}\right)^2 + 4133^2} = \underline{\underline{13369}} \text{ psi}$$

$$\text{Principal } \sigma_1 = \frac{25428}{2} + 13369 = \underline{\underline{26083}} \text{ psi}$$

At left sprocket keyway

$$\frac{T}{M_R} = \frac{15924}{37250} = 0.427$$

$$\begin{aligned} \text{Nominal } \sigma &= \frac{M_R}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M}\right)^2} \right] \\ &= 12230 \text{ psi} \end{aligned}$$

For combined bending and torsion (Peterson Fig. 183a)

$$K_f = 2.28 \text{ (at fillet in keyway)}$$

$$K_{fs} = 1.52 \text{ (at fillet in keyway)}$$

$$\sigma_{\max} = 12230 \times 2.24 = 27395 \text{ psi}$$

$$\tau_{\max} = 12230 \times 1.41 = 17244 \text{ psi}$$

$$\text{At gear, } M_H = 7573 \times 13 - 5550 \times 8.5 = 51274 \text{ in lbs}$$

$$M_V = 533 \times 13 = 6929 \text{ in lbs}$$

$$M_R = \sqrt{51274^2 + 6929^2} = 51740 \text{ in lbs}$$

$$T = 31848 \text{ in lbs, } \frac{T}{M} = \frac{31848}{51740} = 0.616$$

$$\frac{J}{R} = \frac{\pi d^3}{16} = \frac{\pi \times 4^3}{16} = 12.57 \text{ in}^3$$

$$\begin{aligned} \text{Nominal } \sigma &= \frac{M_R}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M} \right)^2} \right] \\ &= \frac{51740}{12.57} \times 2.174 = 8949 \text{ psi} \end{aligned}$$

From Fig. 183a (Peterson) for combined bending and torsion, $K_f = 2.28$,

$K_{fs} = 1.56$ (both in Keyseat fillet)

$$\sigma_{\max} = 8949 \times 2.28 = \underline{20404} \text{ psi}$$

$$\tau_{\max} = 8949 \times 1.56 = \underline{13960} \text{ psi}$$

At section B-B (5.5 inches from right bearing)

$$M_V = 8905 \times 5.5 - 5550 \times 1.25 = 42040 \text{ in lbs}$$

$$M_H = 840 \times 5.5 = 4620 \text{ in lbs. } T = 15924 \text{ in lbs}$$

$$M_R = \sqrt{42040^2 + 4620^2} = 42293 \text{ in lbs}$$

$$\frac{I}{c} = 3.179 \text{ in}^3, \frac{J}{R} = 6.358 \text{ in}^3, K_f = 2.17, K_{fs} = 1.65 \text{ (same as left e)}$$

$$\sigma_{\max} = \frac{42293}{3.179} \times 2.17 = 28869 \text{ psi}$$

$$\tau_{\max} = \frac{15924}{6.358} \times 1.65 = 4133 \text{ psi}$$

$$\text{Combined } \tau = \sqrt{\left(\frac{28869}{2} \right)^2 + 4133^2} = \underline{15014} \text{ psi}$$

$$\text{Principal } \sigma_1 = \frac{28869}{2} + 15014 = \underline{29448} \text{ psi}$$

At right sprocket Keyway

$$\frac{T}{M_R} = \frac{15924}{42293} = 0.376$$

$$\begin{aligned} \text{Nominal } \sigma &= \frac{M_R}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M} \right)^2} \right] \\ &= 13760 \end{aligned}$$

$$\begin{aligned}
 K_f &= 2.23 \\
 K_{fs} &= 1.36
 \end{aligned}
 \left. \vphantom{\begin{aligned} K_f &= 2.23 \\ K_{fs} &= 1.36 \end{aligned}} \right\} \text{both in keyway fillet}$$

$$\begin{aligned}
 \sigma_{\max} &= 13760 \times 2.23 = 30685 \text{ psi} \\
 \tau_{\max} &= 13760 \times 1.36 = 18714 \text{ psi}
 \end{aligned}$$

Shaft 46 Keys

Key 44 in gear 45

Torque 31848 in lbs Shaft radius = 2"

$$\text{Shear stress in key} = \tau = \frac{31848}{2 \times 4 \times 1} = 3981 \text{ psi}$$

$$\text{Bearing stress on key} = \sigma = \frac{31848}{2 \times 4 \times 1/2} = 7962 \text{ psi}$$

Key in sprockets 23

Torque - 15924 in lbs , Shaft radius = 1.594 "

$$\text{Shear stress in key} = \tau = \frac{15924}{1.594 \times 3 \times 3/4} = 4440 \text{ psi}$$

$$\text{Bearing stress on key} = \sigma = 2 \times 4440 = 8880 \text{ psi}$$

The maximum normal stress (30685 psi) and the maximum shear stress (18714 psi) occur at the right sprocket keyway. The material is the same as for shaft 11 and therefore the factor of safety based on normal stress is

$$\frac{138000}{30685} = 4.49$$

and based on shear stress is

$$\frac{78660}{18714} = 4.2$$

The stresses in the keys are low and provide higher factors of safety even for low carbon steel.

$$m = 2 \quad P = 3 \quad \phi = 14\frac{1}{2}^\circ \quad F = 3'' \quad \frac{F}{d} = \frac{3}{6} = \frac{1}{2}$$

$$N_P = 18 \quad d = 6''$$

Fig. VI.2.4.1-8 Gear 29 and Pinion 15

Beam Strength

For pinion, Lewis form factor $y = 0.086$, $Y = \pi \times y = 0.270$

To approximate American Gear Manufacturers (A.G.M.A.) geometry factor J (which includes stress concentration factor and effect of radial force F_R and is not available for $\phi = 14\frac{1}{2}^\circ$) compare Lewis Y and A.G.M.A. J for 18 teeth and $\phi = 20^\circ$

$$\frac{J}{Y} = \frac{0.232}{0.308} = 0.753 \quad (\phi = 20^\circ)$$

$$J \text{ for } \phi = 14\frac{1}{2}^\circ = 0.270 \times 0.753 = 0.203 \text{ (est.)}$$

A.G.M.A. formula for tensile stress S_t

$$S_t = \frac{W_t}{K_V} \cdot K_o \frac{P}{F} \cdot \frac{1}{J} K_m$$

Assume dynamic factor $K_V = \frac{50}{50 + \sqrt{V}}$ (for hobbed or shaped spur gears)

$$V = \frac{\pi \times 6 \times 12.5}{12} = 19.6 \text{ ft/min}, \quad \sqrt{V} = 4.43$$

$$K_V = \frac{50}{54.43} = 0.919$$

K_o = overload factor = 1 for uniform power source and load.

K_m = load distribution factor = 1.4 for good bearings and mounting.

$$S_t = \frac{5308 \times 1 \times 3 \times 1.4}{0.919 \times 3 \times 0.203} = 39833 \text{ psi}$$

For R_c 42-44, A.G.M.A. allowable tensile stress range is 42000-56000 psi

Surface Durability

$$S_c = C_P \sqrt{\frac{W_t C_o C_m}{c_v d F I}} \quad m = \frac{N_G}{N_P} = \frac{36}{18} = 2$$

$$I = \frac{m \sin 2\phi}{4(m+1)} = 0.081$$

$$c_v = K_v = 0.919, c_o = K_o = 1, c_m = K_m = 1.4.$$

$$C_P = \frac{1}{\sqrt{\pi \left\{ \frac{(1-u_P)^2}{E_P} + \frac{(1-u_G)^2}{E_G} \right\}}} = 2300 \text{ (for steel gear and pinion)}$$

$$S_c = 2300 \sqrt{\frac{5308 \times 1 \times 1.4}{0.919 \times 6 \times 3 \times 0.081}} = 171285 \text{ psi}$$

For R_c 42-44 A.G.M.A. allowable compressive stress range is 160000-180000 psi.

VI.2.4.2, OTHER ITEMS

STORAGE DRUM SHAFT

Fig. VI.2.4.2-1 Storage Drum Shaft

$$\begin{aligned}\text{Drum weight} &= \frac{\pi}{4} [5.813^2 \times 0.25 + 4.062^2 \times 2.5625] 0.282 \\ &= 11.24\#\end{aligned}$$

$$\text{Spring weight} = 0.032 \times 2.5 \times 409 \times 0.282 = 9\#$$

$$P = 11.24 + 9 = 20.24\#$$

$$M = 20.24 \times 2 = 40.48 \text{ in lbs}$$

$$\frac{I}{c} = \frac{\pi}{32} \times 0.875^3 = 0.0658 \text{ in}^3$$

$$\sigma = \frac{40.48}{0.0658} = 615 \text{ psi}$$

Material is 304 stainless steel.

$$\sigma_y = 30000 \text{ psi}$$

Factor of safety

$$= \frac{30000}{615} = 49$$

Fig. VI.2.4.2-2 Chain Reel Shaft 28

$$\begin{aligned}\text{Chain reel weight} &= \frac{\pi}{4} [35^2 \times 0.5 + 6^2 \times 4.5] 0.282 \\ &= 172\#\end{aligned}$$

Assume 300# of chain on drum

$$\text{Then } P = 172 + 300 = 472\#$$

$$\begin{aligned}\text{Torque spring drum weight} &= \frac{\pi}{4} [8.5^2 + 0.25 + 6.76^2 \times 2.593] 0.282 \\ &= 30\#\end{aligned}$$

$$\text{Spring weight} = 0.032 \times 2.5 \times 409 \times 0.282 = 9\#$$

$$\text{Then } Q = 30 + 9 = 39\#$$

$$R_2 = 0.61 \times 474 + 0.2 \times 39 = 295.7\#$$

$$R_1 = 215.3\#$$

$$M_{\max} = 295.7 \times 4 = 1183 \text{ in lbs}$$

$$T = 344 \text{ in lbs (from negator)}$$

$$\frac{I}{c} = \frac{\pi d^3}{32} = \frac{\pi \times 1.5^3}{32} = 0.331 \text{ in}^3$$

$$\frac{J}{R} = 0.662 \text{ in}^3$$

$$\frac{T}{M} = \frac{344}{1183} = 0.29$$

$$\left(\frac{T}{M}\right)^2 = 0.084$$

$$\text{Nominal } \sigma = \frac{M}{J/R} \left[1 + \sqrt{1 + \left(\frac{T}{M} \right)^2} \right] = \frac{1183}{0.662} \times 2.04 = 3645 \text{ psi}$$

$$K_t = 2.2 = K_f, \text{ in keyway fillet} = 3645 \text{ psi}$$

$$K_{ts} = 1.3 = K_f \text{ in keyway fillet}$$

$$\sigma_{\max} = 3645 \times 2.2 = 8019$$

$$\tau_{\max} = 3645 \times 1.3 = 4739 \text{ psi}$$

Material is ASTM A-108 cold drawn

$$\sigma_y = 70000 \text{ psi,}$$

$$\tau_y = 0.57 \times 70000 = 49000 \text{ psi}$$

$$\text{Factor of safety} = \frac{70000}{10206} = 6.85 \text{ based on normal stress and}$$

$$\frac{40000}{6561} = 6.09 \text{ based on shear stress.}$$

SPROCKET PULLEY SHAFT IN PULLEY BLOCK 24

11060[#] (CHAIN LOAD ON SPROCKET)

Fig. VI.2.4.2-3

$$\frac{I}{c} = \frac{\pi}{32} \times 2.56^3 = 1.65 \text{ in}^3$$

Sprocket Pulley Shaft in Pulley Block

$$M = 5530 \times 3.5 = 19355 \text{ in lbs}$$

$$K_f = 2.14 \text{ (for Keyway)}$$

$$\sigma_{\max} = \frac{19355}{1.65} \times 2.14 = 25103 \text{ in lbs}$$

Material is SAE 4340, R_c 30, $\sigma_y = 119000$ psi

$$\text{Factor of safety} = \frac{119000}{25103} = 4.74$$

PULLEY BLOCK ANCHOR ROD 26 (TO LOAD CELL)

$1\frac{3}{4}$ " - 12 UN threads 3" long

$$\text{Root diameter} = 1.6478"$$

$$\text{Root area} = 2.1325 \text{ in}^2$$

Chain load on pulley block = 11060#

$$\text{At root } \sigma = \frac{11060}{2.1325} = 5186 \text{ psi}$$

$$\text{Shear } \tau = \frac{11060}{3\pi 1.6478} = 712 \text{ psi}$$

$1\frac{1}{2}$ - 12 UNF threads 4" long

$$\text{Root diameter} = 1.398"$$

$$\text{Stress area} = 1.581 \text{ in}^2$$

$$\text{Stress concentration factor} = 3$$

$$\text{At root } \sigma = \frac{11060}{1.581} \times 3 = 20985 \text{ psi}$$

$$\text{Shear } \tau = \frac{11060}{4\pi 1.390} = 629 \text{ psi}$$

Material is ASTM 4340, R_c 30

$$\sigma_y = 119000 \text{ psi}$$

$$\tau_y = 0.57 \times 119000$$

$$= 67830 \text{ psi}$$

Factor of safety based on normal stress

$$= \frac{119000}{20985} = 5.7 \text{ and } \frac{67830}{629} = 108 \text{ based on shear stress.}$$

COMPRESSION SPRING

$$\text{Mean diameter } D = 2.0625 + 0.312 = 2.3745''$$

$$\text{Spring index } C = \frac{D}{d} = \frac{2.3745}{0.312} = 7.61$$

$$\text{Wahl factor } K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = 1.19$$

$$\text{Solid height} = 6 \times 0.312 = 1.872''$$

Because spring is confined in a space in gear drive box only 1.75 inches long it could conceivably be compressed solid, resulting in a deflection of

$$\Delta = 3.5 - 1.872 = 1.628''$$

$$P = \frac{\Delta G d}{8 C^3 N}, \quad \tau = K \frac{8 C P}{\pi d^2}$$

or

$$\tau = \frac{K \Delta G}{\pi d C^2 N} = \frac{K \Delta G}{\pi d C^2 N} = \frac{K \Delta G}{\pi d C N}$$

$$= \frac{1.19 \times 1.628 \times 11.5 \times 10^6}{\pi \times 2.374 \times 7.61 \times 4} = 98114 \text{ psi}$$

$$\text{For SAE 6150 steel } \tau_y = 90000 \text{ psi}$$

DISK SPRING (BELOW LOAD CELL)

$$t^3 = 0.1575$$

$$t^3 = 0.003907$$

$$E' = \frac{E}{1-\mu^2}$$

$$a = 2.76$$

$$a^2 = 7.618$$

$$b = 1.595$$

$$\ln \frac{a}{b} = \ln 1.73$$

$$= 0.548$$

Fig. VI.2.4.2-4 Disk Spring

$$\text{For steel } E = 30 \times 10^6 \text{ psi}$$

$$\mu = 0.3$$

$$E' = 32.97 \times 10^6$$

$$P = \frac{4E'y}{Ma^2} [(h - 0.5y)(h - y)t + t^3]$$

$$M = \frac{6}{\pi \ln \frac{a}{b}} \left[\frac{\frac{a}{b} - 1}{\frac{a}{b}} \right]^2 = \frac{6}{1.72} \times \left(\frac{0.73}{1.73} \right)^2 = 0.621$$

$$\text{For } y = \frac{h}{2} = 0.0334$$

$$P = \frac{4 \times 32.97 \times 10^6 \times 0.0334}{0.621 \times 7.618} [(0.0669 - 0.0167)(0.0334) 0.1575 + 0.0039]$$

$$= 3884\# \text{ (specification calls for 3980\#)}$$

$$\sigma = \frac{4E'y}{Ma^2} [c_1 (h - 0.5y) + c_2 t]$$

$$c_1 = \frac{6}{\pi \ln \frac{a}{b}} \left[\frac{\frac{a}{b} - 1}{\ln \frac{a}{b}} - 1 \right] = 3.488 \left[\frac{0.73}{0.548} - 1 \right] = 1.158$$

$$c_2 = \frac{6}{\pi \ln \frac{a}{b}} \left[\frac{\frac{a}{b} - 1}{2} \right] = 3.488 \times \frac{0.73}{2} = 1.273$$

$$\text{For } y = \frac{h}{2} = 0.0334$$

$$\sigma = 931093 [1.158 \times 0.0502 + 1.273 \times 0.1575]$$

$$= 240808 \text{ psi (compression at A)}$$

Load to compress spring flat ($y = h$)

$$P_o = \frac{4E'ht^3}{Ma^2} = \frac{4 \times 32.97 \times 10^6 \times 0.0669 \times 0.003907}{0.621 \times 7.618}$$

$$= 7286\# \text{ (per spring)}$$

$$\begin{aligned}
 S_o &= \frac{4E'h}{Ma^2} \left[\frac{c_1 h}{2} + c_2 t \right] \\
 &= \frac{4 \times 32.97 \times 10^6 \times 0.0669}{0.621 \times 7.618} (1.158 \times 0.0334 + 1.273 \times 0.1575) \\
 &= 446291 \text{ psi}
 \end{aligned}$$

Because the springs are stacked 3 in parallel, the total flattening load will be $3 \times 7286 = 21858 \text{ lbs}$

Fig. VI.2.4.2-5 Disk Spring Assembly

There are 24 springs in a parallel-series arrangement. The free height of the assembly $L = m(nt + h)$ in which

m = number of parallel spring sets in a series.

n = number of springs in each parallel set.

$$L = 8(3 \times 0.1575 + 0.0669) = 4.315$$

$$\text{Assembled height (at } y = \frac{h}{2}) = 4.315 - 8 \times 0.0334 = 4.05''$$

Material is chrome-vanadium spring steel ($50 C_r V_4$). Permissible stresses 350000 - 420000 psi.

VI. 2.5 Stress Analysis of Grapple Drive Mounting

The box containing the grapple drive is mounted on the LHM shell by means of 26 brackets that are welded to the shell and bolted to the grapple drive box. (refer to Figures V. 2.5-1 and V. 2.5-2.

Bolt Stresses

The bolt pattern is shown in Fig. VI.2.5-1. For the 5/8-11 UNC bolts the stress area is 0.226 in^2 and the yield strength of the bolt material $S_y = 100,000 \text{ psi}$. The stress in the bolts will depend almost entirely on the pre-load or tightening stress. Lacking information on the tightening torques and corresponding loads that were actually applied to the bolts when installed, it is reasonable to assume that they were great enough to prevent the brackets from separating from the grapple box. To calculate the tightening load required to prevent separation it will be necessary to estimate the stiffnesses of the bolt, (K_B) and the joint (K_J) and the magnitude of the applied load.

To estimate the stiffnesses it will be assumed that the joint is a hollow cylinder with a hole the same size as the bolt, and an outside diameter three times the bolt diameter. Then if d is the bolt diameter

$$\frac{K_J}{K_B} = \frac{(3d)^2 - d^2}{d^2} = 8$$

From Fig. VI.2.5-1 the maximum applied load P_1 is given by

$$P_1 = \frac{181,170}{2 \times 91} \approx 1000 \text{ lb}$$

There are two bolts at the P_1 location so that the applied load P per bolt is 500 lbs.

The compressive load on the joint F_J produced by the tightening load F_0 is given by

$$F_J = F_0 - \frac{P}{1 + \frac{K_B}{K_J}}$$

This will have a value of zero (i.e. the joint will separate) when

$$F_O = \frac{P}{1 + \frac{K_B}{K_J}} = \frac{500}{1 + \frac{1}{8}} = 444 \text{ lb.}$$

This being less than the applied load, all of the latter will act on the bolt and the stress in the bolt will be

$$\sigma = \frac{500}{0.226} = 2212 \text{ psi}$$

which is very small considering the high yield strength of the bolt material.

For a more realistic approach, assume a factor of safety of 5 on the yield strength; then the tightening stress will be

$$\sigma_O = \frac{100,000}{5} = 20,000 \text{ psi}$$

$$\text{and } F_O = 20,000 \times 0.226 = 4520 \text{ lb.}$$

This corresponds, approximately, to a tightening torque of

$$T = 0.2 \times 4520 \times 5/8 = 565 \text{ in.lbs.} = 47 \text{ ft.lbs.}$$

which is a reasonable value for a 5/8 bolt. The load on the bolt is given by

$$F_B = F_O + \frac{P}{1 + \frac{K_J}{K_B}} = 4520 + \frac{500}{9} = 4576 \text{ lb.}$$

and the resultant stress in the bolt

$$\sigma_B = \frac{4576}{0.226} = 20,247 \text{ psi}$$

which is only 1.24% greater than the tightening stress. The factor of safety

$$S.F. = \frac{100,000}{20,247} = 4.9$$

For higher values of the tightening torque, the resultant bolt stress would be less than 1.24% greater than the tightening stress. The total tightening load on the box is

$$26 \times 4520 = 117520 \text{ lb.}$$

The vertical friction force created by this load, assuming $\mu = 0.15$ is

$$V = 117,520 \times 0.15 = 17628 \text{ lb.}$$

which is 2.63 times the vertical weight load W.

Weld Stresses

The weld pattern is similar to the bolt pattern shown in Fig. VI.2.5-1. On each bolt location there are two 5/16 in. welds 2 inches long and the distance from the neutral axis to the remote ends of the four welds in the P_1 position is 37 inches. The moment arm of the weight W is now 29 inches.

$$\text{Throat area of 1 weld} = 0.3125 \times 0.707 \times 2 = 0.442 \text{ in}^2$$

$$\text{Throat area of 4 welds} = 4 \times 0.442 = 1.768 \text{ in}^2$$

$$\text{Throat area of all welds} = 52 \times 0.442 = 22.984 \text{ in}^2$$

From Fig. VI.2.5-1

$$M_{xx} = 6710 \times 29 = 194590 \text{ in.lb}$$

$$P_1 = \frac{194590}{2 \times 91} = 1069 \text{ lb}$$

Average weld tensile stress at P_1 position

$$\sigma = \frac{1069}{1.768} = 604 \text{ psi}$$

$$\sigma_{\max} = 604 \times \frac{37}{36} = 621 \text{ psi}$$

Direct shear stress

$$\tau = \frac{6710}{22.981} = 292 \text{ psi}$$

$$\tau_{\max} = \sqrt{\left(\frac{621}{2}\right)^2 + 292^2} = 426 \text{ psi}$$

$$\text{For 304 annealed S.S., } S_{SY} = \frac{2 \times 30,000}{3} = 20,000 \text{ psi}$$

$$\text{S.F.} = \frac{20,000}{426} = 47$$

FIG. VI.2.5.-1 BOLT PATTERN - GRAPPLE DRIVE MOUNTING

VI.2.6 Stress Analysis of Lifting Lug

A stress analysis of the lifting lug has been carried out for a normal condition and an emergency condition. The normal condition is defined as that condition when the LHM is lifted by means of two lifting lugs from the vertical position through a hoist system composed of crane, crane cables, shackles and hooks. The lifting lug is composed of several plates welded together and then welded to the body of the LHM as shown in Section V.2.6. A pin is welded to three parallel plates which are welded to two end transverse plates which then are welded to the shell body of LHM. A schematic of the lifting lug pin supports is shown in Fig. V.2.6-1.

In Fig. VI.2.6-1, the pin is shown fixed at A and C where it is welded continuously with support plates. However, at the point B in this figure it is considered pin-supported since at this function the weld size is smaller than that at A and C. A lifting link is located between plates B and C. At the pin end, the lifting link has some partial rotary motion whereas at the far end it has another hole in it to facilitate handling by crane hooks. The emergency condition is the one associated with the worst single failure event on the LHM when one of the two crane cables fail and the LHM is hung by a single cable as shown in Fig. VI.2.6.4-1. The lifting link in the span between A and B has clearances with both the plates A and B.

In order to investigate a complete mechanism of a lifting lug, it was decided to conduct the analysis in several stages. Due to the persence of gaps between the lifting link and the supports, it was impossible to model the lifting lug for the computerized finite element analysis. In order not to lose the physical insight of the problem, it was considered essential to perform a step by step analysis.

First, sections VI.2.6.1 to VI.2.6.3 are devoted to investigate the performance of a lifting lug without consideration of the support vertical plate stiffness in the analysis. This allowed precise investigation of gap effects both for the normal condition loading, Section VI.2.6.3 and the emergency condition, section VI.2.6.4.

In section VI.2.6.2, no gap effects were included so as to provide an opportunity to compare the analysis including gap effects. Section VI.2.6.1 is concerned only with section properties and an overall stress situation.

In the analysis of section VI.2.6.2 and VI.2.6.3, a slope deflection method was used to compute moment distributions.

In the analysis of section VI.2.6.4, the plate stiffnesses were accounted and a moment-distribution method was used for the purpose of validation of the analysis conclusions of sections VI.2.6.2 and VI.2.6.3.

In section VI.2.6.6, a specific recommendation is developed for the modification of the design of the lifting lug to handle an emergency hoist condition.

VI.2.6.1 Lifting Lug Cross-section (See Fig. VI.2.6.1-1)

Location of Neutral Axis in x-x plane

$$\begin{aligned}
 &= \frac{\int x \, dA}{\int dA} \\
 &= \frac{\left\{ \begin{aligned} &2 \times 210.625 \times 10.5 + 14.75 \times 0.625 \times (0.3125 + 0.5) \\ &+ 14.75 \times 0.625 \times (2.25 + 0.625 + 0.5) + 14.75 \times 0.625 \\ &\times (7.25 + 0.625 + 4.5 + 0.625 + 0.5) \end{aligned} \right\}}{2 \times 21 \times 0.625 + 3 \times 14.75 \times 0.625} \\
 &= \frac{438.68}{53.0625} = 8.267 \text{ in}
 \end{aligned}$$

Location of Neutral Axis in Y-Y plane

$$= \frac{\int y \, dA}{\int dA} = 8.0 \text{ in. (since symmetrical)}$$

Moment of Inertia in xx-plane about N.A

$$\begin{aligned}
 I_{xx} &= \frac{2 \times 0.625 \times 21^3}{12} + 2 \times 21 \times 0.625 \times (10.5 - 8.267)^2 \\
 &\quad + \frac{14.75 \times (0.625)^3}{12} + 14.75 \times 0.625 \times 6.045^2 \\
 &\quad + \frac{14.75 \times (0.625)^3}{12} + 14.75 \times 0.625 \times 1.8275^2 \\
 &\quad + \frac{14.75 \times (0.625)^3}{12} + 14.75 \times 0.625 \times 6.954^2 \\
 &= 1910 \text{ in}^4
 \end{aligned}$$

[illegible]

About Z-Z Axis

Part	Area	Z	AZ	$(Z-\bar{Z})^2$	$A(Z-\bar{Z})^2$
1	12.50	78.9375	986.72	2348.13	29351.6
2	95.78	40.3125	3861.13	96.68	9269.6
3	9.22	40.8125	376.29	106.76	984.3
4	9.22	22.9375	211.48	56.89	524.5
5	9.22	12.3125	113.52	330.06	3043.2
6	6.25	17.625	110.16	165.25	1032.8
7	6.25	7.000	43.75	551.31	3445.7
8	40.00	1.000	40.00	869.07	34762.8
	<u>188.44</u>		<u>5743.05</u>		<u>82414.5 in⁴</u>

$$\bar{Z} = \frac{5743.05}{188.44} = 30.48''$$

$$\frac{1}{12} \left(20 \times 0.244 + 1.25 \times 449895.3 + 3 \times 14.75 \times 0.244 + 2 \times 0.625 \times 1000 + 20 \times 8 \right)$$

$$= 46982.9 \text{ in}^4$$

$$I_{zz} = 82414.5 + 46982.7 = 129397 \text{ in}^4$$

Total moment under normal hoist

$$= 50000 \times 14.9375 = 746875 \text{ in lbs.}$$

Bending stress at the top weld location in the lifting lug.

$$\sigma_B = \frac{746875 \times 48.77}{129397} = 281.5 \text{ psi at top surface.}$$

Bending stress at the bottom weld location in the lifting lug

$$\sigma_B = \frac{746875 \times 30.48}{129397} = 175.9 \text{ psi at bottom surface.}$$

Moment arm used in the above analysis is 14.9375. The above analysis gives the stresses at the junction of the lifting lug and LHM shell in the lifting lug. In a previous section, the stress analysis of the LHM shell and welds is included. The detailed load distribution in the lifting lugs is given in the following section.

VI.2.6.2 Basic Loading of Lifting Lug

Fig. VI.2.6.2-1 Loading of Lifting Lug - No Gap Considered

Let FEM represent the fixed end moment

$$FEM_{AB} = FEM_{BA} = 0$$

$$FEM_{BC} = -\frac{wl_2^2}{12}$$

$$FEM_{CB} = +\frac{wl_2^2}{12}$$

Then

$$M_{AB} = \frac{2EI}{l_1} \theta_B$$

$$M_{BA} = \frac{4EI}{l_1} \theta_B$$

$$M_{BC} = \frac{4EI}{l_2} \theta_B - \frac{wl_2^2}{12}$$

$$M_{CB} = \frac{2EI}{l_2} \theta_B + \frac{wl_2^2}{12}$$

$$M_{BA} + M_{BC} = 0 = \left(\frac{4EI}{l_1} + \frac{4EI}{l_2} \right) \theta_B - \frac{wl_2^2}{12} = 0$$

or

$$\theta_B = \frac{\frac{wl_2^2}{12}}{4EI \left(\frac{1}{l_1} + \frac{1}{l_2} \right)} = \frac{wl_1 l_2^3}{48EI(l_1 + l_2)}$$

$$M_{AB} = \frac{wl_2^3}{24(l_1 + l_2)}$$

$$M_{BA} = \frac{wl_2^3}{12(l_1 + l_2)}, \quad M_{BC} = -M_{BA}$$

$$M_{CB} = \frac{wl_1 l_2^2}{24(l_1 + l_2)} + \frac{wl_2^2}{12} \quad (2.6.1.1)$$

$$M_{CB} = \frac{wl_2^2}{24} \left[\frac{3l_1 + 2l_2}{l_1 + l_2} \right]$$

Fig. VI.2.6.2-2 - Moment distribution in lifting lug
(lifting lug clearances ignored)

$$l_1 = 7 \frac{1}{4} + \frac{5}{8} = 7.875$$

$$l_2 = 4 \frac{1}{2} + \frac{5}{8} = 5.125$$

$$M_{AB} = \frac{50000 \times (5.125)^2}{24 \times 13} = 4209 \text{ in.lb}$$

$$M_{BA} = 8418.4 \text{ in.lbs}$$

$$M_{BC} = -8418.4 \text{ in.lbs}$$

Fig. VI.2.6.2-3 - Lifting Lug - Normal - Double Hoist

Fig. VI.2.6.2-4

Lifting Lug-Free Body Forces
-Normal Lift-

$$M_{CB} = \frac{50000 \times 5.125 \times 33.875}{24 \times 13} = 27822. \text{ in.lbs}$$

$$R_A = - \frac{(M_{AB} + M_{BA})}{l_1} = -1603 \text{ lbs } \downarrow$$

$$R_C = \frac{50000 \times 5.125/2 + 27822 - 8418.4}{5.125} = 28786 \text{ lbs } \uparrow$$

$$R_B = 50000 - 28786 + 1603 = 22816 \text{ lbs } \uparrow$$

In effect, the gap between the lifting link and the pin plays an important role in establishing pin behavior. Especially during the emergency lift, when all lifting is done by one leg only, the lifting link subjects the pin to an external couple as shown in Fig. VI.2.6.2-5. In the following sections, a step by step analysis is done to study the performance of the lifting lug pin under normal and emergency loading conditions.

Fig. VI.2.6.2-5
Effect of Gap Between Lifting Link and Pin

VI.2.6.3 Normal Condition (Given Gap Between Supports and Link)

Fig. VI.2.6.3-1 Lifting Lug loadings with Lifting Link Gaps

$$FEM_{AB} = FEM_{BA} = 0$$

$$FEM_{BC} = \frac{w(l_2 - \delta_1 - \delta_2)}{24l_2} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2)(l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 - 6(l_2 - \delta_1 + \delta_2)^2 \right]$$

$$FEM_{CB} = \frac{w(l_2 - \delta_1 - \delta_2)}{24l_2} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2)(l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 2(l_2 - \delta_1 - \delta_2)^2 + 12(l_2 - \delta_1 + \delta_2)(\delta_1 - \delta_2) \right]$$

Required Conditions

$$M_{AB} = \frac{2EI}{l_1} \theta_B$$

$$M_{BA} = \frac{4EI}{l_1} \theta_B$$

$$M_{BC} = \frac{4EI}{l_2} \theta_B + FEM_{BC}$$

$$M_{CB} = \frac{2EI}{l_2} \theta_B + FEM_{CB}$$

Since

$$M_{BA} + M_{BC} = 0$$

$$\frac{4EI}{l_1} \theta_B + \frac{4EI}{l_2} \theta_B + FEM_{BC} = 0$$

$$\theta_B = - \frac{FEM_{BC}}{4EI \left(\frac{1}{l_1} + \frac{1}{l_2} \right)}$$

$$\theta_B = - \frac{FEM_{BC} l_1 l_2}{4EI (l_1 + l_2)}$$

or

$$\theta_B = - \frac{w(l_2 - \delta_1 - \delta_2) l_1}{96 EI (l_1 + l_2)} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2) (l_2 - \delta_1 - \delta_2)^2 + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 - 6(l_2 - \delta_1 + \delta_2)^2 \right]$$

$$M_{AB} = - \frac{w(l_2 - \delta_1 - \delta_2)}{48 (l_1 + l_2)} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2) (l_2 - \delta_1 - \delta_2)^2 + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 - 6(l_2 - \delta_1 + \delta_2)^2 \right]$$

$$M_{BA} = \frac{-w(l_2 - \delta_1 - \delta_2)}{24 (l_1 + l_2)} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2) (l_2 - \delta_1 - \delta_2)^2 + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 - 6(l_2 - \delta_1 + \delta_2)^2 \right]$$

$$M_{BC} = \frac{w(\ell_2 - \delta_1 - \delta_2)}{24(\ell_1 + \ell_2)} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(\ell_2 - \delta_1 + \delta_2)^2 \right]$$

$$M_{CB} = \frac{w(\ell_2 - \delta_1 - \delta_2)}{48\ell_2(\ell_1 + \ell_2)} \left[\left\{ \frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \right. \\ \left. \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 2(\ell_2 - \delta_1 - \delta_2)^2 \right\} (\ell_1 + 2\ell_2) \right. \\ \left. + (\ell_1 + \ell_2) \left\{ \frac{24(\ell_2 - \delta_1 + \delta_2)(\delta_1 - \delta_2)}{\ell_1 + \ell_2} \right. \right. \\ \left. \left. - \frac{2(\ell_2 - \delta_1 - \delta_2)^2 \ell_1}{\ell_1 + \ell_2} + \frac{6(\ell_2 - \delta_1 + \delta_2)^2 \ell_1}{\ell_1 + \ell_2} \right\} \right]$$

Verification

When $\delta_1 = \delta_2 = 0$, reduces to the check problem equations (1) - (4)

RESULTS (Normal Condition - $\frac{1}{4}$ " gap each side)

$$M_{AB} = \frac{-50000}{48 \times 13.0} \left[\frac{3 \times 134.6113}{5.125} - \frac{6 \times 4.5625 \times 16}{5.125} + \frac{3 \times 64}{5.125} \right. \\ \left. + 4 \times 16 - 6 \times 26.2656 \right] \\ = -80.128 \left[78.7968 - 85.4634 + 37.4634 + 64.0 - 157.5936 \right] \\ = 5031 \text{ in.lbs } \downarrow$$

$$M_{BA} = 10063.56 \text{ in.lbs } \downarrow$$

$$M_{BC} = -10063.56 \text{ in.lbs } \downarrow$$

$$M_{CB} = \frac{50000}{48 \times 5.125 \times 13.0} \left[\left\{ \frac{3 \times 134.6113}{5.125} - \frac{6 \times 4.5625 \times 16}{5.125} + \frac{3 \times 64}{5.125} + 2 \times 16 \right\} 18.125 + \left\{ -2 \times 16 + 6 \times 26.2656 \right\} 7.875 \right]$$

$$= 15.6347 \left[\left\{ 78.7968 - 85.4634 + 37.4634 + 32 \right\} 18.125 + (125.5936) \times 7.875 \right] = 33258.78 \text{ in.lb}$$

$$R_A = -\frac{M_{AB} + M_{BA}}{l_1} = -\frac{5031 + 10063.56}{7.875} = -1916.76 \text{ lbs}$$

$$R_C = \frac{50000 \times 5.125/2 + 33258.78 - 10063.56}{5.125} = 29525.74 \text{ lbs}$$

$$R_B = 50000 - 29525.74 + 1916.76 = 22391$$

Fig. VI.2.6.3-2 Breakdown of Forces on Vertical Plates

For Vertical Plate B (Normal Condition)

- (a) $P = 22391$ lbs
Tearing of Cross-section above hole

$$\text{Shear Stress} = \frac{P/2}{\text{Area}} \times 1.5 \text{ (shape factor)}$$

$$= \frac{22391}{2 \times 10 \times 0.625} \times 1.5$$

$$= 2687 \text{ psi}$$

- (b) Shearing of Vertical Welds

$$\text{Shear Stress} = \frac{22391}{2 \times 45 \times 0.375} \times 1.5$$

$$= 995 \text{ psi (for continuous weld)}$$

- (c) Shearing of top weld with inclined plate

$$\text{Shear Stress} = \frac{22391/2 \times 1.5}{14 \times 0.375 \times 0.707}$$

$$= 4524 \text{ psi}$$

- (d) Tension at Cross-section through hole

$$\sigma_t = \frac{22391}{(15 - 4.5) \times 0.625}$$

$$= 3411 \text{ psi}$$

$$\text{From Peterson Fig. 69} \quad K_t = 2.35$$

$$\sigma_{t \max} = 2.35 \times 3411 = 8015 \text{ psi}$$

For Vertical Plate C

$$P = 29525 \text{ lbs}$$

- (a) Tearing of Cross-section above hole

$$\text{Shear Stress} = \frac{29525}{2 \times 7.375 \times 0.625} \times 1.5$$

$$= 4804 \text{ psi}$$

(b) Shearing of Vertical Welds

$$\text{Shear Stress} = \frac{29525 \times 1.5}{2 \times 40 \times 0.375}$$

$$= 1476 \text{ psi (for continuous weld)}$$

(c) Shearing of top weld with inclined plate

$$\text{Shear Stress} = \frac{29525 \times 1.5}{2 \times 14 \times 0.375 \times 0.707}$$

$$= 5965 \text{ psi}$$

(d) Tension at Cross-section through hole

$$\sigma = \frac{29525}{(15 - 4.5) \times 0.625} = 4499$$

From Peterson Fig. 69

$$K_t = 2.35$$

$$\sigma_{\max} = 2.35 \times 4499 = 10572 \text{ psi}$$

Fig. VI.2.6.3-3 Lifting Lug Pin Welding with Vertical Plate C

Analysis of weld between the pin and the right end plate

Moment of Inertia of weld ring

Fig. VI.2.6.3-4 Lifting Pin Welding

$$= \frac{\pi}{64} (5.25^4 - 4.5^4) = 17.1623 \text{ in}^4$$

$$\text{Area of the ring} = \frac{\pi}{4} [(5.25)^2 - (4.5)^2] = 5.74 \text{ in}^2$$

$$\text{Shear Area of the ring} = \pi(5.25 \times 0.375 \cos 45) = 4.3734 \text{ in}^2$$

- (e1) Maximum nominal bending stress at the weld between the support plate B and the pin

$$= \frac{10063.56 \times 2.625}{17.1623}$$

$$= 1539.2 \text{ psi}$$

- (e2) Maximum nominal bending stress at the weld between the support plate C and the pin

$$= \frac{33258.78 \times 2.625}{17.1623}$$

$$= 5087 \text{ psi}$$

- (e3) Shear stress at the weld location between outer support plate C and the pin

$$= \frac{VQ}{bI} = \frac{10}{9} \times \frac{28786}{4.3734} = 7313 \text{ psi}$$

Stress Concentration factor for bending

$$\frac{r}{d} = \frac{2.25}{(15 - 4.5)} \approx 0.22$$

$$\frac{D}{d} = \frac{15}{10.5} \approx 1.5$$

$$(K_t)_{\text{bending}} = 1.46 \text{ (Fig. 60 Peterson)}$$

Hence maximum stress at these welds

$$\sigma_{\text{max}} = 1.46 \times 5087 = 7427 \text{ psi}$$

Refer to attached chart from Peterson's book on stress concentration factors.

RESULTS (Vertical Plate-Normal Condition)

Maximum Stress = 10572 psi

Ultimate Strength of Material = 75000 psi

Factor of Safety = 7.09

VI.2.6.4 Emergency Hoist (Given Gap Between Supports and Links)

- (1) (a) Initial Position (before tilting begins)

Loading = 50T on one hoist

Comparable Moment Arm = 14.9375

Dynamic Impact Factor = 2

$$\text{Factor of Safety} = \frac{25}{50} \times \frac{14.33}{14.9375} \times \frac{18.99}{2} = 4.746$$

- (b) 25° tilt - The above-mentioned case is approximately the same as the 25° tilt case.

Fig. VI.2.6.3-5 Stress Concentration in Vertical plates

Fig. VI.2.6.4-1 Lifting Lug - Emergency Hoist

Loading at 25°-18' tilt (single hoist)

$$W_V = 100,000 \cos(25^\circ-18')$$

$$= 100,000 \times 0.904$$

$$= 90,408 \text{ lbs}$$

$$W_H = 100,000 \times \sin(25^\circ-18')$$

$$= 100,000 \times 0.42735$$

$$= 42,735.75 \text{ lbs}$$

Total loading

$$W = \sqrt{(90408)^2 + (42735.75)^2}$$

$$= 100,000 \text{ lbs}$$

$$= 50 \text{ ton (single hoist load)}$$

Fig. VI.2.6.4-2
Lifting Lug - Free Body Forces
-Emergency Hoist-

(2) Emergency Condition (with Gap - No spacers in gap)

Fig. VI.2.6.4-3 Lifting Lug - With Lifting Link Gaps - Emergency Hoist

$$M_0 = F(l_2 - \delta_1 - \delta_2)$$

$$FEM_{AB} = FEM_{BA} = 0$$

$$FEM_{BC} = \frac{w(l_2 - \delta_1 - \delta_2)}{24l_2} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2) (l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(l_2 - \delta_1 + \delta_2)^2 \right] + \frac{M_0}{l_2^2} \left[4l_2 a - 3a^2 - l_2^2 \right]$$

$$\begin{aligned}
 FEM_{CB} = \frac{w(\ell_2 - \delta_1 - \delta_2)}{24\ell_2} & \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\
 & + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 2(\ell_2 - \delta_1 - \delta_2)^2 \\
 & \left. + 12(\ell_2 - \delta_1 + \delta_2) (\delta_1 - \delta_2) \right] + \frac{M_o}{\ell_2} (2\ell_2 a - 3a^2)
 \end{aligned}$$

Required Conditions:

$$M_{AB} = \frac{2EI}{\ell_1} \theta_B$$

$$M_{BA} = \frac{4EI}{\ell_1} \theta_B$$

$$M_{BC} = \frac{4EI}{\ell_2} \theta_B + FEM_{BC}$$

$$M_{CB} = \frac{2EI}{\ell_2} \theta_B + FEM_{CB}$$

$$M_{BA} + M_{BC} = 0$$

$$\begin{aligned}
 \theta_B = \frac{-w(\ell_2 - \delta_1 - \delta_2) \ell_1}{96EI(\ell_1 + \ell_2)} & \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\
 & + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 \\
 & \left. - 6(\ell_2 - \delta_1 + \delta_2)^2 \right] \\
 & - \frac{M_o \ell_1}{4EI(\ell_1 + \ell_2) \ell_2} \left[4\ell_2 a - 3a^2 - \ell_2^2 \right]
 \end{aligned}$$

$$M_{AB} = - \frac{w(\ell_2 - \delta_1 - \delta_2) \ell_1}{48 (\ell_1 + \ell_2)} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(\ell_2 - \delta_1 + \delta_2)^2 \right]$$

$$- \frac{M_o}{2\ell_2(\ell_1 + \ell_2)} \left[4\ell_2 a - 3a^2 - \ell_2^2 \right]$$

$$M_{BA} = - \frac{w(\ell_2 - \delta_1 - \delta_2) \ell_1}{24 (\ell_1 + \ell_2)} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(\ell_2 - \delta_1 + \delta_2)^2 \right]$$

$$- \frac{M_o}{\ell_2(\ell_1 + \ell_2)} \left[4\ell_2 a - 3a^2 - \ell_2^2 \right]$$

$$M_{BC} = \frac{w(\ell_2 - \delta_1 - \delta_2)}{24 (\ell_1 + \ell_2)} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(\ell_2 - \delta_1 + \delta_2)^2 \right]$$

$$+ \frac{M_o (+\ell_2)}{\ell_2^2 (\ell_1 + \ell_2)} \left[4\ell_2 a - 3a^2 - \ell_2^2 \right]$$

$$\begin{aligned}
M_{CB} = \frac{w(l_2 - \delta_1 - \delta_2)}{48(l_1 + l_2)l_2} \left[\left\{ \frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2) (l_2 - \delta_1 - \delta_2)^2 \right. \right. \\
\left. \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 2(l_2 - \delta_1 - \delta_2)^2 \right\} (l_1 + 2l_2) \right. \\
\left. + \left\{ 24(l_2 - \delta_1 + \delta_2) (\delta_1 - \delta_2) (l_1 + l_2) \right. \right. \\
\left. \left. - 2(l_2 - \delta_1 - \delta_2)^2 l_1 + 6(l_2 - \delta_1 + \delta_2)^2 l_1 \right\} \right] \\
- \frac{M_o}{2l_2^2(l_1 + l_2)} \left[3l_1 a^2 + 6l_2 a^2 - l_1 l_2^2 - 4l_2^2 a \right]
\end{aligned}$$

Verification

(1) When $\delta_1 = \delta_2 = 0$ and $M_o = 0$ reduces to the check problem equations (1) to (4).

(2) When $M_o = 0$ only reduces to normal condition equations (5) to (8).

$$M_o = -42736 \times 18.5 = F \times 4$$

$$F = 197654$$

$$M_o = -790616 \text{ in.lbs} \checkmark$$

If this bending moment which is equivalent to a couple of $F(l_2 - \delta_1 - \delta_2)$ is applied at the center of span l_2 at $a = l_2/2$.

$$\begin{aligned}
M_{AB} = \frac{-90408}{48 \times 13} \left[\frac{3 \times 134.6113}{5.125} - \frac{6 \times 4.5625 \times 16}{5.125} + \frac{3 \times 64}{5.125} + 4 \times 16 - 6 \right. \\
\left. \times 26.2656 \right] + \frac{790616}{2 \times 5.125 \times 13} \left[\frac{4 \times 5.125^2}{2} - \frac{3 \times 5.125^2}{4} - 5.125^2 \right]
\end{aligned}$$

$$= 9532.9 + 38960.44$$

$$= 48493.34 \text{ in.lb} \checkmark$$

$$M_{BA} = 19065.8 + 77921.288$$

$$= 96987 \text{ in.lb} \checkmark$$

$$M_{BC} = -19065.8 - 77921.288$$

$$= -96987 \text{ in.lb } \downarrow$$

$$M_{CB} = \frac{90408}{48 \times 5.125 \times 13} \left[\left\{ \frac{3 \times 134.6113}{5.125} - \frac{6 \times 4.5625 \times 16}{5.125} + \frac{3 \times 64}{5.125} + 2 \times 16 \right\} \right. \\ \left. \times 18.125 + \left\{ -2 \times 16 + 6 \times 26.2656 \right\} \times 7.875 \right] + \frac{790616}{2 \times 5.125^2 \times 13} \\ \times \left[\frac{3 \times 7.875 \times 5.125^2}{4} + \frac{6 \times 5.125^3}{4} - 7.875 \times 5.125^2 - \frac{4 \times 5.125^3}{2} \right]$$

$$= 60137 + \frac{790616}{26} \left[\frac{3 \times 7.875}{4} - \frac{5.125}{2} - 7.875 \right]$$

$$= 60137 - 98827$$

$$= -38690 \text{ in.lbs } \downarrow$$

$$R_A = - \frac{M_{AB} + M_{BA}}{l_1} = -18473 \text{ lbs } \downarrow$$

$$R_C = \frac{90408 \times 5.125/2 - 790616 - 38690 - 96987}{5.125}$$

$$= -135536 \text{ lbs } \downarrow$$

$$R_B = 90408 + 135536 + 18473 = 244417 \text{ lbs } \uparrow$$

For Vertical Plate B (Emergency Condition) without spacer

$$P = 244417 \text{ lbs}$$

(a) Tearing of Cross-section above hole

$$\text{Shear Stress} = \frac{244417}{2 \times 10 \times 0.625} \times 1.5$$

$$= 29330 \text{ psi}$$

(b) Shearing of Vertical Welds

$$\text{Shear Stress} = \frac{244417 \times 1.5}{2 \times 45 \times 0.375} = 10862 \text{ psi (for continuous weld)}$$

(c) Shearing of top weld with inclined plate

$$\text{Shear Stress} = \frac{244417/2 \times 1.5}{14 \times 0.375 \times 0.707} = 49387 \text{ psi}$$

(d) Tension at Cross-section through hole

$$\sigma_t = \frac{244417}{(15 - 4.5) \times 0.625} = 37244.5 \text{ psi}$$

From Peterson Fig. 69 $K_t = 2.35$

$$(\sigma_t)_{\max} = 2.35 \times 37244.5 = 87524 \text{ psi}$$

For Vertical Plate C (Emergency Condition) Without Spacers

$$P = 135536 \text{ lbs}$$

(a) Tearing of Cross-section above hole

$$\text{Shear Stress} = \frac{135536 \times 1.5}{2 \times 7.375 \times 0.625} = 22053 \text{ psi}$$

(b) Shearing of Vertical Welds

$$\text{Shear Stress} = \frac{135536 \times 1.5}{2 \times 40 \times 0.375} = 6776 \text{ psi}$$

(c) Shearing of top weld with inclined plate

$$\text{Shear Stress} = \frac{135536 \times 1.5}{2 \times 14 \times 0.375 \times 0.707} = 27386 \text{ psi}$$

(d) Tension at Cross-section through hole

$$\sigma_t = \frac{135536}{(15 - 4.5) \times 0.625} = 20653 \text{ psi}$$

$$\text{From Peterson Fig. 69} \quad K_t = 2.35$$

$$(\sigma_t)_{\max} = 2.35 \times 20653 = 48534 \text{ psi}$$

(e1) Maximum Nominal Bending Stress at the weld between the support plate B and the pin

$$= \frac{96987 \times 2.625}{17.1623} = 14834 \text{ psi}$$

(e2) Maximum nominal bending stress at the weld between the support plate C and the pin

$$= \frac{38690 \times 2.625}{17.1623} = 5917 \text{ psi}$$

(e3) Shear Stress at the weld location between outer support plate B and the pin

$$\tau = \frac{10}{9} \times \frac{244417}{4.3734} = 62096 \text{ psi}$$

RESULTS (Emergency Condition without spacers)

- (1) Support plate B and the pin will be highly stressed.
- (2) As the support plate begins to yield, there would be a tendency for some of the reaction load at support B to be shifted to the support plate C. This is expected to relieve some stresses at the support plate B.

(3) Emergency Condition (with spacer)

As the support plate A begins to take the additional load, it begins to apply additional moment on the pin in that region

M_0 = external moment due to the loading on the outer support plate C
(vertical) - applied moment

$$FEM_{AB} = FEM_{BA} = 0$$

$$FEM_{BC} = \frac{w(\ell_2 - \delta_1 - \delta_2)}{24\ell_2} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(\ell_2 - \delta_1 + \delta_2)^2 \right] + \frac{M_0}{\ell_2^2} \left[4\ell_2 a - 3a^2 - \ell_2^2 \right]$$

$$FEM_{CB} = \frac{w(\ell_2 - \delta_1 - \delta_2)}{24\ell_2} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 2(\ell_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + 12(\ell_2 - \delta_1 + \delta_2) (\delta_1 - \delta_2) \right] + \frac{M_0}{\ell_2^2} (2\ell_2 a - 3a^2)$$

Required Conditions:

$$M_{AB} = \frac{2EI}{\ell_1} \theta_B$$

$$M_{BA} = \frac{4EI}{\ell_1} \theta_B$$

$$M_{BC} = \frac{4EI}{l_2} \theta_B + FEM_{BC}$$

$$M_{CB} = \frac{2EI}{l_2} \theta_B + FEM_{CB}$$

$$M_{BA} + M_{BC} = 0$$

$$\theta_B = \frac{-w(l_2 - \delta_1 - \delta_2)l_1}{96EI(l_1 + l_2)} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2)(l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 - 6(l_2 - \delta_1 + \delta_2)^2 \right. \\ \left. - \frac{M_o l_1}{4EI(l_1 + l_2)l_2} \left[4l_2^2 a - 3a^2 - l_2^2 \right] \right]$$

$$M_{AB} = \frac{-w(l_2 - \delta_1 - \delta_2)l_1}{48(l_1 + l_2)} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2)(l_2 - \delta_1)^2 \delta_2 \right. \\ \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(l_2 - \delta_1 + \delta_2)^2 \right] \\ - \frac{M_o}{2l_2(l_1 + l_2)} \left[4l_2^2 a - 3a^2 - l_2^2 \right]$$

$$M_{BA} = - \frac{w(l_2 - \delta_1 - \delta_2)l_1}{24(l_1 + l_2)} \left[\frac{3}{l_2} (l_2 - \delta_1 + \delta_2)^3 - \frac{6}{l_2} (l_2 - \delta_2)(l_2 - \delta_1)^2 \delta_2 \right. \\ \left. + \frac{3}{l_2} (l_2 - \delta_1 - \delta_2)^3 + 4(l_2 - \delta_1 - \delta_2)^2 \right. \\ \left. - 6(l_2 - \delta_1 + \delta_2)^2 \right] \\ - \frac{M_o}{l_2(l_1 + l_2)} \left[4l_2^2 a - 3a^2 - l_2^2 \right]$$

$$\begin{aligned}
M_{BC} = & \frac{w(\ell_2 - \delta_1 - \delta_2)}{24(\ell_1 + \ell_2)} \left[\frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \\
& + \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 4(\ell_2 - \delta_1 - \delta_2)^2 - \\
& \left. 6(\ell_2 - \delta_1 + \delta_2)^2 \right] \\
& + \frac{M_0 (+\ell_2)}{\ell_2^2 (\ell_1 + \ell_2)} \left[4\ell_2^2 a^2 - 3a^2 - \ell_2^2 \right] \\
M_{CB} = & \frac{w(\ell_2 - \delta_1 - \delta_2)}{48(\ell_1 + \ell_2)\ell_2} \left[\left\{ \frac{3}{\ell_2} (\ell_2 - \delta_1 + \delta_2)^3 - \frac{6}{\ell_2} (\ell_2 - \delta_2) (\ell_2 - \delta_1 - \delta_2)^2 \right. \right. \\
& + \left. \left. \frac{3}{\ell_2} (\ell_2 - \delta_1 - \delta_2)^3 + 2(\ell_2 - \delta_1 - \delta_2)^2 \right\} (\ell_1 + 2\ell_2) \right. \\
& + \left. \left\{ 24(\ell_2 - \delta_1 + \delta_2) (\delta_1 - \delta_2) (\ell_1 + \ell_2) \right. \right. \\
& - \left. \left. 2(\ell_2 - \delta_1 - \delta_2)^2 \ell_1 + 6(\ell_2 - \delta_1 + \delta_2)^2 \ell_1 \right\} \right] \\
& - \frac{M_0}{2\ell_2^2 (\ell_1 + \ell_2)} [3\ell_1^2 a^2 + 6\ell_2^2 a^2 - \ell_1 \ell_2^2 - 4\ell_2^2 a]
\end{aligned}$$

Verification

- (1) When $\delta_1 = \delta_2 = 0$ and M_0 reduces to the check problem equations (1) to (4).
- (2) When $M_0 = 0$ only reduces to normal condition equations (5) to (8).

Loading on outer support plate C due to spacer

From Timoshenko's "Theory of plates and shells, case (1) and case (2) are studied to estimate the maximum bending moment that the pin can tolerate.

In case (1) and case (2), maximum bending moment occurs at the clamped edge. However, for the pin the maximum bending moment will be considerably less than that at the clamped edges. For the lifting lug configuration, the clamped edges occur at one side plate and vertical plate junction, and the other at the counter balance mounting.

Case (1) - Fig. VI.2.6.4-5

Maximum Bending Moment = $0.343 P$ where P is the load applied by the space

Case (2) - Fig. VI.2.6.4-6

Maximum Bending Moment = $0.5 P$

Timoshenko's Theory of Plates and Shells

In conclusion, let us consider a ~~line~~ load P (Fig. VI.2.6.3-10) uniformly distributed over a straight-line segment of some length u . The moment caused by such a load at the mid-point of the built-in edge is readily found as follow

Fig. VI.2.6.4-5 Line Load
Plate C by Lifting Link
Contact

Substituting $x = a/2$ and $P d\xi/u$ for P in this expression and integrating we obtain for the required moment

$$\begin{aligned}
 (M_y)_{x=a/2, y=0} &= -\frac{2P\eta}{au} \sinh \frac{\pi\eta}{a} \int_{(a-u)/2}^{(a+u)/2} \frac{\sin \frac{\pi\xi}{a} d\xi}{\cosh \frac{2\pi\eta}{a} + \cos \frac{2\pi\xi}{a}} \\
 &= -\frac{2P\eta}{\pi u} \arctan \frac{\sin \frac{\pi u}{2a}}{\sinh \frac{\pi\eta}{a}}
 \end{aligned}$$

Table VI.2.6.4 gives the position of the load producing the numerically largest clamping moment and the value of that moment for various values of the ratio, u/a

From Timoshenko's "Plate and Shell"
Case (2) for this report.

Fig. VI.2.6.4-6 Line Load on Plate B - Modified Assumption

Table VI.2.6.4-1 Largest Clamping Moments at $x = a/2$ Due to a Single Load Distributed Over A Length u (Fig. VI.2.6.4-5)

u/a	0	0.1	0.2	0.4	0.6	0.8	1.0
η/a	0	0.147	0.203	0.272	0.312	0.321	0.343
M_y/P	-0.318	-0.296	-0.275	-0.237	-0.204	-0.172	-0.143

Case (1) for this report

It is conservative, for our purpose of designing adequate spacers and reinforcements, to use the maximum of these two cases.

Hence to incorporate into pin analysis

$$M_o = 0.5 P$$

From link analysis in attached pages

$$P = 148000 \text{ lbs}$$

$$M_o = 0.5 \times 148000$$

$$= 74000 \text{ in.lbs}$$

This bending moment will now be used as an applied external moment in the pin analysis.

$$M_{AB} = 9532.9 - \frac{74000}{2 \times 5.125 \times 13} \left[\frac{4 \times 5.125^2}{2} - \frac{3 \times 5.125^2}{4} - 5.125^2 \right]$$

$$= 9532.9 - 3646.6$$

$$= 5886.26 \text{ in.lbs } \downarrow$$

$$M_{BA} = 11772.5 \text{ in.lbs } \downarrow$$

$$M_{BC} = -11772.5 \text{ in.lbs } \downarrow$$

$$\begin{aligned}
 M_{CB} &= 60137 - \frac{74000}{2 \times 5.125^2 \times 13} \left[\frac{3 \times 7.875 \times 5.125^2}{4} + \frac{6 \times 5.125^3}{4} - 7.875 \times 5.125^2 \right. \\
 &\quad \left. - \frac{4 \times 5.125^3}{2} \right] \\
 &= 60137 - \frac{74000}{26} \left[\frac{3 \times 7.875}{4} - \frac{5.125}{2} - 7.875 \right] \\
 &= 60137 - (-12896.6) \\
 &= 60137 + 12896.6 \\
 &= 73033.6 \text{ in.lbs}
 \end{aligned}$$

$$R_A = - \frac{(M_{AB} + M_{BA})}{l_1}$$

$$= -2242.38 \text{ lbs } \downarrow$$

$$R_C = \frac{90408 \times 5.125/2 + 73033.6 - 11772.5}{5.125}$$

$$= 67157 \text{ lbs } \uparrow$$

$$R_B = 90408 - 67157 + 2242.38$$

$$= 35492.9 \text{ lbs } \uparrow$$

VI.2.6.5 Stresses for Emergency Hoist

For Vertical Plate B

(a) $P = 35493 \text{ lbs}$

Tearing of cross-section above hole

$$\begin{aligned} \text{Shear stress} &= \frac{P/2}{\text{Area}} \times 1.5 \text{ (shape factor)} \\ &= \frac{35493}{2 \times 10 \times 0.625} \times 1.5 \\ &= 4259 \text{ psi} \end{aligned}$$

(b) Shearing of Vertical welds

$$\begin{aligned} \text{Shear stress} &= \frac{35493}{2 \times 45 \times 0.375} \times 1.5 \\ &= 1577.4 \text{ psi} \end{aligned}$$

(c) Shearing of top weld with inclined plate

$$\begin{aligned} \text{Shear stress} &= \frac{35493/2 \times 1.5}{14 \times 0.375 \times 0.707} \\ &= \frac{14343}{2} \text{ psi} = 7171 \text{ psi} \end{aligned}$$

(d) Tension at cross-section through hole

$$\sigma_t = \frac{35493}{(15 - 4.5) \times 0.625} = 5408 \text{ psi}$$

$$(\sigma_t)_{\max} = 2.35 \times 5408 = 12709 \text{ psi}$$

For Vertical Plate C

$$P = 67157 \text{ lbs}$$

- (a) Tearing of Cross-section above hole

$$\text{Shear stress} = \frac{67157}{2 \times 7.375 \times 0.625} \times 1.5 = 10927 \text{ psi}$$

- (b) Shearing of Vertical welds

$$\text{Shear stress} = \frac{67157 \times 1.5}{2 \times 40 \times 0.375} = 3357.8 \text{ psi}$$

- (c) Shearing of top weld with the inclined plate

$$\text{Shear stress} = \frac{67157 \times 1.5}{2 \times 14 \times 0.375 \times 0.707}$$

- (d) Tension at Cross-section through hole

$$\sigma_t = \frac{67157}{(15 - 4.5) \times 0.625} = 10233$$

$$\begin{aligned} (\sigma_t)_{\max} &= 2.35 \times 10233 \\ &= 24048 \end{aligned}$$

$$\text{F.S.} = \frac{75000}{24048} = 3.11$$

VI.2.6.6 Improved Assumptions and Independent Verification of Lifting Lug Analysis

The object of this study is to evaluate the flexural stiffness of the support plates on the SLSF Loop Handling system and include such a factor into the system analysis.

A simplified analysis was used to determine the approximate flexural stiffnesses of support plates A and C. These stiffnesses were included in the analysis of the supporting pin rather than the assumed fixed conditions. The system was reanalysed for both the normal and the emergency conditions. The stresses in the system were rechecked and compared to allowable stresses in the members and in the welded connections.

The computations used in the following pages of this section have improved assumptions from the analysis of Sections VI.2.6.3 and VI.2.6.4 in that, here, the vertical plate stiffnesses are included in an approximate way.

The inclusion of vertical plate stiffnesses in the analysis was now considered since the analysis in Sections VI.2.6.3 and VI.2.6.4 indicated that the lifting lug design is not capable of handling an emergency hoist condition. The analysis in this section, with these improved assumptions, further validates conclusions of previous sections.

Estimation of Plate
Stiffnesses for Support
Plates A and C:

d = diam of the pin

Due to anti-symmetry, the plate can be approximated by the following:

In order to find an estimated value of rotation at the center of the pin, assume a plate simply supported on all edges with the shown dimensions relative to the original dimensions. First order approx. of the solution is:

$$w(x,y) = \bar{w} \sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$$

$$P(x,y) = \frac{1.0}{.785d} \left(\frac{4}{ab}\right) \sin \frac{\pi \xi}{a} \sin \frac{\pi \eta}{b}$$

$$\sin \frac{\pi x}{a} \sin \frac{\pi y}{b}$$

$$\bar{w} = \frac{4.0}{.785dab} \frac{\sin \pi \xi/a \sin \pi \eta/b}{[(\pi/a)^2 + (\pi/b)^2]^2 D}$$

$$\approx \frac{5.1}{\pi^4 d D} \frac{a^3 b^3 \sin(.393d\pi/b)}{(b^2 + a^2)^2}$$

Fig. VI.2.6.6-1 Lifting Lug - Vertical plates A and C

In order to find the plate stiffness, it is necessary to find the max rotation about x axis at $x = a/2$ and $y = 0$.

$$\theta @ x\text{-axis } (x,y) = \bar{w} \frac{\pi}{b} \sin \frac{\pi x}{a} \cos \frac{\pi y}{b} \text{ at } x = a/2 \text{ and } y = 0 \quad \theta_{\max} = \frac{\pi}{b} \bar{w}$$

$$\theta_{\max} = \frac{5.1}{\pi^3} \frac{a^3 b^2}{dD(b^2 + a^2)^2} \sin \left(\frac{.393d\pi}{b} \right)$$

$$D = \frac{Et^3}{12(1 - \nu^2)} \quad \text{use } E = 30 \times 10^3 \text{ ksi, } \nu = 0.3$$

given following data:

$$d = 4\frac{1}{2} \text{ "}, \quad t = \frac{5}{8} \text{ "}$$

$$l_2 = 14\frac{3}{4} \text{ "}$$

$$\frac{l_1}{2} = 7\frac{3}{8} \text{ "}$$

$$a = 0.8l_2 = 11.8 \text{ "}, \quad b = 14.75 \text{ "}$$

$$D = \frac{30 \times 10^3 \times (0.625)^3}{12(1 - .09)} = 0.671 \times 10^3$$

$$\theta_{\max} = \frac{5.1}{\pi^3} \frac{(11.8)^3 (14.75)^2 \sin \left[\frac{.393\pi(4.5)}{14.75} \right]}{(4.5)(.671)(10)^3 [(11.8)^2 + (14.75)^2]^2} = 0.563 \times 10^{-4}$$

$$K = 17.74 \times 10^3 \text{ k.in}$$

$$\text{Pin stiffness} = \frac{4EI}{L} = \frac{4 \times 30 \times 10^3 \times \pi(4.5)^4}{64(5.125)} = 471 \times 10^3 \text{ k.in}$$

at support C

$$\text{or at support A} = 471 \times 10^3 \times \frac{5.125}{7.875} = 306 \times 10^3 \text{ k.in}$$

The above approximate analysis shows that the support plates are not stiff enough to be considered a fixed support. Spring supports at the ends A and C are considered as shown below:

Fig. VI.2.6.6-2 Lifting Lug Loading - with plate stiffness considered

Use Moment Distribution to find moments

Fixed End Moments = 21354 lb.in					
$wL_2 = 50000 \text{ lb}$					
$.232 EI/L_1 \quad 4EI/L_1 \quad \frac{4EI}{L_1}$					
$.151 EI/L_2$					
$l_1 = 7.875 \quad \frac{4EI}{L_2} \quad l_2 = 5.125$					
Dist. Factors	.055	.945	.394	.606	.964 .036
F.E.M.	0	0	21354	-21354	
	0	-8414	-12940	20585	769
	-4206	0	10293	-6470	
231	+3975	-4055	-6238	6237	233
	-2028	+1987	3118	-3118	
112	1916	-2011	-3094	3006	112
	-1006	958	1503	-1547	
55	951	-970	-1491	1491	56
	-485	475	746	-746	
27	458	-481	-740	719	27
	-241	229	360	-370	
13	228	-232	-357	357	13
438	-438	-12514	12514	-1210	1210

Fig. VI.2.6-3 Lifting Lug Moment Distribution - Normal Condition

Fig. VI.2.6.6.-4 - Lifting Lug Loading - Plate Stiffness Accounted
max moment on pin: - Normal Condition

$$x = \frac{22794.3}{50,000/5.125} = 2.336 \text{ in}$$

$$M_{\max} = -1210 + (22794.3)(2.336) - \frac{50,000}{5.125} \frac{(2.336)^2}{2} = 25418 \text{ lb.in}$$

$$\text{max stress in pin} = 25418 \left(\frac{32}{\pi(4.5)^3} \right) = 2841 \text{ psi}$$

Emergency Condition (without spacer):

$$0.232 EI/L_1$$

Fig. VI.2.6.6-5 Lifting Lug Loading - Plate Stiffness Accounted-
Emergency Condition
 $w = 90408/5.125 = 17640 \text{ lb/in}$

$$M_o = (8 + 18.5) 42735.75 = 1132497 \text{ lb.in}$$

Fixed End Moments

$$FEM_{BC} = + \frac{(17640)(5.125)^2}{12} + \frac{1132497}{4} = +321735 \text{ lb.in}$$

$$FEM_{CB} = -38610 + 283124 = +244514 \text{ lb.in}$$

More realistic to assume the moment M_o distributed over the pin length

Fig. VI.2.6.6-6 Moment M_o Distribution

$$F = \frac{1132497(6)}{(5.125)^2} = 258702 \text{ lb/in}$$

$$FEM_{BC} = +38610 + \frac{258702 (5.125)^2}{15(4)} = 151860 \text{ lb.in}$$

$$FEM_{CB} = -38610 + 113250 = 74640 \text{ lb.in}$$

Vertical Plate B: (emergency condition without spacer)

$$P = 281350 \text{ lbs}$$

$$(a) \text{ Shearing Stress} = \frac{281350}{2 \times 10 \times 0.625} \times 1.5 = 13188 \text{ psi}$$

(tearing of section above)

$$(b) \text{ Shearing stress on welds (vertical)}$$

$$\text{Shear Stress} = \frac{281350}{2 \times 45 \times 0.375} \times 1.5 = 8336 \text{ psi}$$

$$(c) \text{ Shearing of Top Weld with Inclined Plate}$$

$$\text{Shear Stress} = \frac{281350}{2 \times 14 \times 0.375 \times 0.707} \times 1.5 = 56850 \text{ psi}$$

$$(d) \text{ Tension at Cross-section through Hole}$$

$$\sigma_t = \frac{281350}{(15 - 4.5)0.625} = 42872 \text{ psi}$$

$$(\sigma_t)_{\max} = 2.35 \times 42872 = 100750 \text{ psi}$$

$$= 100 \text{ ksi}$$

Vertical Plate C:

$$P = 184919 \text{ lbs}$$

$$(a) \text{ Tearing of Cross-section above Hole}$$

Force is such that this will not occur

$$(b) \text{ Shearing Stress} = \frac{184919}{2 \times 40 \times 0.375} \times 1.5 = 9246 \text{ psi}$$

$$(c) \text{ Shear Stress} = \frac{184919}{2 \times 14 \times 0.375 \times 0.707} \times 1.5 = 37365 \text{ psi}$$

$$(d) \sigma_t = \frac{184919}{(15 - 4.5)0.625} = 28178 \text{ psi}$$

$$\sigma_{t(\max)} = 2.35 \times 28178 = 66219 \text{ psi} = 66.22 \text{ ksi}$$

$$(e_2) \text{ Maximum Nominal Bending Stress at the weld between the support plate and the pin}$$

$$= \frac{1047 \times 2.625}{17.1623} = 160 \text{ psi}$$

$$(e_3) \text{ Shear Stress at the weld location between plate B and the pin}$$

$$\tau \approx \frac{10}{9} \frac{281350}{4.3734} = 71480 \text{ psi} = 71.5 \text{ ksi}$$

Moment Distribution for Emergency Condition
(without Spacers):

.055	.945	.394	.606	.964	.036
	0	0	151860	74640	0
		-59833	-92027	-71953	-2687
	-29916		-35976	-46014	
+1645	+28271	+14175	+21801	44357	1657
	7087	+14135	+22179	+10901	
-390	-6697	-14308	-22006	-10509	-392
	-7154	-3349	-5254	-11003	
+393	+6761	+3390	+5213	+10607	+396
	+1695	3380	+5303	+2607	
-93	-1602	-3421	-5262	-2513	-94
	-1711	-801	-1257	-2631	
+94	+1617	+811	+1247	+2536	+95
	+405	+808	+1268	+624	
-22	-383	-818	-1258	-602	-22
1627	-1627	-45831	45831	+1047	-1047

Fig. VI.2.6.6-7 Lifting Lug Moment Distribution - Emergency Condition

Check max bending stress in pin:

$$S = \frac{\pi d^3}{32} = \frac{\pi (4)^3}{32} = 6.28 \text{ in}^3$$

$$M_{\max} = 45831 \text{ lb.in}$$

$$f_{\max} = \frac{45831}{6.28} = 7294 \text{ psi}$$

From the previous computations, it is shown that there are several weak connections in the lifting lug under the emergency condition. The following is only a scheme to illustrate how the system weaknesses can be efficiently eliminated. However the scheme shown in Drawing R1030-0103-DE-03 is the recommended reinforcing system.

$\frac{3}{8}$ " spacer pl. to be welded
to ensure contact between
link and plate C full penetration weld

Fig. VI.2.6.6-8 Lifting Lug Design Modification Concept for Emergency Hoist.

The basic concept here is to take the contact force at the tip of support plate C directly to the side plate, and avoid its effect going back into the support pin and the other support plates. The force at this contact point between the lifting link and plate C will be assumed equal to

$$W_H = 42735.75 \text{ lbs} \approx 42740 \text{ lbs}$$

The Reinforcing Beam

Total Force = 42740 lbs

$$\text{width of Lifting Link at contact pt.} = 10.75 - \frac{3.25}{18.5} (7.375) = 9.5"$$

$$42740/9.5$$

Fig. VI.2.6.6-9 Lifting Lug - Vertical Plate C - Reinforced Beam

$$M_{\max} = \frac{42740}{2} \left[(7.5) - \frac{9.5}{4} \right] = 109521 \text{ lb.in}$$

assuming that this moment will be totally taken by the reinforcing beam

$$S_{\text{req'd}} = \frac{109521}{24} = 4.56 \text{ in}^3$$

use M 4 x 13 or its equivalent $S_x = 5.24 \text{ in}^3$

$$\text{Reaction} = \frac{42740}{2} = 21370 \text{ lbs} = 21.37^k$$

use 1/4" weld E70xx

$$\text{cap.} = 0.925 \times 4 = 3.7 \text{ k/in length}$$

$$\text{required weld length} = \frac{21.37}{3.7} = 5.7 \text{ in}$$

weld along both sides of web and flange

Design of support plate

$$\text{Tension Force} = 21.37^k$$

$$\text{Area of plate} = 4 \times 0.625 = 2.5 \text{ in}^2$$

$$\text{tensile stress} = \frac{21.37}{2.5} = 8.55 \text{ ksi}$$

Analysis of the Handling System with Spacers

$$W_H = 42735.75 \text{ lbs}$$

$$W_V = 90408 \text{ lbs}$$

$$M = 42735.75 (18.5 + 8 - 5.375) = 902793 \text{ lb.in}$$

Fig. VI.2.6.6-10 Lifting Lug Loading with Reinforced Condition

Moment distribution for the emergency condition with spacers:

.055	.945	.394	.606	.964	.036
			128889	51669	
		-50782	-78107	-49809	-1860
	-25391	0	-24904	-39053	
+1396	+23995	+9812	+15092	+37647	+1406
	+4906	+11997	+18824	-7546	
-270	-4636	-12143	-18678	-7274	-272
	-6072	-2318	-3637	-9339	
+334	+5738	+2346	+3609	+9003	+336
	+1173	+2869	+4501	+1804	
-65	-1108	-2904	-4466	-1739	-65
	-1452	-554	-870	-2233	
+80	+1372	+561	+863	+2153	+80
	+281	-686	-1076	+431	
-15	-266	+694	+1068	-415	-16
+1460	-1460	-41108	41108	+391	-391

VI.2.6.6-11

Lifting Lug - Moment Distribution with Spacers in Gaps

From the previous analysis without a spacer, only the following points need to be rechecked again. The other stresses can be concluded to be satisfactory by comparison.

- (1) Tension stresses at cross-section through hole for support plate B

$$\sigma_t = \frac{234861}{(15 - 4.5)0.625} = 35788 \text{ psi}$$

with stress concentration factor of 2.35

$$\sigma_{\max} = 2.35(35788) = 84103 \text{ psi}$$

- (2) Shear stress at weld location between plate B and the pin

$$\tau = \frac{10}{9} \frac{234861}{4.3734} = 59669 \text{ psi}$$

- (3) Tension stresses at cross-section through hole for support C

$$\sigma_t = \frac{176155}{(15 - 4.5)(.625)} = 268843 \text{ psi}$$

$$\sigma_{\max} = 2.35(268843) = 63080 \text{ psi}$$

- (4) Shear stresses at weld location between plate C and the pin

$$\tau = \frac{10}{9} \frac{176155}{4.3734} = 44754$$

Support Plates B and C need to be reinforced at the pin-plate connection. The following solution is suggested:

Fig. VI.2.6.6-12 Lifting Lug - Reinforcing Consideration of Vertical Plate C

Consider that the force is transferred from the pin to the plates by bearing

$$\text{bearing area plate B} = (0.625)(4) = 2.5 \text{ in}^2$$

$$\text{bearing area additional plate} = (1.0)(4) = 4.0 \text{ in}^2$$

$$\text{total bearing area} = 6.5$$

$$f_p = \frac{234861}{6.5} = 36132 \text{ psi (allowable for 36 steel is 48.6 ksi)}$$

$$\text{Force in plate B} = \frac{234861}{6.5} (2.5) = 90331 \text{ lbs}$$

$$\text{Force in additional plate} = 144530 \text{ lbs}$$

Tensile stress in the two plates

$$= \frac{234861}{(15 - 4.5)(1.625)} = 13765 \text{ psi}$$

stress factor = 2.35

$$\sigma_{\max} = 2.35 (13.765) = 32.25 \text{ ksi}$$

Req'd length of welding for the additional plate can be obtained as follows:

$$\text{cap. of } 1/2" \text{ E70xx weld} = 8 \times 0.925 = 7.42 \text{ k/in}$$

Req'd length of each side =

$$\frac{144.53}{2(7.42)} = 9.7 \text{ in}$$

use 10" each side

Check welding of handling system with the main structure

Fig. VI.2.6.6-13 Lifting Lug - Loading Layout for Welding Analysis

$$\Sigma V = 234.86 - 139.05 - 5.4 = 90.41^k$$

$$\Sigma H = 42.7^k$$

$$\begin{aligned}\Sigma M @ C.G. &= -(5.4)(8.5625) + 234.86(16.4375) - 139.05(21.5625) - 42.7(5.375) \\ &\quad + 36.5 - 27.5) \\ &= 202.2 \text{ k.in.}\end{aligned}$$

$$\text{area of weld} = 2(55)(.375) = 41.25 \text{ in}^2$$

$$\text{section modulus} = \frac{2(55)^2}{6} (.375) = 378.125 \text{ in}^3$$

$$f_{1 \text{ max}} = \frac{42.7}{41.25} + \frac{202.2}{378.125} = 1.04 + 0.53 = 1.57 \text{ ksi}$$

$$f_2 = \frac{90.41}{41.25} = 2.2 \text{ ksi}$$

$$f_{\text{max}} = \sqrt{(1.57)^2 + (2.2)^2} = 2.7 \text{ ksi}$$

A conservative assumption is to assume the weld only effective in the top 36" only

$$\begin{aligned}\Sigma M @ C.G. &= -5.4(8.5625) + 234.86(16.4375) - 139.05(21.5625) - 42.7(5.375) \\ &= 607.8 \text{ K}\end{aligned}$$

$$\text{Area of weld} = 2(36)(.375) = 27 \text{ in}^2$$

$$\text{section modulus} = \frac{2(36)^2}{6} (.375) = 162 \text{ in}^3$$

$$f_1 = \frac{42.7}{27} + \frac{607.8}{162} = 5.33$$

$$f_2 = \frac{90.41}{27} = 3.34$$

$$f_{\text{max}} = \sqrt{(3.34)^2 + (5.33)^2} = 6.3 \text{ ksi}$$

Conclusions

The incorporation of the support plate flexural stiffness has had appreciable effect on the analysis. It can be seen that the reaction at the support plate B has increased when the flexibility of the end support plates was included as compared with the previous assumption of fixed supports. As a result of this analysis, it was found that the support plates A and B require reinforcing in order to safely accommodate the emergency condition. It is very obvious that the support B needs strengthening against shear splitting, which can be attained by welding a doubler plate which must go around the pin support. In the case of the end support plate, two weaknesses were detected; namely, (1) shear splitting as in the case of support B which can be solved in a similar manner; and, (2) flexural weakness resulting from supporting the component W_H in the emergency condition. The latter weakness can be resolved by reinforcing the support plate through a system of supports and stiffness that would support the force W_H and directly transfer it to the two supporting side plates. The above provisions are shown in drawing No. R 1030-0103-DE-03 and sketched in the following discussion. Also a preliminary analysis and design of the proposed reinforcement is included.

$$F = \frac{902793}{(5.125)^2} (6) = 206230 \text{ lbs/in}$$

$$FEM_{BC} = 38610 + \frac{206230 (5.125)^2}{15(4)} = 128889 \text{ lb.in}$$

$$FEM_{CB} = -38610 + 90279 = 51669 \text{ lb.in}$$

Fig. VI. 2.6.6-14 Back Up Plate Loading - Development of Reinforcement of Lifting Lug

VI.2.6.7 Recommended Design for Emergency Hoist

Fig. VI.2.6.7-1 Stress Distribution in Back-Up Plate - Development of Reinforcement of Lifting Lug

Fig. VI.2.6.7-2 - Lifting Link Modified Loading in the Presence of Spacers

LOADING ON STRUCTURE DUE TO CONTACT WITH LIFTING LINK

The loading from the lifting link will be somewhat modified due to current spacer configuration. It is not considered significant, however, to redo the lifting link analysis since the previous analysis was conservative

Taking moment at P_1 location

$$WH(13.59 + 1.375) = P_2 \times (4.91 + 2.166)$$

$$P_2 = \frac{42736.75 \times 14.965}{7.076}$$

$$= 90383.75 \text{ lbs}$$

$$P_1 = P_2 + W_H = 133120 \text{ lbs}$$

LOADING

From lifting link calculations maximum loading on the outer support structure that is possible due to the tilting of the machine under the emergency conditions is:

$$P = 133120$$

Maximum bending moment on the "L" frame radius

$$M = 133120 \times 7.375$$

$$= 981763 \text{ in.lb}$$

Fig. VI.2.6.7-3 - Cross-section of Reinforcement - Section BB

Neutral Axis

$$\bar{y} = \frac{8 \times 0.625 \times 0.3125 + 2 \times 6 \times 1 \times 3.625 + 2 \times 3 \times 1 \times 2.125 + 4 \times 1.125 + 4 \times 6.125}{8 \times 0.625 + 2 \times 6 + 2 \times 3 + 4 \times 1 \times 2}$$

$$= 2.8 \text{ in}$$

Moment of Inertia

$$\begin{aligned} I_{yy} &= \frac{8}{12} (0.625)^3 + 8 \times 0.625 \times (2.4875)^2 + \frac{2 \times 1 \times 3^3}{12} + 2 \times 3 \times (.675)^2 + \frac{2 \times 1 \times 6^3}{12} \\ &\quad + 2 \times 6 \times 1 \times (0.825)^2 + 4 \times \frac{1^3}{12} + 4 \times 1 \times (1.675)^2 + \frac{4 \times 1^3}{12} + 4 \times 1 \times (3.325)^2 \\ &= 138.613 \text{ in}^4 \end{aligned}$$

Total Area

$$\begin{aligned} A &= 8 \times 0.625 \times 2 \times 3 + 2 \times 6 + 4 \times 1 + 4 \times 1 \\ &= 31.0 \text{ in}^2 \end{aligned}$$

First Moment (above neutral axis)

$$\begin{aligned}
 Q_{yy} &= 8 \times 0.625 \times 2.4875 + 4 \times (2.80 - 0.625) \times 1.4 + 4 \times 1 \times (2.80 + 2.5) \\
 &= 31.3175 \text{ in}^3
 \end{aligned}$$

Tension Stress

$$\begin{aligned}
 \sigma_t &= \frac{P}{A} \\
 &= \frac{133120}{31} \\
 &= 4294 \text{ psi}
 \end{aligned}$$

$$\frac{r}{d} = \frac{2}{12} = 0.166$$

$$\frac{D}{d} = \frac{18}{12} = 1.5$$

Figure 57 Peterson $K_t = 1.8$

$$\begin{aligned}
 (\sigma_t)_{\max} &= 1.8 \times 4.294 \\
 &= 7729 \text{ psi}
 \end{aligned}$$

There will not be any shear stress at cross-section BB. The combined maximum normal stress will be composed of tension and bending components.

Bending Stress (at end of section near radius)

$$\begin{aligned}
 \sigma_b &= \frac{Mc}{I} = \frac{981763 \times 2.8}{138.613} \\
 &= 19,831 \text{ psi}
 \end{aligned}$$

$$\frac{r}{d} = \frac{2}{12} = 0.1667$$

$$\frac{L}{D} = \frac{16}{30} = 0.5333$$

Figure 62 Peterson $K_t = 1.6$

$$\begin{aligned} (\sigma_b)_{\max} &= 1.6 \times 19831 \\ &= 31730 \text{ psi} \end{aligned}$$

Bending Stress (at end of section away from radius)

$$\begin{aligned} \sigma_b &= \frac{981763 \times 3.2}{138.613} \\ &= 22664 \text{ psi} \end{aligned}$$

$$K_t = 1.0$$

$$(\sigma_b)_{\max} = 22664 \text{ psi}$$

Maximum combined normal stress at the end of the cross-section near the radius

$$\begin{aligned} \sigma_{\text{combined}} &= 31730 + 7729 \\ &= 39459 \text{ psi} \end{aligned}$$

Maximum combined normal stress at the end of the cross-section away from the radius

$$\begin{aligned} \sigma_{\text{combined}} &= 22664 - 7729 \\ &= 15547 \text{ psi} \end{aligned}$$

Fig. VI.2.6.7-4 Cross-section of Reinforcement - Section AA

Neutral Axis

$$\bar{y} = \frac{8 \times 0.625 \times 0.3125 + 2 \times 10 \times 5.625 + 2 \times 5 \times 3.125 + 4 \times (10.125 + 1.125)}{8 \times 0.625 + 2 \times 10 + 2 \times 5 + 4 \times 2}$$

$$= 4.425 \text{ in}$$

$$I = \frac{8}{12} (0.625)^3 + 8 \times 0.625 \times (4.1125)^2 + \frac{2 \times 1 \times 5^3}{12} + 2 \times 5 \times (1.3)^2 + \frac{2 \times 10^3}{2 \times 1} \\ + 2 \times 10 \times (1.2)^2 + 2 \times 4 \times \frac{1^3}{12} + 4 \times 1 \times (3.3)^2 + 4 \times 1 \times (5.075)^2$$

$$= 465.175 \text{ in}^4$$

$$Q = 8 \times 0.625 \times 4.1125 + 2 \times 5 \times 1.3 + 2 \times 10 \times 1.2 + 4 \times 1 \times (3.3 + 5.075)$$

$$= 91.0625 \text{ in}^3$$

Shear Stress (At end of section near the radius)

$$\begin{aligned}\sigma_s &= \frac{VQ}{bI} \\ &= \frac{133120 \times 91.0625}{4 \times 465.175} \\ &= 6514 \text{ psi}\end{aligned}$$

Bending Stress (A

$$\begin{aligned}\sigma_b &= \frac{Mc}{I} = \frac{981763 \times 4.425}{465.175} \\ &= 9339 \text{ psi}\end{aligned}$$

$$\frac{r}{d} = \frac{2}{20} = 0.1$$

$$\frac{L}{D} = \frac{12}{36} = 0.333 \quad \text{Figure 62 Peterson} \quad K_t = 1.68$$

$$(\sigma_b)_{\max} = 1.68 \times 9339 = 15689 \text{ psi}$$

Bending Stress (At end of section away from the radius)

$$\begin{aligned}\sigma_b &= \frac{981763 \times 5.575}{465.175} \\ &= 11766.0 \text{ psi}\end{aligned}$$

$$K_t = 1.0$$

$$(\sigma_b)_{\max} = 11766.0 \text{ psi}$$

Maximum Combined Stress at Radius

$$\begin{aligned}\sigma_{\text{combined, total}} &= \sqrt{(39459)^2 + (15689)^2} \\ &= 42463 \text{ psi}\end{aligned}$$

Fig. VI.2.6.7-5 Lifting Lug - Final Recommended Design -
GJSSET & SPACER ARRANGEMENT

VI.2.7 Stress Analysis of Counterbalance

Counterbalance Analysis

This analysis includes three parts. Part 1 is a static analysis of the cantilever built-in box beam subjected to the vertical load (counterweight) not through the shear center. Part 2 finds the natural frequencies of the three degrees of freedom system. Part 3 is a fatigue analysis based on the fatigue stress.

Fig. VI.2.7.1-1 Lifting Lug Counterbalance Beam

VI.2.7.1 Static Analysis

Fig. VI.2.7.1-2 Dimension of Cross-Section

$$I_{xx} = 90.7 + \frac{1}{12} \times 0.3175 \times 10.75^3$$

$$= 123.05 \text{ in.}^4$$

$$I_{yy} = 17.2 \text{ in.}^4$$

$$J = 0.7 + \frac{1}{3} \times 10 \times 0.3125^3$$

$$= 1.32 \text{ in.}^4$$

To locate the shear center, it will be assumed that the direct shear is carried by the 9.128 in web on the channel (subscript C) and the 10.75 in plate (subscript P). Both parts bend together on parallel planes and hence have the same curvature in bending. Therefore the bending moments carried

by each of the parts are in proportion to their moments of inertia about the X axis.

$$\frac{1}{P} = \frac{M_c}{EI_c} = \frac{M_p}{EI_p} \quad \text{or} \quad \frac{M_c}{I_c} = \frac{M_p}{I_p} = \frac{MI_p}{I_c + I_p}$$

where M is the total bending moment

$$\text{Then} \quad M_c = \frac{MI_c}{I_c + I_p} \quad \text{and} \quad M_p = \frac{MI_p}{I_c + I_p}$$

The shear forces V_c and V_p will be on the same proportion as the bending moments, so that

$$V_c = \frac{VI_c}{I_c + I_p} \quad \text{and} \quad V_p = \frac{VI_p}{I_c + I_p}$$

where V is the total shear force. (W on figure).

Fig. VI.2.7.1-3 Shear Center Location

The position of the shear center

S from the channel web can be obtained

by taking moments about the point

C on the channel web, namely, $V \cdot hc = V_p \cdot h$

or

$$V \cdot hc = \frac{V \cdot I_p \cdot h}{I_c + I_p} \quad \text{or} \quad hc = \frac{I_p \cdot h}{I_c + I_p}$$

From the figure, $I_c = \frac{1}{12} \times 0.526 \times 9.128^3 = 33.33 \text{ in.}$

$$I_p = \frac{1}{12} \times 0.3125 \times 10.75^3 = 32.35 \text{ in.}^4$$

$$hc = \frac{32.35 \times 3.203}{65.68} = 1.577 \text{ in}$$

Torque on section = $2968 \times (8.17 + 1.577)$

$$= 28,929 \text{ in lbs.}$$

The torsional shear stress in thin hollow sections is given by

$$\tau = \frac{T}{2At}$$

in which T is the torque in inch pounds, t is the thickness of the section in inches, and A is the area enclosed by the mean dimension as shown below.

$$A = 9.564 \times 2.784 = 26.63 \text{ in}^2$$

The upward shear on the plate of thickness

$t_2 = 0.3125$ is given by

$$\tau = \frac{28,929}{2 \times 26.63 \times 0.3125} = 1738 \text{ psi}$$

The downward shear on the channel of thickness

$t_1 = 0.526$ is given by

$$\tau = \frac{28,929}{2 \times 26.63 \times 0.526} = 1033 \text{ psi}$$

The direct shear area is

$$9.128 \times 0.526 + 10.75 \times 0.3125 = 8.16 \text{ in}^2$$

Direct shear stress

$$\tau = \frac{2968}{8.16} = 364 \text{ psi (downward)}$$

Fig. VI.2.7.1-4 Cross Section
for Torsion

The total downward shear stress (on the channel) is

$$\tau = 1033 + 364 = 1397 \text{ psi}$$

The net upward shear stress (on the plate) is

$$\tau = 1738 - 364 = 1374 \text{ psi}$$

Bending Moments

Fig. VI.2.7.1-5 Counterbalance Beam Loading

Due to W

$$\delta_B = \frac{Wba^2}{2EI} + \frac{Wa^3}{3EI} \quad \begin{array}{l} a = 20.25 \text{ in} \\ b = 40.063 \text{ in} \end{array}$$

Due to R_B

$$\delta_B = \frac{R_B \cdot a^3}{3EI}$$

$$\therefore \frac{Wba^2}{2EI} + \frac{Wa^3}{3EI} = \frac{R_B a^3}{3EI}, \quad R_B = W \left(\frac{3b}{2a} + 1 \right) = 3.97 W \\ = 11783 \text{ lb.}$$

$$\sum F_y = R_C - R_B + W = 0$$

$$R_C = R_B - W = 2.97W = 8815 \text{ lb}$$

$$\sum M_C = M_C + W(a+b) - R_B a = 0$$

$$M_C = R_B a - w(a+b)$$

$$= 3.97W \times 20.25 - W \times 60.313$$

$$= 20.08 W$$

$$= 59,597 \text{ in. lb}$$

Fig. VI. 2.7.1-6 Reaction and Moment Diagram

The maximum bending stress occurs at point B

$$\begin{aligned} \sigma_b &= \frac{MY}{I_{xx}} \\ &= \frac{118907 \times 5}{123.05} \\ &= 4831 \text{ psi} \end{aligned}$$

VI. 2.7.2 Part Natural Frequency

It is assumed that the mass of beam is negligible compared to the mass of counterbalance, x_1 , x_2 , x_3 are three generalized coordinates. They are in vertical, twist, and sidewise direction, respectively.

Fig. VI.2.7.2-1 Counterbalance Lumped Mass System

$$\rho = \sqrt{\frac{16.34^2 + 16.34^2}{12} + 9.747^2} = 11.8 \text{ in}$$

ρ = radius of gyration with respect to shear center

$$m = \frac{2968}{32.3 \times 12} = 7.681 \text{ lbs-sec/in}^2, \quad r = 8.17'' + 1.577'' = 9.747''$$

$$\text{Total kinetic energy} = \frac{1}{2}m(\dot{x}_1 + r\dot{x}_2)^2 + \frac{1}{2}m\rho^2\dot{x}_2^2 + \frac{1}{2}m\dot{x}_3^2$$

$$\text{Total strain energy} = \frac{1}{2}ax_1^2 + \frac{1}{2}bx_2^2 + \frac{1}{2}cx_3^2$$

$$a = \frac{EI_{xx}}{29333} = \frac{30 \times 10^6 \times 123}{27666} = 133376 \text{ lbs/in}$$

$$b = \frac{GJ}{l} = \frac{12 \times 10^6 \times 0.8}{40.063} = 239622 \text{ in-lb/rad}$$

$$c = \frac{EI_{yy}}{27666} = \frac{30 \times 10^6 \times 17.2}{27666} = 18651 \text{ lb/in}$$

Calculation of a and c

Fig. VI.2.7.2-2 Stiffness Calculation

$$a = 40.063", \quad b = 20.25"$$

$$l = a + b = 60.313"$$

$$\delta_A = \frac{l^3}{3EI} = \frac{73132}{EI}$$

$$\delta_B = \frac{1}{6EI} (2l^3 - 3l^2a + a^3)$$

$$= \frac{11046}{EI}$$

$$\delta_B = \frac{P \cdot b^3}{3EI} = \frac{2668P}{EI} = \frac{11046}{EI}$$

$$\therefore P = 4.14 \text{ lb}$$

$$f = \delta_A + d_A = \frac{73132}{EI} - \frac{4.14 \times b^2}{6EI} (3l - b) = \frac{27666}{EI} \text{ in/lb}$$

$$k = \frac{EI}{27666} \text{ lb/in}$$

So it is reasonable to assume that the dynamic stress is mostly caused by sidewise vibration. The allowable amplitude δ of sidewise vibration at free end of beam is obtained as follows:

$$\sigma = \frac{My}{I} = \frac{P \cdot 40 \times 1.72}{I} = 7970 \text{ psi}$$

$$P = k \cdot \delta = \frac{EI}{27666} \cdot \delta$$

$$\therefore \delta = \frac{27666 \times 7970}{30 \times 10^6 \times 40 \times 1.72}$$

$$= 0.107 \text{ in}$$

Check the groove weld at 20" from fixed end

From AISC code, the allowable fatigue stress of groove weld with number of loading cycles greater than 2,000,000 subjected to reversal stress is 12 ksi, which is very close to the allowable fatigue stress of the box beam. So the allowable amplitude of 0.107 in. also governs the groove weld.

Check stress of bolts at fixed end

Static Shear Stress

6-5/8" ASTM grade L7 bolts

(yield stress = 100 ksi)

$$A = 0.3068 \text{ in}^2$$

Fig. VI.2.7.3-1 Counterbalance Bolt Locations

The Lagrangian equation yield

$$\left\{ \begin{array}{l} \ddot{x}_1 + r\dot{x}_2 + Ax_1 = 0 \\ r\dot{x}_1 + (r^2 + \rho^2) \dot{x}_2 = 0 \\ \ddot{x}_3 + cx_3 = 0 \end{array} \right. \quad \begin{array}{l} (1) \\ (2) \\ (3) \end{array}$$

Where

$$A = \frac{a}{m} = \frac{133371}{7.681} = 17363 \text{ in/sec}$$

$$B = \frac{b}{m} = \frac{239622}{7.681} = 31197 \text{ rad/sec}$$

$$C = \frac{c}{m} = \frac{18651}{7.681} = 2428 \text{ in/sec}$$

Equation (3) is independent from (1) and (2). The natural frequency equation of Eq. (1) and (2) is

$$\frac{2\omega^2}{A} = 1 + x^2 + y^2 \pm \sqrt{[x^2 + (y-1)^2][x^2 + (y+1)^2]}$$

where

$$x = \frac{r}{\rho} = \frac{9.747}{11.8} = 0.826$$

$$y = \left(\frac{B}{A\rho^2} \right)^{\frac{1}{2}} = \left(\frac{31197}{17363 \times 139.24} \right)^{\frac{1}{2}} = 0.113$$

yields

$$\Rightarrow \left\{ \begin{array}{l} \omega_1 = 171 \text{ rad/sec (circular velocity in } x_1 \text{ direction)} \\ \omega_2 = 11.63 \text{ rad/sec (circular velocity in } x_2 \text{ direction)} \end{array} \right.$$

$$\Rightarrow \left\{ \begin{array}{l} f_1 = 27.2 \text{ cps (natural frequency in } x_1 \text{ direction)} \\ f_2 = 1.85 \text{ cps (natural frequency in } x_2 \text{ direction)} \end{array} \right.$$

in Eq. (3)

$$\begin{aligned}
 \omega_3 &= \sqrt{C} \\
 &= \sqrt{2428} \\
 &= 49.3 \text{ rad/sec} \\
 f_3 &= 7.8 \text{ cps}
 \end{aligned}$$

VI. 2.7.3 Fatigue Analysis

Each loop handling machine operation is assumed to last approximately 20 hours. The operations are performed four times every year. The total cycles of free vibration in ten years will be:

$$\begin{aligned}
 N_1 &= (60 \times 60 \times 20 \times 4 \times 10) \times f_1 \\
 &= 78.4 \times 10^6 \text{ cycles} \quad (\text{vertical vibration}) \\
 N_2 &= (60 \times 60 \times 20 \times 4 \times 10) \times f_2 \\
 &= 5.3 \times 10^6 \text{ cycles} \quad (\text{torsional vibration}) \\
 N_3 &= (60 \times 60 \times 20 \times 4 \times 10) \times f_3 \\
 &= 22.5 \times 10^6 \text{ cycles} \quad (\text{sidewise vibration})
 \end{aligned}$$

The endurance limit of A36 steel, subjected to tension stress at room temperature, (approximately 2×10^6 cycles) is 39 ksi. If a safety factor of 3 is chosen, the allowable dynamic stress should not exceed 8 ksi (13 - 5.03 = 7.97 ksi, where 5.03 ksi is the resultant stress in static case).

Since the stiffness of the box beam in the vertical direction is much greater than it is in the sidewise direction.

The force R through C.G.

$$R = \frac{2968 \times 12}{14} = 2544 \text{ lb}$$

$$f_s = \frac{2544}{6 \times 0.3068} = 1.38 \text{ ksi}$$

Static bending stress

Fig. VI.2.7.3-2 Counterbalance Bolts and Supports

$$I_{xx} = 6 \times 0.3068 \times 3^2 = 16.56 \text{ in}^4$$

$$\sigma = \frac{2968 \times 20.25}{16.56} = 10.7 \text{ ksi}$$

$$\text{combined stress} = \sqrt{f_s^2 + \sigma^2} = \sqrt{1.38 + 10.7} = 10.75 \text{ ksi}$$

VI.2.8 STRESSES ON RESTRAINTS DURING ERECTION

VI.2.8.1 ERECTION LOADS ON LHM RESTRAINTS

During the raising of the CRADLE-LHM combination from the horizontal to the vertical position by means of the hydraulic cylinders, the upper tie-down strap is disconnected and the LHM is restrained only by the lower tie-down strap, the T-BAR-SLOT combination at the top of the cradle, and the hold-down clamps on the base plate. If the cradle is stopped suddenly in the vertical position, the LHM will have a tendency to topple forward out of the cradle and its kinetic energy will do work on the restraining elements. The forces created on the elements will depend on their stiffnesses. In the following sections, the kinetic energy of the LHM, and the stiffnesses and loads produced on the T-BAR-SLOT and the tie-down strap will be estimated. (The effect of the hold-down clamps will be neglected.)

VI.2.8.2 Kinetic Energy of LHM

The moment of inertia of the LHM about the cradle trunnion was found to be $I_0 = 4.009 \times 10^6 \text{ lb-sec}^2\text{-in}$ on Table VI.2.8.2-1 and Fig. VI.2.8.2-1. About the center of gravity it would be

$$\begin{aligned} I_{CG} &= 4.009 \times 10^6 - \frac{91808}{386} (156.9 - 70)^2 \\ &= 2.213 \times 10^6 \text{ lb-sec}^2\text{-in.} \end{aligned}$$

As a rough check, an approximation using a symmetrical cylinder yields

$$K_{xx}^2 = K_{yy}^2 = \frac{1}{12}(3r^2 + h^2)$$

$$r_{ave} = \frac{27 + 29}{4} = 14''$$

$$h = 28' = 336''$$

$$K_{xx}^2 = \frac{113484}{12} = 9457 \text{ in}^2$$

$$I_{CG} = \frac{91808}{386} \times 9457 = 2.249 \times 10^6 \text{ lb-sec}^2\text{-in}$$

Table VI. 2.8.2-1 LHM-Loop Moment of Inertia Computation (Refer to Fig.VI. 2.8.2-1)

LHM COMPONENTS	W (lbs)	X ₁ (in)	X ₂ (in)	X ₂ ² (in) ²	K ² (in) ²	$\frac{W}{386}(X_2^2 + K^2)(\text{lbs-sec}^2\text{-in})$
Shield Valve Assembly	3560	4.0	66.0	4356	1105	50,336
Mainbody - Lower Half						
Flange - Upper	1205	150.3	80.3	6448	153	20,607
Flange - Lower	3075	14.5	55.5	3080	245	26,488
Loop Support - Upper	435	150.9	80.9	6545		7,376
Loop Support - Lower	430	16.3	53.7	2884		3,213
Outer Shell 13.8 #/in	1781	83.9	13.9	193	1503	7,825
Inner Shell 2.4 #/in	301	83.9	13.9	193	1440	1,273
Shield - Upper Section 122.6 #/in	5647	125.3	55.3	3058	243	48,292
Shield - Lower Section 216.9 #/in	18003	60.8	9.2	85	600	31,948
Chain Break Box	305	137.0	67.0	4489	36	3,575
Hold Down Strap	100	126.0	56.0	3136		812
Grapple Drive Assembly	6710	65.0	5.0	25	692	12,464
Mainbody - Upper Half						
Flange - Upper	709	341.6	271.6	73766	208	135,874
Grapple and Sleeve	294	134.6	264.6	70010	132	180,575

Table VI.2.8.2-1

LHM-Loop Moment of Inertia Computation (cont'd)

LHM COMPONENTS	W (lbs)	X ₁ (in)	X ₂ (in)	X ₂ ² (in) ²	K ² (in) ²	$\frac{W}{386}(X_2^2 + K^2)$ (lbs-sec ² -in)
Outer Shell 18.5 #/in	3394	248.6	178.6	31898	1802	296,315
Inner Shell 2.3 #/in	422	248.6	178.6	31898	1767	36,805
Shield 122.6 #/in	22489	248.6	178.6	31898	1782	1,962,253
Lifting Lugs (2)	1219	189.4	119.4	14256	366	92,353
Counterweights (2)	3160	201.4	131.4	17266	120	284,662
Shackles (2)	80	237.9	167.9	28190		11,685
Hold Down Strap	100	253.0	183.0	33489		8,676
Dome Assembly	854	346.1	276.1	76231	200	169,099
Chain Conduit	234	230.0	160.0	25600	2804	17,219
Grapple Drive Mnt'g Angles	620	83.9	13.9	193	1387	2,538
LHM SUBTOTALS						
Total WT (Computed)	80749					
Net WT (Measured)	81050	171.8	1.8	10363		33,586
Net/Total	1.5%					

Table VI. 2.8.2-1 LHM-Loop Moment of Inertia Computation (cont'd)

LOOP COMPONENTS	W (lbs)	X ₁ (in)	X ₂ (in)	X ₂ ² (in) ²	K ² (in) ²	$\frac{W}{386}(X_2^2+K^2)$ (lbs-sec ² -in)
Miscellaneous (Upper)	790	313.3	243.3	59195	-	121,150
Heater	3355	255.8	185.8	34522	-	300,055
Pump	5224	169.3	99.3	9860	-	133,442
Miscellaneous (Lower)	439	79.3	9.3	86.5	-	98
LOOP SUBTOTALS	9808					
LHM and LOOP TOTALS	91808	about cradle trunnion I ₀ = 4.009 x 10 ⁶ lb-sec ² -in				
		(X ₁) _{CG} = 156.9 in.				

Table VI.2.8.2-2 Elevating Cylinders

Stage	Bore	Area	Stroke (approx)
1	10.5 in	86.59 in ²	22.0
2	9	63.62	23.9
3	7.5	44.13	26.5
4	6.5	33.18	28.1
5	5.5	23.76	29.8
			<hr/> 130.3

Closed Length = 51 in

Open Length = 181.3 in

TOTAL WEIGHT = 91808 #

- : x_1 = DISTANCE FROM LOWER END (IN)
- : x_2 = DISTANCE FROM CRADLE TRUNNION (IN)
- : K = RADIUS OF GYRATION OF WEIGHT W (IN)

COORDINATES IN TABLE VI.2.8.2-1

FIG. VI.2.8.2-1 COORDINATES IN TABLE VI.2.8.2-1

The geometry of the CRADLE-LHM-HYDRAULIC CYLINDERS combination during the raising of the CRADLE-LHM to the vertical position is shown on Fig. 2.8.2-2. From this geometry an analytical solution for the angular velocity of the LHM can be obtained.

From Fig. VI.2.8.2-2

$$S \cos (180 - \phi) + R \cos \theta = b$$

$$S \sin (180 - \phi) - R \sin \theta = a$$

Simplifying,

$$- S \cos \phi = b - R \cos \theta \quad (1)$$

$$S \sin \phi = a + R \sin \theta \quad (2)$$

Squaring both sides

$$S^2 \cos^2 \phi = b^2 + R^2 \cos^2 \theta - 2bR \cos \theta \quad (3)$$

$$S^2 \sin^2 \phi = a^2 + R^2 \sin^2 \theta + 2aR \sin \theta \quad (4)$$

Adding (3) and (4)

$$S^2 = a^2 + b^2 + R^2 - 2bR \cos \theta + 2aR \sin \theta$$

Using numerical values of a, b, and R. (Fig. 2.8.2-2)

$$S^2 = 24047 (1 + 0.4736 \sin \theta - 0.8786 \cos \theta) \quad (5)$$

Differentiating (5) with respect to time

$$2S \frac{ds}{dt} = 21127 \sin \theta \frac{d\theta}{dt} + 11388 \cos \theta \frac{d\theta}{dt}$$

$$\frac{d\theta}{dt} = \omega = \frac{2S \frac{ds}{dt}}{21127 (\sin \theta + 0.539 \cos \theta)}$$

For last stage of elevating cylinders, bore = 5.5", area = 23.76 in².

(Table VI.2.8.2-2 For 17 gals/min from pump to two sets of cylinders)

$$\frac{ds}{dt} = \frac{17 \times 231}{60 \times 2 \times 23.76} = 1.377 \text{ in/sec}$$

A = CYLINDER PIVOT ON TRAILER

B = CYLINDER PIVOT ON CRADLE

C = CRADLE TRUNNION

S = CYLINDER EXTENSION LINE

$$\phi_{\min} = -1.5^\circ, S_{\min} = 5127''$$

$$\epsilon = \sin^{-1}\left(\frac{24}{112.98}\right) = 12.26^\circ$$

$$\phi_{\max} = 90 - 12.26 = 77.74^\circ, S_{\max} = 175.2''$$

$$\begin{aligned} a &= 50.4'' \\ b &= 93.5'' \\ R &= 112.98'' \end{aligned}$$

FIG. VI.2.8.2-2 CRADLE - LHM ERECTION GEOMETRY

At $\theta_{\max} = 77.74^\circ$, $S = 175.2''$

$$\omega = \frac{2 \times 175.2 \times 1.377}{21127(0.977 + 0.539 \times 0.212)} = 0.0209 \text{ rad/sec}$$

Because the CRADLE, carrying the LHM with it, rotates about the trunnion, the LHM may be considered as rotating about a fixed axis - namely, the trunnion - and its kinetic energy will be given by

$$\begin{aligned} \text{KE} &= \frac{1}{2} I_o \omega^2 \\ &= \frac{1}{2} \times 4.009 \times 10^6 \times (0.0209)^2 \\ &= 876 \text{ in lbs} \end{aligned}$$

VI. 2.8.3 LOAD ON T-BAR AND SLOT

It is estimated that the maximum load on the T-BAR and SLOT will be obtained by assuming that it provides a rigid connection between the CRADLE and the LHM. During a sudden stoppage, the LHM pivots freely about the base plate and is restrained by only the T-BAR and SLOT. This bends the CRADLE like a cantilever beam fixed at the fourth cross beam which is 45 inches above the trunnion. (See Fig. VI.2.8.3-1)

CRADLE STIFFNESS.

From Fig. VI. 2.8.3-1

$$\begin{aligned} \Delta_2 &= \frac{P}{E} \left[\frac{L_B^3}{3I_B} + \frac{L_A L_B^2}{2I_B} + \frac{L_A L_B^2}{2I_B} + \frac{L_A^2 L_B}{I_B} + \frac{L_A^3}{3I_A} \right] \\ &= \frac{P}{E} \left[\frac{1}{I_B} \left(\frac{L_B^3}{3} + L_A L_B^2 + L_A^2 L_B \right) + \frac{L_A^3}{3I_A} \right] \quad (\text{Refer to Figs. V.2.2-1, V.2.2-2}) \end{aligned}$$

$$L_A = 99.35'', L_B = 38.75'', I_A = 580 \text{ in}^4, I_B = 712 \text{ in}^4$$

$$\Delta_2 = \frac{P}{E} 3220, K_c = \frac{P}{\Delta_2} = \frac{30 \times 10^6}{3220} = 9317 \text{ lb/in}$$

Fig. VI.2.8.3-1 Stiffness Formulas for Cradle
and Tie-Down Strap

From section VI.2.8.2 the LHM kinetic energy on reaching the vertical position is 876 in-lbs. Work done in bending the cradle in terms of maximum load P and deflection Δ

$$= \frac{1}{2} P \Delta = \frac{1}{2} \frac{P^2}{K_c}$$

then

$$\begin{aligned} \frac{1}{2} \frac{P^2}{K_c} &= 876, \text{ or } P = \sqrt{2 \times 9317 \times 876} \\ &= 4040 \text{ lbs} \end{aligned}$$

This is the load acting on the T-bar and T-slot combination.

VI.2.8.4 LOAD ON TIE-DOWN STRAP (See Fig. VI.2.8.3-1)

Because the strap is wrapped around the cask, when loaded, the deflection $L_1/2$, (because of the restraint imposed by friction between the strap and the drum) will not stretch as much as an unrestrained strap. This restraint results in the load at the mid-point of the strap being less than the load at either end. This difference can be estimated by the relationship between the tight and slack side tensions on a band brake or leather belt drive. For the tie-down strap, this relationship will give a larger difference than actually exists, resulting in a greater stiffness and higher stresses.

The relation between tight (T_1) and slack side tensions (T_2) is given by

$$T_1 = T_2 e^{\mu \theta}$$

in which μ is the coefficient of friction and θ the angle of wrap. For a stainless steel strap on the cask, assume $\mu = 0.3$. Then for $\theta = \pi/2$

$$T_2 = \frac{T_1}{e^{\mu \theta}} = \frac{T_1}{e^{0.3\pi/2}} = 0.625 T_1$$

To obtain the factor C in the formula for Δ in Fig. VI.2.8.3-1, consider a bar of length L under a load varying linearly from 0.625 T to T.

$$\text{Load at } x = 0.625T + 0.375 \frac{x}{L}T$$

Then increase in length L

$$\begin{aligned} \delta &= \frac{T}{AE} \int_0^L (0.625 + 0.375 \frac{x}{L}) dx \\ &= \frac{T}{AE} [0.625L + 0.375 \frac{L}{2}] \\ &= \frac{TL}{AE} 0.8125 \end{aligned}$$

Then factor C = 0.8125.

For the strap,

$$\Delta = \frac{P}{4E} \left[\frac{L_1 C}{A_1} + \frac{L_2}{A_2} \right] \quad (\text{Fig. VI. 2.8.3-1})$$

For the cask

$$D = 29.75 \text{ in}$$

$$L_1 = 29.75 \times \pi/2 = 46.73 \text{ in}$$

$$A_1 = 3.75 \times 0.5 = 1.875 \text{ in}^2$$

$$L_2 = 9 \text{ in.}$$

$$A_2 = \frac{\pi}{4} = 0.7854 \text{ in}^2$$

$$\Delta = \frac{P}{4E} [31.71]$$

$$K_S = \frac{P}{\Delta} = \frac{4E}{31.71} = 3.784 \times 10^6 \text{ \#/in}$$

From Section VI.2.8.2 the LHM kinetic energy is 876 in.-lbs. Work done in stressing the strap mechanism in terms of maximum load P and deflection Δ

$$= \frac{1}{2} P \Delta = \frac{1}{2} \frac{P^2}{K_s}$$

Then $\frac{1}{2} \frac{P^2}{K_s} = 876$, or $P = \sqrt{2 \times 3.784 \times 10^6 \times 876}$
 $= 81,422\#$

Load on strap $= \frac{81422}{2} = 40711 \# = \frac{P}{2}$

VI.2.8.5 STRESSES IN TIE-DOWN STRAP

From Fig. V.3.1.6-2 the area of the strap is $3.75 \times 0.5 = 1.875 \text{ in}^2$.

Because it encircles the cask, the stress on the strap will be

$$\sigma_T = \frac{40,711}{1.875} = 21,713 \text{ psi}$$

For a yield point of 30,000 psi in the stainless steel, strap factor of safety

$$FS = \frac{30,000}{21,713} = 1.38$$

VI. 2.8.6 T-SLOT ATTACHMENT WELD STRESSES DURING ERECTION

The T-slot attachment is shown in Fig. V.2.8-1, and the weld pattern in Fig. VI. 2.8. 6-1. Assuming that all of the stresses produced by the load P are carried by the welds, the neutral axis will pass through the geometric center of the pattern and the moment of inertia will be

$$I = 10 \left[\frac{1}{12} \times \frac{3}{8} \times 2^3 \right] + 4 \times \frac{3}{8} \times 2 \left[8^2 + 4^2 \right] + 2 \times 5.5 \times \frac{3}{8} \times 10^2$$

$$= 655 \text{ in}^4$$

During erection $P = 4040 \text{ lb}$ (Section VI.2.8.3)

Weld area $A = 10 \times \frac{3}{8} \times 2 + 2 \times 5.5 \times \frac{3}{8} = 11.625 \text{ in}^2$

Bending moment $M = 4040 \times 6 = 24,240 \text{ in. lbs.}$

At weld AB,

$$\sigma = \frac{MC}{I} + \frac{P}{A} = \frac{24,240 \times 10}{655} + \frac{4040}{11.625}$$

$$= 370 + 348 = 718 \text{ psi (tension)}$$

At weld CD,

$$\sigma = -370 + 348 = -22 \text{ psi (compression)}$$

Maximum shear stress (on throat)

$$\tau = \frac{718}{0.707} = 1015 \text{ psi}$$

$$SF = \frac{28,000}{1015} = 19.7$$

$$P = 4040/b \text{ (SECT. VI.2.8.3)}$$

FIG. VI.2.8.6-1 - T-SLOT WELD PATTERN

VI.2.8.7 STRESSES IN T-BAR BOLTS

The bolt pattern and the position of load P are shown in Fig. V.3.1.5-1. Assuming that the T-Bar pivots about the right hand edge, and neglecting any tightening load that may have been applied, the maximum bolt load will be in bolts nearest the external load P; call the force in these bolts F_5 and the force in the other bolts F_4 , F_3 , F_2 , and F_1 . The latter forces will be proportional to their distances from the right hand edge. From Section VI.2.8.3 the load P is 4040 lbs. Then

$$4040 \times 14 = F_5 \left[10.5 + \frac{8.75^2 + 6^2 + 3.25^2 + 0.5^2}{10.5} \right]$$

$F_5 = 2542$ lb and the load per bolt $\frac{2542}{2} = 1271$ lb.

The bolts are 1/2" - 13 UNC with a stress area $A_s = 0.1419$ in.²: Then the maximum bolt stress

$$\bar{\sigma} = \frac{1271}{0.1419} = 8957 \text{ psi}$$

$$SF = \frac{100,000}{8957} = 11.2$$

Since it is difficult to make an appropriate assumption of tightening in the bolts, a final factor of safety of 5.25 is assumed, and a tightening stress is computed using the yield strength of the bolts in the following calculations.

Assume tightening stress

$$\sigma = \frac{100,000}{5.25} \approx 19,000 \text{ psi}$$

$$\text{Tightening or pre-load } F_0 = 19,000 \times 0.1419 = 2696 \text{ \#/bolt}$$

To estimate relative stiffnesses

(K_J and K_B) of joint and bolt, assume

bolt compresses area around bolt

shown at left:

$$A = \frac{\pi}{4} (9d^2 - d^2) = 2\pi d^2$$

Then

$$\frac{K_J}{K_B} = \frac{2\pi d^2}{\frac{\pi d^2}{4}} = 8$$

Center of gravity of bolt pattern

$$\bar{y} + S = \frac{2A(S + 2.75 + S + 5.5 + S + 8.25 + S + 10 + S)}{10A}$$

$$= \frac{2A(5S + 26.5)}{10A} = S + 5.3$$

$$\bar{y} = 5.3 \text{ in}$$

$$M = 4.7P_5 + 2.95P_4 + 0.2P_3 + 2.55P_2 + 5.3P_1$$

$$= P_5 \left[4.7 + \frac{2.95^2}{4.7} + \frac{0.2^2}{4.7} + \frac{2.55^2}{4.7} + \frac{5.3^2}{4.7} \right]$$

$$P_5 = \frac{M}{13.92}$$

$$M = 4040 \times 8.2 = 33,128 \text{ in lb}$$

$$P_5 = \frac{33,128}{13.92} = 2380 \text{ lb}$$

Applied maximum bending load per bolt

$$P_B = \frac{2380}{2} = 1190 \text{ lb}$$

Applied direct load per bolt

$$P_D = \frac{4040}{10} = 404 \text{ lb}$$

Total maximum applied tensile load per bolt

$$P = P_B + P_D = 1190 + 404 = 1594 \text{ lb}$$

Resultant maximum tensile load per bolt

$$F_B = F_O + \frac{P}{1 + \frac{K_J}{K_B}} = 2696 + \frac{1594}{1 + 8} = 2873 \text{ lb}$$

$$\sigma_B = \frac{2873}{0.1419} = 20,247 \text{ psi}$$

$$SF = \frac{100,000}{20,247} = 4.94$$

Load on joint

$$F_J = F_O - \frac{P}{1 + \frac{K_B}{K_J}} = 2696 - \frac{1594}{1 + \frac{1}{8}} = + 1279 \text{ lb}$$

∴ Joint has not opened.

VI. 2.9 STRESS ANALYSIS OF LHM WELDS AND CONNECTIONS

Fig. VI.2.9-1 LHM Cask

The LHM Cask is required to withstand a $10g$ horizontal load without failure in either the flange welds or the flange bolts.

WELDS A

These welds must carry the total weight of the LHM, plus the loop, minus the lower flange and the shield valve. From Table II.2

$$W = 91808 - 3075 - 3560 = 85173 \text{ lb}$$

The weld pattern is shown in Fig. V.2.2-2. The butt weld has a transverse or throat area

$$A_1 = \frac{\pi}{4}(29.75^2 - 28.75^2) = 45.95 \text{ in}^2$$

The five gussets have a total throat area of

$$A_2 = 5 \times 2 \times 4.5 \times 0.25 \times 0.707 = 7.95 \text{ in}^2$$

The total load is carried by both the butt weld and the gusset welds, but the division of the load between the two weld patterns is indeterminate. Because the butt weld has a much greater area than the gusset welds, the latter will be neglected in arriving at a factor of safety.

The butt weld is in tension across the throat and consequently the tensile stress is

$$\sigma_t = \frac{85173 \times 10}{45.95} = 18,536 \text{ psi}$$

For a yield stress of 39,000 in the stainless steel weld, the factor of safety is

$$SF = \frac{30,000}{18,536} = 1.62$$

WELD B

This weld is the same as the butt weld at A - there are no gussets; the transverse area is 45.95 in^2 . The load on this weld is equal to the upper half of the main body, plus the loop, plus the upper flange on the lower body, plus the lower shield. From Table II.2,

$$W = 39188 + 9808 + 1205 + 5647 + 18003 = 73851 \text{ lb.}$$

$$\sigma_t = \frac{73851 \times 10}{45.95} = 16,072 \text{ psi}$$

$$SF = \frac{30,000}{16,072} = 1.87$$

WELD C

From Fig. V.2.2.-2 the transverse area of the butt weld is

$$A = \frac{\pi}{4}(27^2 - 25.5^2) = 61.85 \text{ in}^2$$

The load carried by this weld is equal to the upper half of the main body, minus the lower flange, plus the loop.

$$W = 39188 - 454 + 9808 = 48,542 \text{ lb}$$

$$\sigma_t = \frac{48,542 \times 10}{61.85} = 7848 \text{ psi}$$

$$SF = \frac{30,000}{7848} = 3.82$$

WELD D

The transverse area of this butt weld is the same as Weld C, namely, 61.85 in^2 . The load carried is equal to the upper flange, plus the dome assembly, plus the upper shield.

$$W = 709 + 854 + 22489 = 24052 \text{ lb}$$

$$\sigma_t = \frac{24052 \times 10}{61.85} = 3889 \text{ psi}$$

$$SF = \frac{30,000}{3889} = 7.71$$

BOLTS

The flanges between the B and C welds are bolted together with 12, 3/4-10 UNC bolts having a total stress area of $12 \times 0.334 = 4 \text{ in}^2$. The load on the bolts is equal to the upper half of the main body plus the loop.

$$W = 39188 + 9808 = 48,996 \text{ lb}$$

Neglecting the tightening stress, (i.e. assuming that the flanges will separate) the stress in the bolts is

$$\sigma = \frac{48,996 \times 10}{4} = 122,490 \text{ psi}$$

Actually the bolts were tightened to some unknown stress. If that stress were below the stress required to keep the flanges from separating, it would have no effect on the calculated stress; if it were above the stress required to keep the flanges from separating, the calculated stress would be higher.

Assuming medium carbon steel quenched and tempered bolts for which the ultimate strength is 125,000 psi

$$SF = \frac{125,000}{122,490} = 1.02$$

The upper flange on the upper half of the main body is bolted to the dome assembly by 15, 5/8-11 UNC cap screws having a total stress area of $15 \times 0.226 = 3.39 \text{ in}^2$. The load on the screws is equal to the dome assembly plus the loop, plus the grapple.

$$W = 9808 + 854 + 994 = 11656$$

$$\sigma = \frac{11656 \times 10}{3.39} = 34,383 \text{ psi}$$

$$SF = \frac{125,000}{34383} = 3.6$$

VI.2.10 STRESS ANALYSIS OF LOOP SUPPORTS

In Section II.1 the maximum downward reaction on the loop supports was found to be 8203 lb. Weight of the support assembly with actuator is 145 lbs. For the 2g requirement, the support must sustain

$$W = 2(8203 + 145) = 16,696 \text{ lb}$$

The actuator housing is attached to the LHM by four 5/16-18 UNC cap screws that are screwed to a depth of 5/8 in. in the cast iron housing.

On account of the high load on the cap screws under the 2g load, it will be necessary to calculate the initial tightening load F_o required to keep the joint from separating.

To estimate the relative stiffnesses of joint (K_J) and screw (K_S), assume that the screw compresses an area around the screw equal to three times the screw diameter. Then the compressed area of each joint

$$A_J = \frac{\pi}{4}(9d^2 - d^2) = 2\pi d^2$$

and $K_J/K_S = 2\pi d^2 \div \pi/4d^2 = 8$. For zero joint load

$$F_J = F_o - \frac{P}{1 + \frac{K_S}{K_J}} = 0$$

or

$$F_o = \frac{16,696 \times 8}{4 \times 9} = 3710 \text{ lb.}$$

$$\text{Then load on screw, } F_S = F_o + \frac{P}{1 + \frac{K_J}{K_S}} = \frac{16,696}{4} = 4174 \text{ lb.}$$

$$F_S = 3710 + \frac{16,696}{4 \times 9} = 4174 \text{ lb}$$

For a stress area of 0.0524 in^2

$$\sigma_T = \frac{4174}{0.0524} = 79,656 \text{ psi}$$

For medium carbon steel screws

$$SF = \frac{100,000}{79,656} = 1.26$$

In this case F_o has no effect on the bolt stress. A higher F_o to prevent separation would create a higher bolt stress.

The total shear area of the threads on the cast iron is

$$A_S = \pi(4)(0.3125)(0.625) = 2.45 \text{ in}^2$$

For the high grade cast iron in the housing, $\tau_{ULT} = 50,000 \text{ psi}$
the load to shear off the threads is

$$W_{ULT} = 50,000 \times 2.45 = 122,500 \text{ lb.}$$

The total load on the screws is

$$W = 4 \times 4174 = 16,696 \text{ lb}$$

$$SF = \frac{122,500}{16,696} = 7.34$$

VII. LHM THERMAL CONSIDERATIONS

VII.1. LHM Thermal Analysis

Steady-state and transient thermal analyses were performed in Ref. 20 by EG & G Company of Idaho Falls to define the thermal conditions required or present in the SLSF and LHM cask during loop storage in the LHM with minimum loop pump power. Also, the thermal conditions during loop "quench" when the loop is inserted into the ETR with zero loop pump power.

The minimum SLSF sodium pump power necessary to maintain a steady-state sodium temperature of 450°F, while stored in the LHM, was determined to be approximately 3kW. The axial and radial steady-state temperature gradients, which occur during loop storage, are shown in Figures VII.1.6-1 and VII.1.6-2.

The maximum axial and radial transient temperature gradients which occur during an extended insertion sequence were determined to be at 165 minutes.

VII.1.1 INTRODUCTION

A modification of the SLSF thermal-hydraulic model Reference 21, to incorporate the loop handling machine (LHM) cask and the loop insertion sequence from the LHM into the reactor cooling water, is described in this report. The following thermal analyses were the required objectives of this study:

1. While in the storage or "holding" posture, determine the pump power required to maintain the sodium temperature at or above 450°F, and the resulting steady-state temperature gradients throughout the SLSF and LHM cask.
2. Determine transient temperature gradients in the SLSF during loop "quench". i.e., loop insertion into the ETR water vessel from the LHM holding posture with zero pump power.
3. Determine the minimum length of time in which sodium freezing would occur during an extended SLSF insertion operation.

The CINDA-3G thermal computer program (Reference 22) was used to generate steady-state and transient thermal gradients. A general description of the nodal network used, material properties, film coefficients, and analysis results follow.

VII.1.2 NODAL NETWORK

The thermal-hydraulic model used in this analysis is a modification of the model described in Reference 21, the major change being the addition of the LHM cask around the loop. Minor changes include substitution of stagnant argon for the purge and coolant helium, and a reduction in the number of nodes and conductors in the original model. An additional transient sequence was generated to simulate a constant rate of loop insertion into the ETR.

The model was programmed by the use of the CINDA-3G thermal code, Reference 22 and FORTRAN logical operations.

The nodal model is illustrated in Figures VII.1.2-1,-2,-3,-4, featuring details of the LHM cask and a generalized layout of the SLSF in the storage or "holding" posture. In this position, the loop is bounded by an air gap between the loop and the cask, and ambient air on the outer side of the LHM cask. Radiation as well as air conduction was considered between the loop and cask. During the transient insertion sequence, the loop is bounded by the ETR cooling water, into which it is quenched, and has flowing air above the water level.

The thermal network contains a total of 637 nodes of which 30% comprise the LHM cask, and a total of 870 conductors of which 35% are related to the cask.

VII.1.3 MATERIAL PROPERTIES

Structural material throughout the LHM cask is 316 stainless steel. Depleted uranium is used as shielding between the inner and outer cask skins. A length of aluminum alloy is used as a structural filler in the lower cask body. Fabrica, a rubber-fabric material, is incorporated in the cask support saddles. Details of the LHM cask are shown in Figures VII.1.2-1,-2,-3,-4.

Table VII.1.3-1 lists the thermal and physical properties of the LHM cask materials and their values, plus those of the argon used in the loop. References are given in parentheses for temperature-dependent values. Properties previously used in the SLSF portion of the model are noted in Reference 21.

SECTION D-D (LHM CASK SECTION) **SECTION D-D (FEFPL SECTION)** **Thermal-hydraulic Model**

Figure VII.1.2-1 - Node Diagram for SLSF and Handling Machine Cask Thermal Model

UPPER LHM CASK BODY

Figure VII.1.4-2 Node Diagram for SLSF and Handling Machine Cask Thermal-Hydraulic Model

SECTION A-A

Figure VII.1.2-3 - Node Diagram for SLSF and Handling Machine Cask Thermal-Hydraulic Model

Figure VII.1.2-4 - Node Diagram for SLSF and Handling Machine Cask Thermal-Hydraulic Model

TABLE VII.1.3-1

SLSF & LHM Material Properties

	Density <u>LB/FT³</u>	Specific Heat <u>Btu/LB-°F</u>	Conductivity <u>Btu/HR-FT</u>
316 SS	488.	(3)	(3)
U Shielding	1169.	(4)	(5)
Argon	(8)	(6)	(7)
Aluminum (1100)	169.	(9)	(9)
Fabrica	60.	.50	.10

Note: Numbers in parentheses are reference for temperature dependent properties

VII.1.4 Film Coefficients

In the air gap or enclosed air space between the loop and cask bodies in the holding posture, free convection was calculated using the overall heat transfer coefficient U, Reference 30 which includes pure air conduction between the surfaces. For the lower portion of the loop where the loop-cask air gap is relatively large, the Grashof number (Gr) is calculated to be greater than 2×10^5 indicating fully turbulent flow. Therefore, the relationship $U = \frac{k}{b} (.0426) Gr^{.37}$ was used where the characteristic dimension b is the air gap width Reference 50. In the enclosed air space between the upper portion of the loop and the cask where b is relatively small, Gr is calculated to be less than 2000. Therefore, $\frac{Ub}{k} = 1$ and heat flows only by conduction, or $U = k/b$ Reference 30.

Free convection in laminar flow between the LHM cask outer surface and ambient air was calculated using the relationship $Nu = .555 (GrPr)^{.25}$, Reference 30.

During the SLSF insertion sequence from the LHM into the ETR water vessel,

the loop is, in effect, quenched. Due to the relatively low surface temperatures of the loop before quench (less than 400°F), it was assumed that film boiling does not occur since stable film boiling usually requires a ΔT between the surface temperature and water saturation temperature of at least 500°F, Reference 32. Nucleate boiling was assumed to occur between a ΔT (at water/loop interface) of 10°F, up through the maximum ΔT that occurs of less than 200°F. When the loop surface temperature drops below 10 degrees above the water saturation temperature of 203°F, forced convection is assumed to occur.

The nucleate boiling film coefficients were obtained from test data evaluated in Reference 33. The coefficients vary from 600 Btu/ft²-hr-°F at a ΔT of 10°F, to a maximum of 2000 Btu/ft²-hr-°F at a ΔT of 100°F, then dropping to 1000 Btu/ft²-hr-°F at a ΔT of 200°F.

A forced convection film coefficient for an ETR water coolant flowrate of 2000 gpm was calculated using Wiegand's relationship for turbulent flow in an annulus,

$$Nu = .023 Re^{.8} Pr^{.4} \left(\frac{d_2}{d_1} \right)^{.45}, \text{ Reference 34.}$$

As the loop is lowered into the ETR vessel, the water film coefficient varies from 32.4 to 106. to 493 Btu/ft²-hr-°F as the annulus diameters vary.

During the insertion sequence, the loop is exposed to flowing air between the coolant water top surface and the bottom of the LHM cask. A forced convection turbulent flow film coefficient was calculated using the relationship $Nu = C Re^n$, which gives a film coefficient value of 1.6 Btu/ft²-hr-°F for a Reynolds number of 10,000, Reference 35.

VII.1.5 RESULTS OF STEADY-STATE ANALYSIS

The SLSF will be stored in the LHM while out of the reactor. While in the LHM, the sodium pump power is adjusted to maintain a sodium temperature of 450°F or greater. Of interest are the steady-state thermal gradients resulting in both the loop and the LHM cask.

The loop, as situated in the LHM, is bounded rather loosely from a modeling standpoint with both the ambient air outside the LHM and the air gap between the loop and LHM. As a result of the loose bounding, over 5000 iterations were required to reach a steady-state convergence factor of 0.20 with reasonable initial temperature estimations. A pump power of 3 kW was required to maintain the sodium temperature above 450°F during the steady-state condition. Figure VII.1.5-1 shows the resulting steady-state temperature of the loop and cask structural components superimposed on a layout diagram. Figure VII.1.5-2 gives detailed temperatures of the middle flange region of the LHM cask. Maximum axial temperature gradients occur in the upper and lower pump regions in both the loop secondary wall and the inner LHM cask skin. The maximum axial gradient between the cask middle flange and the cask body is approximately 118°F. There are little or no radial temperature gradients across the primary or secondary walls of the loop. However, a 73°F gradient exists between the inner and outer cask skins just above the middle flange.

Appendix A (included in Volume IV of this report) contains a computer printout of the steady-state temperatures of all loop and cask nodes.

VII.1.6 Results of Transient Analysis

To insert the SLSF into the ETR reactor, it will be lowered from the LHM situated above the reactor vessel. The sodium pump must be disconnected during the entire insertion operation, therefore, no heat source would be available to the sodium to maintain its previously constant temperature.

Figure VII.1.5-1 SLST LOOP IN LHM CASK-SS TEMPERATURES (°F) WITH PUMP POWER 3KW

301

Figure VII.15-2 LHM CASK FLANGE DETAIL SS TEMPERATURES WITH ~~SLST~~ PUMP POWER = 3kw

Of interest are the transient thermal gradients produced in the loop as it is quenched in the reactor coolant water, and the amount of time available before sodium solidification or freezing takes place. Pure sodium freezes at approximately 208°F, Reference 36.

The loop can be lowered into the reactor at a rate varying from 0 to 16 ft/min. The maximum rate of 16 ft/min was used in this analysis since it would produce the most severe thermal gradients during loop quench. Due to limitations of the nodal model, the loop was programmed to be lowered in a step function sequence, each step being the length of the succeeding node.

Figure VII.1.6-1 gives pertinent transient temperature data for the secondary vessel wall during loop quench, with zero pump power, and a loop insertion rate of 16 ft/min. Since the largest gradients were experienced in the pump region, most of the data shown are from that area.

Since relatively large thermal gradients were experienced across the loop secondary wall during the quenching operation, three supplementary nodes were generated across the secondary wall in both the core and pump regions to define the shape of the radial gradients in those areas. Figure VII.1. 6-2 presents the data from these nodes at the time the maximum thermal gradients occur. It should be noted that if the pump power during steady-state holding in the LHM is greater than 3 kW, not only do the sodium and loop temperatures rise accordingly, but the transient gradients produced during loop quench also rise since the secondary wall temperature is immediately lowered to near the water saturation temperature as the quenching takes place.

Figure VII.1.6-1 Temperature History for Nodes 64, 5362, 5561 & 65, 5761 & 62, 5962, 6162

Figure VII.1.6-2 TEMPERATURE GRADIENTS ACROSS LOOP SECONDARY WALL AT CORE & PUMP LEVELS

Figure VII.1.6-3 SODIUM TEMPERATURE HISTORY FOR LOOP INSERTION SEQUENCE INTO ETR

In order to determine the minimum time available before sodium freezing takes place, the transient analysis was continued with the loop fully inserted in the reactor. The sodium temperature reached 208°F, the pure sodium freezing point, in the pump region 165 minutes after the quench operation was initiated, as shown in Figure VII.1.6.-3. This figure can be used to determine freezing times for contaminated sodium if the new freezing point is known.

VII.1.7 Conclusions and Recommendations

A minimum pump power of 3 kW is required to maintain a sodium temperature of 450°F when the SLSF is situated in the LHM. FEFPL steady-state and transient temperature gradients, both axial and radial, are greatest in the pump region as shown in Figures VII.1.5-1 and VII. 1.6-1. If the loop remains in the reactor for an extended period time with zero pump power, pure sodium will begin freezing in the loop after approximately 165 minutes.

Fig. VII.2-1 Terminology of Temperature Gradient at LHM Flange

Fig. VII.2-2 Effect of Temperature Gradient at LHM Flange

VII.2 THERMAL STRESS EFFECTS IN LHM LOCATIONS

It is recognized that the Loop Handling Machine is subjected to thermal stresses since it is maintained at higher temperatures inside and ambient conditions outside. Specific areas that are likely affected are the needle point loop support and the LHM flanges. The two ambient conditions considered are 130°F (normal day) and -40°F (cold day). This analysis was performed by EG & G and is separately reported in their stress report but is included here for completeness.

SHELL THERMAL STRESSES ADJACENT TO "COLD" NEEDLE POINT RING

Referring to the -40° ambient condition, the almost vertical gradient in the shells is sketched as in Fig. VII.2.-1.

- (a) Because of insufficient data the slope is indeterminate.
- (b) It is so close to being vertical the calc stress would be extremely high, even if the slope were known.
- (c) Assuming more data points would give a more realistic slope, it is felt that the leveled out portion of the temp curve is still good, and it will be assumed that the best approximation which can be made at this time to that condition is to use the shell discontinuity stresses that would be produced by rigidly squeezing a shell locally to the amount of differential thermal expansion produced by the effective at values shown in Figure VII.2.-2.

1st APPROX - RIGID END RINGS

CYL A grows by an amount:

$$\left(K = \frac{W}{\Delta} = \frac{tE}{L} \right)$$

$$\delta_A = \alpha_A T_A L + W_A / K_A$$

VII.2.-1

Figure VII.2-5 Variation of Modulus of Elasticity with Temperature

THERMAL EXPANSION COEFFICIENT

U- 238 (CAST) FROM PRINCIPLES OF NUCLEAR REACTOR
ENGINEERING (GLASSTONE), P. 843

304 STAINLESS FROM SECTION III, NPVC

Figure VII.2-6 Variation of Thermal Coefficient of Expansion with Temperature

CALC THERMAL STRESS IN SHELL - 1st APPROX

$$\eta = \frac{(Rt)_B}{(Rt)_A} + 1 = \frac{10.06 (1/8)}{14.62 (1/2)} + 1 = 1.172$$

-40°F AMBIENT COND

Inner Shell $T_B = 218^\circ \text{ Max}$

Outer shell $T_A = 95^\circ \text{ Max}$

EFF., $\alpha E = 264$

$$\begin{aligned} \text{Inner shell } w_B &= \alpha E (T_B - T_A) \left(\frac{t_B}{\eta} \right) \\ &= \frac{(264) (123) (1/8)}{1.172} = 3460 \text{ Lbs/in} \end{aligned}$$

$$\text{Outer shell } w_A = 3460 (10.06/14.62) = 2370 \text{ Lbs/in}$$

$$\left. \begin{aligned} \text{Inner shell Merid } f_C &= 3460/.125 = 27,700 \text{ psi} \\ \text{Outer shell Merid } f_T &= 2370/.5 = 4740 \text{ psi} \end{aligned} \right\}$$

130° AMBIENT CONDITION

$T_B = 407^\circ \text{ Max}$, $T_A = 370^\circ \text{ Max}$, EFF., $\alpha E = 256$

$$(T_B - T_A) = 37^\circ$$

$$\text{Inner shell } w_B = \frac{256 (37) (1/8)}{1.172} = 1010 \text{ Lbs/in}$$

$$\text{Outer shell } w_A = 1010 (10.06/14.62) = 694 \text{ Lbs/in}$$

$$\left. \begin{aligned} \therefore \text{Inner Shell Merid } f_C &= 1010/.125 = 8070 \text{ psi} \\ \text{Outer Shell Merid } f_T &= 694/.5 = 1388 \text{ psi} \end{aligned} \right\}$$

INSTABILITY OF INNER SHELLLOCAL INSTABILITY (CRIPPLING)

Not critical. Yield strength limitation.

PRIMARY INSTABILITY (COLUMN BUCKLING)

The end rings and outer shell supply sufficient constraint that the inner cylinder may be considered to be fixed at both ends for a column calculation. Although we may calculate a buckling load, it is rather academic in any sense because the U-238 cylinders are only 1/16" away and they will immediately constrain the shell. However, by the numbers:

$$P_{CR} = \frac{4\pi^2 EI}{L^2} = \frac{4\pi^2 (27.6) (10^6) (399)}{(127)^2} = 27 (10^6)$$

$$\frac{P_{CR}}{A} = \frac{27(10^6)}{2\pi (10.06) (.125)} = 3.42 (10^6)$$

Not critical for elastic buckling. Again, yield strength limitation.

From the shell eq's in Roark, Chapt. 8, it can be shown that:

$$\text{Meridional bending } f_{MB} = \frac{1.816 E(AR)}{R}$$

$$\text{Hoop stress } f_H = E(AR)/R$$

$$\therefore f_{MB} = \frac{1.816 E \alpha(\Delta T) R}{R} = 1.816 E\alpha(\Delta T)$$

INNER SHELL THERMAL STRESS APPROX, -40° AMBIENT

$$\Delta T_i = 181^\circ - 65^\circ = 116^\circ$$

$$\therefore f_{MB} = 1.816 (27.7) (9.15) (116) = 53,400 \text{ psi (compr-outer)}$$

$$f_H = 53,400/1.816 = 29,400 \text{ psi (compr)}$$

$$f_{HB} = \mu f_{MB} = 16,000 \text{ psi (compr-outer)}$$

OUTER SHELL STRESS APPROX

$$\Delta T_o = 76^\circ - 54^\circ = 22^\circ$$

$$\therefore f_{MB} = 1.816 (28.2) (10.5) (22) = 11,800 \text{ psi (compr-outer)}$$

$$f_H = 11,800/1.816 = 6,500 \text{ psi (compr)}$$

$$f_{HB} = \mu f_{MB} = 3,540 \text{ psi (compr-outer)}$$

Figure VII.2-7 Temp Data for -40°F Ambient

LOCATION OF NULL POINT FOR THESE EFFECTS ON SHELLS

A "small" effect exists at $\lambda L = \pi$ }
 A "NIL" effect exists at $\lambda L = \pi (1.5)$ }

$$\lambda = \frac{1.284}{\sqrt{Rt}}$$

$$\text{"small" effect } L_o = \frac{\pi}{\lambda} = 2.45\sqrt{Rt}$$

$$\text{Outer shell } L_o = 2.45 \sqrt{(14.62) (1/2)} = 6.65"$$

$$\text{Inner shell } L_o = 2.45 \sqrt{10.06 (1/8)} = 2.74"$$

(Per Timoshenko). However, stress-coat tests and strain guage tests show the effects die out at a small % of these values.

SUMMARY OF THERMAL STRESSES IN SHELLS

The -40° ambient condition is critical.

Fig. VII.2-8 Thermal Stresses in LHM

$$\textcircled{1} \Sigma f_{M_O} = 4740 - 11800 = -7060$$

$$\Sigma f_{M_I} = 4740 + 11800 = 16540$$

$$\textcircled{2} \Sigma f_{H_O} = -6500 - 3540 = -10040$$

$$\Sigma f_{H_I} = -6500 + 3540 = -2960$$

$$\textcircled{3} \Sigma f_{M_O} = -27700 - 53400 = -81,100$$

$$\Sigma f_{M_I} = -27700 + 53400 = 25700$$

$$\textcircled{4} \Sigma f_{H_O} = -29400 - 16000 = -45400$$

$$\Sigma f_{H_I} = -29400 + 16000 = -13400$$

$\textcircled{5}$ ALL STRESSES TRIVIAL EXCEPT AXIAL STRESS

CHECK AXIAL CLEARANCE OF U-235 BLOCKS IN SHELLS

- inner/outer shell growth for -40° ambient condition will be most critical for comparison to U-238 -- see temp plot, Fig. VII.2.

$$\begin{aligned}
 \text{- shell net growth} &= \alpha_A T_A L + w_A / K_A \\
 &= \alpha_A T_A L + w_A \left(\frac{L}{t_A E_A} \right) \\
 &= 9.17 (10^{-6}) (95^\circ - 70^\circ) (127) + (23.70) \frac{127}{(1.5) (28.15) (10^6)} \\
 &= .0291 + .0214 \\
 &= .0515"
 \end{aligned}$$

$$\begin{aligned}
 \text{- U-238 net growth} &\approx (\alpha \Delta T)_{\text{EFF}} (L) \\
 &= 11.4 (10^{-6}) (180^\circ - 70^\circ) (127) = .1597"
 \end{aligned}$$

$$\text{- excess U-238 DEFL} = .1597 - .0515 = .1082"$$

DIFFERENTIAL THERMAL EXPANSION LOADS IN TIE-DOWN BOLT ASSY

Fig. VII.2-9 - Stiffness in Tie-Down Mechanism

$$\left. \begin{aligned} \delta_B &= \alpha_B T_B L + B / K_B \\ \delta_A &= \alpha_A T_A A + \alpha_B T_B (L - A) - B / K_A \end{aligned} \right\}$$

$$B \left(\frac{1}{K_B} + \frac{1}{K_A} \right) = \alpha_A T_A A - \alpha_B T_B A, \text{ Since } \delta_B = \delta_A$$

For similar α , E, $\Lambda = K_B/K_A$; $\Delta T = T_A - T_B$

$$B \left(\frac{1}{K_A} \right) \left(\frac{1}{\Lambda} + 1 \right) = \alpha A (\Delta T)$$

$$\therefore B = \frac{\alpha A (\Delta T) (K_A)}{\frac{1}{\Lambda} + 1}$$

$$\begin{aligned} \text{OR, } B &= \frac{\alpha A (\Delta T) (K_B)}{1 + \Lambda} \\ &= \frac{\alpha A (\Delta T)}{1 + \Lambda} \left(\frac{A_B E}{L} \right) \end{aligned} \quad (VII)$$

EVALUATING K_A APPROX, "MASH" IGNORED:

$$\begin{aligned} \Delta A1 &= \{ .7 \} \{ .0476 \bar{w} R^4 / EI \} \\ \text{W.R.T.} & \\ \text{Center} & \end{aligned}$$

$$= \{ .7 \} \{ .0952 BR^3 / EI \} = \{ .06664 / EI \} BR^3 \quad (VIII)$$

$$\begin{aligned} \Delta A2 &= \int_0^{\pi/2} \frac{M m ds}{EI}, \quad V_A \\ \text{W.R.T.} & \\ \text{Center} & \end{aligned}$$

$$V_A = .6366 \frac{M_A}{R}$$

$$m = \frac{R}{2} (.3183 - 1/2 \sin \theta)$$

$$= \frac{R}{4} (.6366 - \sin \theta) \quad (IX)$$

$$M = M_A (.6366 \cos \theta - 1/2)$$

$$= \frac{M_A}{2} (1.2732 \cos \theta - 1) \quad (X)$$

$$M_m = \frac{RM_A}{4} \{ .81052 \cos \theta - .6366 - 1.2732 \cos \theta \sin \theta + \sin \theta \}$$

$$ds = R d\theta \quad (XI)$$

$$\begin{aligned}
\int M_{md} dS &= \frac{R^2 M_A d\theta}{4} \int (\theta) (.81052) \cos \theta - .6366 (\theta) - (\theta) .6366 \sin 2\theta + \theta \sin \theta \\
&= \Gamma \left[.81052 \left\{ \cos \theta + \theta \sin \theta \right\} - .3183 \theta^2 - .6366 \left\{ \frac{1}{2} \left(\frac{\sin 2\theta}{2} - 2\theta \cos \theta \right) \right. \right. \\
&\quad \left. \left. + \sin \theta - \theta \cos \theta \right\} \right]_0^{\pi/2} \\
&= \Gamma \left[.81052 \left\{ \begin{matrix} 0 + \pi/2 \\ -1 - 0 \end{matrix} \right\} - .3183 \left(\frac{\pi^2}{4} \right) - \frac{.6366}{2} \left\{ 0 - \frac{\pi}{2} (-1) \right\} \right. \\
&\quad \left. + 1 \right] \\
&= \Gamma \left[.81052 \left(\frac{\pi}{2} - 1 \right) - \frac{.3183}{4} (\pi^2) - \frac{.6366}{2} \frac{\pi}{2} + 1 \right] \\
&= \Gamma \left[.464387 - .785372 - .499984 + 1 \right] = .17903\Gamma \quad (\text{VII.2.11})
\end{aligned}$$

+ sign indicates correct defl sign assumed

$$\therefore K_{A2} = -K_{A1}$$

I_A = EFF shell mom of inertia

A_B = Cross sectional area of bolt

$$\Gamma = \frac{R^2 M_A}{4} = \frac{R^2 B e}{4}$$

$$\Delta_{A2} = -.04476 B e R^2 / E I_A$$

$$\frac{\Delta_{A2}}{\Delta_{A1}} = \frac{-.04476 B e R^2}{.06664 B R^3} = -.6717 (e/R) \quad (\text{VII.2.12})$$

$$\Delta_{A1} + \Delta_{A2} = \Delta_{A1} \left\{ 1 - .6717 \left(\frac{e}{R} \right) \right\} \quad (\text{VII.2.13})$$

$$\Delta_A = \frac{.06664 B R^3}{E I_A} \left\{ 1 - .6717 \left(\frac{e}{R} \right) \right\} \quad (\text{VII.2.14})$$

$$\Delta_B = \frac{B L}{A_B E} \quad (\text{VII.2.15})$$

$$\Lambda = \frac{K_B}{K_A} = \frac{\Delta_A}{\Delta_B} = (.06664) \left(\frac{A_B}{I_A} \right) \left(\frac{R^3}{L} \right) \left\{ 1 - .6717 \left(\frac{e}{R} \right) \right\} \quad (\text{VII.2.16})$$

Now we can solve for B in Eq. (VII.2.6)

Figure VII.2-10 Temp Data for -130°F Ambient

Using Eqs. (VII.2.6) and (VII.2.16)

$$\text{EFF WIDTH} = 2R + \text{SADDLE WIDTH} = 42"$$

$$I_A = \frac{42 (1/2)^3}{12} = .438$$

$$A_B = \frac{\pi}{4} (2.5)^2 = 4.91$$

Assume tie bolt temp @ bottom = ambient = 130° (Fig. VII.2-9)

Assume tie bolt temp @ top = cyl ~200° = 205° Ave = 162°

$$\Delta T = 405^\circ - (162^\circ_{\text{AVE}}) = 243^\circ$$

$$E_B = 27.8 (10^6), \quad \alpha_A = 9.6 (10^{-6})$$

$$L = 18.5, \quad e = 4, \quad R = 13.25$$

$$A = 13.25 + 2.5 = 15.75$$

$$\Lambda = (.06664) \left(\frac{4.91}{.438} \right) \left\{ \frac{(13.25)^3}{18.25} \right\} \left\{ 1 - .6717 \left(\frac{4}{13.25} \right) \right\} = 75$$

This says that the shell is approx 75 times as flexible as the tie-bolt. We should thus have very little load B.

$$B = \frac{\alpha_A (\Delta T)}{1 + \Lambda} \left(\frac{A_B E}{L} \right) = \frac{9.6 (10^{-6}) 15.75 (243)}{1 + 75} \left\{ \frac{4.91 (27.8) (10^6)}{18.5} \right\}$$

$$= 3570\# \quad (\text{obviously no stress problem exists in the bolt})$$

$$2(3570)/1 \text{ gv saddle load} < 14\%$$

Likewise no stress problems exist in the shell, unless the shell $\delta_{\Delta T}$ is greater than the U-238 clearance - which might drive the U-238 and shells together

$$\text{Shell } \delta_{\Delta T} = \alpha_A (\Delta T) = 9.6 (10^{-6}) (15.75) (243^\circ) = .0366"$$

This is about half the nominal shell clearance, leaving .0259" to accommodate the U-238 D.W. "Mash" possibility. But comparison of beam tip defl of .094" (pg. 34) with a reasonable defl curve eyeballed through the actual saddle spacing (pg. 35), shows that maximum possible defl of the total beam would be only about 25% of this (.024") in the adjacent spans

and less than half of that would be attempting to generate "Mash" action i.e., less than .012". Thus, our .0259" left after accounting for the shell temp deflection constraint could not drive the U-238 and shells together.

Consideration of all possible combinations of all differential thermal expansions and flexibilities shows that no stress problems exist in the tie bolt assy and its effect on the shell.

VII.3 Projected LHM Radiation Levels Resulting from a Meltdown of P2 (Ref. 37)

The reference (38) analysis predicts LHM radiation levels stemming from a fuel meltdown wherein 50% of the fuel collects in the meltdown cup with the other 50% being uniformly distributed in the loop sodium. Data are presented on the bases of a 730-kW power level, 25-day irradiation time, and decay periods of 5, 10, 20, and 30 days. These data are generally consistent with the results of analyses and/or measurements reported under references (39) and (40) - see table VII.3-1.

There is, however, one anomaly common to all analyses. All analytical predictions are consistent with measured radiation levels obtained during the LHM gamma scan with the 135 curie source, except for predictions of axial contact at the shield valve. For the 135 curie source, calculations predicted 0.43 R/hr while measurements showed 14.0 R/hr, a variance factor of about 33. The reason for the variance is not obvious and it might be worthwhile to have ANC check into the matter.

If the variance is applied to other source cases for the same point, the radiation level at the underside of the gate valve can range from 300 to 960 R/hr. Should this occur, we may have to resort to the lead shot approach for additional shielding as described in Ref. (40).

Table VII.3-1 SLSF-LHM RADIATION DOSE RATES

Source		LHM Region		
		Below Valve Gate	Steel Flange Above Gate	4-1/8 U Sec Above Flange
1.	P-2 50% in cup 50% in place 730 kW-25 day irradiation 10 day decay	9.5 R/hr	63.0 R/hr	0.21 R/hr
2.	P-N 50% in cup 50% dispersed 1.5 MW-70 day irradiation 10 day decay	108.0 R/hr	756 R/hr-4.5"Fe 15.5 R/hr-10.5"Fe	0.24 R/hr
3.	P-N 50% in cup 50% dispersed 730 kW-25 day irradiation 10 day decay	32 R/hr	230 R/hr-4.5"Fe 4.7 R/hr-10.5"Fe	0.07 R/hr
	135 Curie Measured	14.0 R/hr	48 R/hr	4 to 8 Mr/hr
	Co-60 Calculated	0.437 R/hr	57 R/hr	4.4 Mr/hr

- NOTES: a. Case 1 data from reference (38).
b. Case 2 data from reference (39).
c. Case 3 data are normalized Case 2 information using J. Herceg correction factors.
d. Case 4 data from reference (40) with mid flange dose rates substituted for "steel flange above gate".

VIII. CRITICAL ATTACHMENTS

SLSF loop handling system is composed of several members and design attachments that can be considered critical since their failure could imply a severe accident of the entire system. Only selected and worst cases have been selected for the analysis in the following sections.

VIII.1 Conformance to Design Criteria (on Table of Contents)

All critical weld attachments for the SLSF loop handling machine have been covered in Chapter VI. The SLSF cradle assembly weld attachments analysis requires a knowledge of force and moment distribution in the entire structure. A three-dimensional structural analysis of the cradle frame is performed and reported separately in Volume II of this document.

The compliance of AISC code criteria on all cradle weld attachments is also reported in Volume II of this document. It is shown in Volume II of this document the all weld attachments meet the AISC code specifications.

However, the compliance to other aspects of the cradle structure to AISC code criteria is to be given in Volume III of this document.

VIII.2 Stress Analysis of Cylinder Mounting Pins

VIII.2.1 Cylinder Mounting Pins

The dimensions of a cylinder mounting pin and its attachments are shown on Fig. V.3.4.2-1. Refer to Drawing Numbers R1030-0040-DE-00 and R1030-0303-DE-02.

The sketch (Fig. VIII.2.1-1) shows the essential dimensions and the disposition of the clearances C_1 and C_2 when load F is applied.

The maximum load F is 100,000 lbs. when one cylinder has failed and all the load is carried on the other cylinder.

There are two lifting cylinder mounting pins similar to that shown on Fig. VIII.2.1-1, one at the upper end of the cylinder and one at the lower end. From the drawings the tolerances provide the following clearances between the pin and the bearing and between the pin and the blocks.

Block	2.255 <u>2.248</u>	2.253 <u>2.252</u>	Lower End Ave. = 0.004 in
	0.007 max.	0.001 min.	
Bearing	2.250 <u>2.248</u>	2.2495 <u>2.2520</u>	Lower End Ave. = 0
	0.002 max.	0.0025 interference	
Block	2.254 <u>2.247</u>	2.252 <u>2.249</u>	Upper End Ave. 0.005 in
	0.007 max.	0.003 min.	
Bearing	2.250 <u>2.247</u>	2.2495 <u>2.2490</u>	Upper End Ave. = 0.0017 in
	0.003 max.	0.0005 min.

Fig. VIII.2.1-1 Cradle Mounting Pin

Because of the tighter fit between the bearings and the pin than between the blocks and the pin, the following estimate of the way in which the pin is loaded is proposed. (Fig. VIII.2.1-2)

At 0-0,

$$M_{\max} = 50,000 (0.643 + 0.437 + 0.985 - 0.985/2) = 78,625 \text{ in lbs}$$

For pin,

$$\frac{I}{c} = \frac{\pi}{32} \times 2.5^3 = 1.118 \text{ in}^3 \quad A = \frac{\pi}{4} (2.25)^2 = 3.976 \text{ in}^2$$

$$\sigma_B = \frac{78,625}{1.118} = 70,326 \text{ psi}$$

The pin material is 4340 steel heat treated to R_c between 35 and 38, giving a yield strength of $S_y = 138,000 \text{ psi}$. Then factor of safety

$$F. S. = \frac{138,000}{70,326} = 1.96$$

In order to compare the deflection of the pin under the assumed loading with the clearances in the block, the deflection was calculated as shown below.

The sketch (Fig. VIII.2.1-3) represents the right half of the beam in Fig. VIII.2.1-2 with a tangent drawn at section 0. The distances from this tangent at sections 1, 2 and 3 will be obtained by the cantilever beam formulas for the three beams between 0 and 3. At sections 1 and 2 the shear forces are each 50K. At section 1 the bending moment is $50K (0.643 + 0.437) = 54K \text{ in. lbs}$. At section 2 the bending moment is $50K \times 0.643 = 32.15K \text{ in. lbs}$.

Fig. VIII.2.1-2 Assumed Load Pattern on Pin

For pin, $I = \frac{\pi}{64} 2.254 = 1.258 \text{ in.}^4$

$$L_1 = 0.985$$

$$L_1^2 = 0.970$$

$$L_1^3 = 0.956$$

$$L_2 = 0.437$$

$$L_2^2 = 0.191$$

$$L_2^3 = 0.0834$$

$$L_3 = 1.93$$

$$L_3^3 = 7.189$$

Fig. VIII.2.1-3

Deflected Right Half of Pin with Shears and Moments
at Sections 1 and 2

$$E\theta_1 = \frac{50K \times 0.97}{2 \times 1.2581} + \frac{54K \times 0.985}{1.2581} - \frac{50K \times 0.97}{6 \times 1.2581}$$

$$= 55.13 \text{ K}$$

$$EC_1 = \frac{50K \times 0.956}{3 \times 1.2581} + \frac{54K \times 0.97}{2 \times 1.2581} - \frac{50K \times 0.956}{8 \times 1.2581}$$

$$= 28.73 \text{ K}$$

$$E\theta_2 = \frac{50K \times 0.191}{2 \times 1.2581} + \frac{32.15K \times 0.437}{1.2581}$$

$$= 14.96 \text{ K}$$

$$EC_2 = \frac{50K \times 0.0834}{3 \times 1.2581} + \frac{32.15K \times 0.191}{2 \times 1.2581}$$

$$= 3.54 \text{ K}$$

$$EC_3 = \frac{50K \times 7.189}{60 \times 1.2581} = 4.76 \text{ K}$$

Distances for tangent at 0

$$\Delta_1 = \frac{EC_1}{E} = \frac{28.73 \times 10^3}{30 \times 10^6} = 0.00096 \text{ in.}$$

$$\begin{aligned} \Delta_2 &= \frac{I}{E} (EC_1 + L_2 E\theta_1 + EC_2) \\ &= \frac{(28.73 + 0.437 \times 55.13 + 3.54)10^3}{30 \times 10^6} \\ &= \frac{56.36 \times 10^3}{30 \times 10^6} = 0.0019 \text{ in.} \end{aligned}$$

$$\begin{aligned}
 \Delta_3 &= \frac{\Delta_2 + E (\theta_1 + \theta_2) L_3 + E C_3}{E} \\
 &= 0.00188 + \frac{(55.13K + 14.96K) 1.93 + 4.76K}{30 \times 10^6} \\
 &= 0.00188 + 0.00482 \\
 &= 0.0067 \text{ in.}
 \end{aligned}$$

Because the pin diameter is large compared with its length, the deflections due to shear may be significant compared with the bending deflection. For the beam of length L_1 the shear deflection at section 1 relative to the tangent at section 0 is

$$\frac{5}{9} \frac{WL}{AG} = \frac{5 \times 50,000 \times 0.985}{9 \times 3.976 \times 11.5 \times 10^6} = 0.0006 \text{ in.}$$

Adding this to the bending deflection Δ_1

$$\Delta_{\max} = 0.00096 + 0.0006 = 0.00156 \text{ in.}$$

This is less than the average clearance of 0.0017 in. at the bearing at the upper end.

For the beam of length L_3 the shear deflection at section 3 relative to the tangent at section 2 is

$$\frac{10}{54} \frac{WL}{AG} = \frac{10 \times 52,000 \times 1.93}{54 \times 3.976 \times 11.5 \times 10^6} = 0.00039 \text{ in.}$$

The corresponding bending deflection within the block will be

$$\Delta_3 - \Delta_2 = 0.0067 - 0.0019 = 0.0048 \text{ in.}$$

Adding this to the shear deflection

$$\Delta_{\max} = 0.0048 + 0.00039 = 0.0052 \text{ in.}$$

This compares favorably with the block clearance which is 0.004 in. at the lower end and 0.005 in. at the upper end.

The shearing stress in the pin is

$$\tau = \frac{4}{3} \frac{V}{A} = \frac{4}{3} \times \frac{50,000}{3.976} = 16,767 \text{ psi}$$

Assume

$$(S_S)_Y = \frac{2}{3} S_Y = \frac{2}{3} \times 138,000 = 92,000 \text{ psi}$$

Then factor of safety based on shear

$$F. S. = \frac{92,000}{16,767} = 5.48$$

VIII.3 Cradle Trunnion Pins

VIII.3.1 Stress Analysis of Cradle Trunnion Pins

A trunnion pin is shown in Fig. V.3.2-1.

The sketch (Fig. VIII.3.1-1) shows the basic dimensions of the pin and the clearances C_1 and C_2 as they would appear when the pin is loaded.

From drawings* the diametral dimensions are:

<u>ITEM</u>	<u>EITHER SIDE</u>
Cradle	5.011/5.008
Pin	5.002/4.998
Trunnion	5.008/5.005

*R1030-0393-DE-02

R1030-0042-DE-00

R1030-0380-DC-02 (see page III.17)

C_1

$$5.011 - 4.998 = 0.013 \text{ max.}$$

$$5.008 - 5.002 = .006 \text{ min.}$$

C_2

$$5.008 - 4.998 = .010 \text{ max.}$$

$$5.005 - 5.002 = .003 \text{ min.}$$

Fig. VIII.3.1-1

Trunnion Pin and Assumed Loading Pattern

The pin is modeled as a cantilever beam supported in the trunnion bearing and loaded by a uniformly varying load equal to one half of the total weight of the LHM-CRADLE, namely, 50,000 lbs. The position of the point D being indeterminate, it was assumed to be the mid-point of the 9 in. length of the pin in the trunnion bearing. From Fig. VIII.3.1-1

$$\sum M_{F3} = 50,000 \times 4.67 = F_2 \times 6.$$

From which $F_2 = 38917$ lb.

$$F_3 = F_1 + F_2 = 50,000 + 38917 = 88,917 \text{ lb.}$$

The maximum bending moment will occur between sections C and D. The maximum unit loading for the 88,917 lb load will be $q_0 = \frac{88,917 \times 2}{4.5}$
 $= 39419$ lbs./in at section C. Consequently, measuring X from point D,
 $M = \frac{q_0 X^3}{6L}$ which for $q_0 = 39519$ lbs./in and $L = 4.5$ inc. becomes $M = 1464 X^3$.

The 88,917 lb. load can be shown as a concentrated force.

(Fig. VIII.3.1-2)

The maximum bending moment of 191,169 occurs one half inch to the right of the vector representing the 88,917 lb. load. (F_3 in Fig. VIII.3.1-1)

For the cross section of the pin

$$I = \frac{\pi}{64} [5^4 - 1^4] = 30.63 \text{ in}^4$$

Loading on Section CD

		A	B	NET MOMENT
X	X ³	1464 X ³	38,917(3+X)	B-A
0.5	0.125	183	136,210	136,027
1.5	3.375	4941	175,126	170,185
2.5	15.625	22875	214,044	191,169 *
3.5	42.875	62769	252,961	190,192
4.5	91.125	133407	291,878	158,471

* M max

Fig. VIII.3.1-2

Bending Moments on CD

The bending stress is equal to

$$\sigma_B = \frac{MC}{I} = \frac{191,169 \times 2.5}{30.63} = 15,603 \text{ psi.}$$

The maximum yield strength of the pin material is $S_Y = 90,000$ psi.

The factor of safety based on bending stress is

$$F. S. = \frac{90,000}{15,603} = 5.77$$

The maximum shearing force is 50,000 lb and the maximum direct shear stress is:

$$\tau = \frac{4}{3} \frac{V}{A}$$

$$A = \frac{\pi}{4} \left[5^2 - 1^2 \right] = 18.85 \text{ in}^2$$

$$\tau = \frac{4}{3} \times \frac{50,000}{18.85} = 3537 \text{ psi}$$

The yield strength of the pin material in shear is approximately

$$(S_S)_Y = \frac{2}{3} \times 90,000 = 60,000 \text{ psi}$$

The factor of safety based on shear stress is

$$F. S. = \frac{60,000}{3537} = 16.96$$

The bending deflection of end A relative to the tangent at Section;B is

$$\Delta_A = \frac{WL^3}{60EI} = \frac{50,000 \times 8^3}{60 \times 30 \times 10^6 \times 30.63}$$

$$= 0.00046 \text{ in.}$$

The shear deflection is

$$\frac{10}{54} \frac{WL}{AG} = \frac{10 \times 50,000 \times 8}{54 \times 18.85 \times 11.5 \times 10^6} = 0.00034 \text{ in.}$$

$$\Delta_{\max} = 0.00046 + 0.00034 = 0.00080 \text{ in.}$$

This is less than the minimum clearance of 0.004 in. in the cradle bearing.

VIII.4 T-Bar Stress Analysis

The weakest links in the "Harpoon Beam" are: The "T" bar web and the bolts fastening the "T" bar to the frame. The bearing stresses between the "T" bar and the channel were not checked at this time because of lack of dimensions of the channel.

Stresses in "Harpoon" Due to 10° Tilt from Vertical Position

a. Center of Gravity: The loads for C:G calculations were taken from Ref. 23, Section 2.1 and are shown in Fig. VIII.4-1. The dimensions are in accordance to the 3-D frame analysis.

$$W = \Sigma P = 101600 \text{ lbs}$$

$$X = \frac{\Sigma P_i x_i}{W} = \frac{15148722}{101600} = 149. \text{ in}$$

b. Reaction loads due to 10° tilt of the LHM from vertical position. In this case the cylinder is held in place only by the "Harpoon" beam and the base plate camlocks.

Section of "Harpoon Beam"

Fig. VIII.4-1

Loading on Cradle

$$\Sigma M_A = R_1 b - W a \sin \alpha = 0$$

$$R_1 = \frac{W a \sin \alpha}{b} = \frac{101,600 \times 149 \times 0.174}{292.5} = 9005 \text{ lb.}$$

$$R_2 = W \cos \alpha = 101,600 \times 0.984 = 100,056 \text{ lb.}$$

$$P = W \sin \alpha - R_1$$

$$= 101,600 \times 0.174 - 9005$$

$$= 8673 \text{ lb.}$$

c. Stresses in the "T" bar

Since the 10° tilt is a part of lifting requirements, the 3g requirement, according to reference 23 must be satisfied. The design load (R_1) in the Harpoon then is

$$R_1 = 9005 (3) = 27015$$

Minimum cross section area of the "T" bar is

$$A = tL = 1.25 (11.0) = 13.75 \text{ in}^2$$

and the stresses due to $R_1 = 27015$ lbs.

$$\sigma_T = \frac{27015}{13.75} = 1965 \text{ psi}$$

d. Stresses in the channel

This part is rigidly attached to the cask and the 3g requirement must be satisfied.

Load from "T" bar on each side of the channel:

$$P/2 = \frac{1}{2} (27015) = 13508 \text{ lbs}$$

$$A = 1.2 \times 13 = 15.6 \text{ in}^2$$

$$I = \frac{13(1.20)^3}{12} = 1.87 \text{ in}^2$$

$$\sigma_c = \frac{13508}{15.6} \pm \frac{(13508 \times 1.2) (.60)}{1.87} = 866 \pm 5201$$

$$\sigma = 6067 \text{ psi}$$

Stresses in "Harpoon" due to Transverse Acceleration (x-direction)

During Transport

From 3-D frame analysis (L,z + 1 cond.) axial force in harpoon (Beam No. 3) due to 1g acceleration:

$$F_3 = 549. \text{ lbs.}$$

The specified acceleration for vertical (transverse acceleration) bounce of the LHM is 2g for all parts (Ref. 23) so that the design load is:

$$F = 549 \times 2 = 1098 \text{ lbs}$$

The stresses in the harpoon for this loading case were obtained by rationing the axial forces:

- a. Highest stresses in the "T" bar

$$\sigma_T = \frac{1098}{27015} (1965) = .041 (1965) = 81 \text{ psi}$$

- b. Stresses in bolts

$$\sigma_B = .041 (52000) = 2132 \text{ psi}$$

- c. Stresses in the channel

$$\sigma_C = .041 (10200) = 418 \text{ psi}$$

Table VIII.4-1

Summary of "Harpoon" Analysis

Loading Cond.	Stresses in "T" Bar (psi)	Bolt Stresses (psi)	Stresses in Channel (psi)
10° tilt from vert. (3g)	1965	52000	6067
Vertical Accel. (2g)	81	2132	418
Lateral Accel. (5g)	2653	70200	17600
Axial Accel. (10g)	0	0	0
Vertical Accel. (-2g)	0	0	0
Vertical Accel. (1-2g) cyl. = 0	0	0	0

Stresses in "Harpoon" due to Lateral Acceleration (Y-direction)During the Transport

From 3-D frame analysis ($L_1Y + 1$ condition) axial force in the harpoon beam due to 1g lateral acceleration:

$$F_3 = 18260. \text{ lbs}$$

The specified lateral acceleration (Ref. 23) is, 2g not rigidly attached to the cask, and 5g for rigid attachments

$$F = 18260 (5) = 91300 \text{ lbs (channel)}$$

$$F = 18260 (2) = 36520 \text{ lbs (T-Bar, bolts)}$$

a. Stresses in "T" bar

$$\sigma_T = \frac{36520}{27015} (1965) = 1.35 (1965) = 2653. \text{ psi}$$

b. Stresses in bolts

$$\sigma_B = 1.35 (52000) = 70200. \text{ psi}$$

c. Stresses in the channel

$$\sigma_C = 1.73 (10200) = 17600. \text{ psi} \quad \frac{91300}{27015} = 3.38$$

Conclusions and Summary of the "Harpoon" Beam Analysis

Since the harpoon beam is capable of carrying only axial tension, there were no stresses in this beam due to axial acceleration or due to downward vertical acceleration as shown in Table VIII.4-1. The highest stresses of 70,200 psi occurred in the bolts during the 5g lateral acceleration.

The performed stress analysis indicated rather low stress levels in the harpoon beam (AS14, T1 material) for all loading conditions. It can be concluded, therefore, no clearance between the "T" bar and the "Harpoon" channel is necessary in order to avoid loading of the "T" bar during travel.

VIII.5 Stresses in Cradle Base Plate

VIII.5.1 Stresses in Toe Weld of Cradle Base Plate

The geometry of the cradle base plate is shown on Fig. VIII.5.1-1. Total load acting on the plate is 100,000#. Assumption is made that the load is concentrated at the center of 22" I.D. beam opening. Part of the beam which is not supported is ignored in this analysis (conservative). Fig. VIII.5.1-2 shows the weld size and weld area of cross section. The analysis performed indicates that the stresses in all cross-sections and welds are less than AISC Code allowable stresses.

NOTE: ASSUMPTION IS MADE THAT LOAD IS CONCENTRATED AT CENTER OF 22" I.D. BEAM OPENING (AS SHOWN IN SKETCH). PART OF BEAM WHICH IS NOT SUPPORTED IS IGNORED IN ANALYSIS (CONSERVATIVE).

Fig. VIII.5.1-1 Cradle Base Plate

**TOTAL WELD AREA = 12×0.66
+ $1.54 + 1.23 + 1.85 = 12.54$ IN²**

WELD SIZE	WELD AREA
5/16 - 3	$0.312 \times 0.707 \times 3 = 0.66 \text{ IN}^2$
5/16 - 7	$0.312 \times 0.707 \times 7 = 1.54 \text{ IN}^2$
1/4 - 7	$0.25 \times 0.707 \times 7 = 1.23 \text{ IN}^2$
3/8 - 7	$0.375 \times 0.707 \times 7 = 1.85 \text{ IN}^2$

Fig. VIII.5.1-2 Weld Pattern on Toe

$$\bar{x} = \frac{2 \times 0.66 [2 + 9 + 16 + 23 + 30 + 37]}{12.54} \\ + \frac{1.23 \times 23.25 + 1.85 \times 39}{12.54} = 20.4''$$

$$I_{33} = 12 \times \frac{1}{12} \times 0.3125 \times 0.707 \times 27 \\ + 2 \times 0.66 [18.4^2 + 11.4^2 + 4.4^2 + 2.6^2 + 9.6^2 + 16.6^2] \\ + 1.54 \times 20.4^2 + 1.23 \times 2.6^2 + 1.85 \times 18.85^2$$

$$I_{33} = \underline{2451} \text{ in}^4$$

$$M_{33} = 7.6 \times 50,000 = 380,000 \text{ in lbs}$$

$$\text{At CD, } \sigma = \frac{P}{A} + \frac{MC}{I} = \frac{50,000}{12.54} + \frac{380,000 \times 18.85}{2451} = 3987 + 2920 = 6907 \text{ psi}$$

$$\text{At BE, } \sigma = 3987 + \frac{380,000 \times 2.6}{2451} = 4390 \text{ psi}$$

$$\text{At AF, } \sigma = 3987 - \frac{380,000 \times 20.4}{2451} = 827 \text{ psi}$$

These three stresses are tensile.

On Vertical Welds - Moment of horizontal welds (see Fig. VII.5.1-3)

$$\begin{aligned}
 M &= \left[6655 \times 14.1 + 5552 \times 7 + \frac{4550 \times 0.75}{2} \right] 3 \times 0.3125 \times 0.707 \times 2 \\
 &+ 6907 \times 15.75 \times 0.375 \times 0.707 \times 7 \\
 &= 380,064 \text{ in lbs}
 \end{aligned}$$

For Vertical Welds (Fig. VIII.5.1-4)

$$\begin{aligned}
 \bar{y} &= \frac{0.3125 \times 0.707 \times 7.75 \times 13.5 + 2 \times 0.3125 \times 0.707 \times 4 \times 11.5}{0.3125 \times 0.707 \times 23.75 + 1.5 \times 7.75} \\
 &+ \frac{0.3125 \times 0.707 \times 7.75 \times 1.5 + 2 \times 0.3125 \times 0.707 \times 4 \times 3.5 + 1.5 \times 7.75 \times 1.5}{6.96 + 11.63} \\
 &= \frac{60.93}{18.59} = 3.28''
 \end{aligned}$$

$$\begin{aligned}
 I &= \frac{4 \times 0.3125 \times 0.707 \times 4^3}{12} + \frac{7.75 \times 1.5^3}{12} \\
 &+ 0.3125 \times 0.707 \times 7.75 \times 10.22^2 \\
 &+ 2 \times 0.3125 \times 0.707 \times 4 \times 8.22^2 \\
 &+ 7.75 \times 1.5 \times 2.53^2 = 379.5 \text{ in}^4
 \end{aligned}$$

$$\sigma_{zz} = \frac{380,064 \times 10.22}{379.5} = 10,235 \text{ psi}$$

$$y = 827 + \frac{6907 - 827}{39.25} x = 827 + 154.9x$$

Fig. VIII.5.1-3 Stress at Several Locations

Fig. VIII.5.1-4 Vertical Welds

$$\begin{aligned} \text{TOTAL SHEAR FORCE} &= [6655 + 5552] 3 \times 0.3125 \times 0.707 \times 2 + 4550 \\ &\times 1.5 \times 0.3125 \times 2 = 20448 \# \end{aligned}$$

$$\text{AVERAGE SHEAR STRESS} = \frac{20448}{18.59} = \underline{1110} \text{ psi}$$

An additional effect on toe weld would be that due to transverse fixity moment.

Fig. VIII.5.1-5

Transverse Fixity Moment

$$\theta = \frac{PL^2}{16EI} - \frac{ML}{2EI}$$

Let K = Stiffness of support (in.lbs/radian)

$$\text{Then } \theta = \frac{M}{K}, \quad \frac{M}{K} = \frac{PL^2}{16EI} - \frac{ML}{2EI}$$

$$M = \frac{\frac{PL^2}{16EI}}{\frac{1}{K} + \frac{L}{2EI}}$$

CHECK For $K = \infty$

$$M = \frac{PL}{8} \quad (\text{OK})$$

Fig. VIII.5.1-6 Assume 3 Panels Condition

Assume 3 panels - instead of 2 and toe piece .

$$I_{22} = 12 \times 0.375 \times 3.25^2 \times 6 + \frac{1}{12} \times \frac{1}{2} \times 6.5^3 \times 4$$

$$= 330 \text{ in}^4$$

$$K = \frac{EI}{L} = \frac{30 \times 10^6 \times 330}{52} = 0.1904 \times 10^9$$

$$\frac{1}{K} = 5.25 \times 10^{-9}$$

$$\frac{PL^2}{16EI} = \frac{100,000 \times 62.125^2}{16 \times 30 \times 10^6 \times 307.61} = 0.002614$$

For Plate

$$= 2.614 \times 10^{-3}$$

$$\frac{L}{2EI} = \frac{62.125}{2 \times 30 \times 10^6 + 307.61} = 3.366 \times 10^{-9}$$

$$M_F = M_{22} = \frac{2.614 \times 10^{-3}}{5.25 \times 10^{-9} + 3.366 \times 10^{-9}} = \underline{\underline{303,390 \text{ in.lbs}}}$$

5/16" Welds Considered as Lines

$$I_{xx} = 7 \times 6^2 \times 2 + \left(\frac{1}{12} \times 4^3 + 4 \times 4^2\right) 4$$

$$= 504 + 277 = 781 \text{ in}^3$$

$$I_{yy} = 8 \times 3.5^2 \times 2 + \frac{1}{12} \cdot 7^3 \times 2$$

$$= 196 + 57 = 253 \text{ in}^3$$

$$I_{pL} = 781 + 253 = 1034 \text{ in}^3$$

$$t = 0.3125 \times 0.707 = 0.221 \text{ in}$$

$$I_p = 1034 \times 0.211 = 228 \text{ in}^4$$

$$r = \sqrt{3.5^2 + 6^2} = 6.94"$$

Fig. VIII.5.1-7

Geometry of Vertical Welds

Fig. VIII.5.1-8
Shear Stress Component

Applying total fixity moment to these welds (actually they will carry only part of it)

$$\tau = \frac{303,390 \times 6.94}{228} = 9235 \text{ psi}$$

$$\theta = \tan^{-1} \frac{3.5}{6} = 30.25^\circ$$

$$\tau_x = 9235 \cos \theta = 7977 \text{ psi}$$

$$\tau_y = 9235 \sin \theta = 4653 \text{ psi}$$

$$\tau_y + \tau_{ave} = 4653 + 1110 = 5763$$

$$\tau_R = \sqrt{5763^2 + 7977^2} = 9841 \text{ psi}$$

combined with $\sigma_t = 10,235 \text{ psi}$ at upper weld

$$\tau_{max} = \sqrt{\left(\frac{10,235}{2}\right)^2 + 9841^2} = \underline{\underline{11,092}} \text{ psi} < 20000 \text{ psi}$$

Fig. VIII.5.1-9 Moment Distribution on Plate

This increased moment effect at the center is associated in the following analysis.

From Fig. VIII.5.1-10

$$\bar{x} = \frac{(2.75 + 10.75 + 22.75 + 31.75 + 40.75 + 46.75) 2A}{12A} = 25.9"$$

$$I = 1.202 (23.15^2 + 15.15^2 + 3.15^2 + 5.85^2 + 14.85^2 + 20.85^2) + \frac{\pi}{64} \left(\frac{7}{8}\right)^4 \times 12$$

$$= \underline{1761} \text{ in}^4$$

$$\text{Total Body Area of Bolts} = 12 \times 0.601 = 7.21 \text{ in}^2$$

$$\text{Stress Area of Bolts} = 0.462 \text{ in}^2$$

$$\text{Load at Load Line} = 50,000 \text{ \#}$$

$$\text{Moment} = 7.1 \times 50,000 = 355,000 \text{ in. lbs}$$

$$\text{Body Stress at F - F Bolts} = \frac{F}{A} + \frac{Mc}{I}$$

$$= \frac{50,000}{7.21} + \frac{355,000 \times 20.85}{1761}$$

$$= 11,138 \text{ psi}$$

$$I_{2-2} = 6 \times 0.601 \times 2.0625^2 \times 2 = 30.68 \text{ in}^4$$

$$\sigma = \frac{303390 \times 2.0625}{30.68} = 20396 \text{ psi}$$

$$\text{Total stress at F bolts} = 20396 + 11138$$

$$= 31534 \text{ psi}$$

The root total stress

$$= \frac{31534 \times 0.601}{0.462} = 41021. \text{ psi}$$

$$\text{Yield stress of the material} = 172000 \text{ psi}$$

Factor of safety based on yield

$$= \frac{172000}{41021} = 4.19$$

If it had been assumed that rotation had taken place about the edge AA instead of the neutral axis previously found, then the moment of the bolt loads about AA would be

$$M = F_1 \times 2.75 + F_2 \times 10.75 + F_3 \times 22.75 + F_4 \times 31.75 \\ + F_5 \times 40.75 + F_6 \times 46.75$$

These loads would be proportional to their distances from AA. That is $F_1/F_6 = 2.75/46.75$, $F_2/F_6 = 10.75/46.75$ and so on. Then

$$M = F_6 \left[\frac{2.75^2 + 10.75^2 + 22.75^2 + 31.75^2 + 40.75^2}{46.75} + 46.75 \right] \\ = 117.54 F_6$$

Moment of applied load about AA = $50,000 \times 33 = 1,650,000$ in lb

$$F_6 = \frac{1,650,000}{117.54} = 14038 \text{ lb}$$

$$\text{Load per } F_6 \text{ bolt} = \frac{14038}{2} = 7019 \text{ lb.}$$

$$\text{Bolt stress} = \frac{7019}{0.462} = 15,193 \text{ psi}$$

Assume rotation about side AF due to side fixity moment. Then calling the higher stressed row of bolts F_7 and the lower stressed bolts F_8

$$M = 6 F_7 \times 5.5625 + 6 F_8 \times 1.4375$$

$$F_8/F_7 = 1.4375/5.5625$$

$$M = 6 F_7 \times 5.5625 + 6 F_7 \frac{1.4375^2}{5.5625} \\ = 6 F_7 (5.5625 + 0.371) = 35.6 F_7$$

$$35.6 F_7 = 303,390 \quad F_7 = 8522 \text{ lb.}$$

$$\text{Bolt stress} = \frac{8522}{0.462} = 18,446 \text{ psi}$$

$$\begin{aligned} \text{Total bolt stress} &= 15,193 + 18,446 \\ &= 33,639 \text{ psi} \end{aligned}$$

This is lower than the stress of 41,021 psi found by the other method. Both analyses assume that all of the applied load is carried by the bolt; in other words that the bolts are not tightened. This is not true but the exact magnitude of the tightening load is unknown. From the second analysis the total load applied to the higher stressed F_6 bolt is $7019 + 8522 = 15,541$ lb. Assume that a tightening load equal to one half of the yield load, namely $86000 \times 0.462 = 39,732$ lb. is applied to each bolt and that the bolts compress an area equal to three times the bolt diameter d at each hole. The area of these rings would be $\pi/4[(3d)^2 - d^2] = 2\pi d^2$ and the ratio of the stiffness of the joint to that of the bolt would be $2\pi d^2 \div \pi/4d^2 = 8$. Then the total load on the bolt would be

$$F_B = 39,732 + \frac{15,541}{1+8} = 41,458 \text{ lb.}$$

$$\text{Bolt stress } \sigma = \frac{41,458}{0.462} = 89,738 \text{ psi}$$

Note that the applied load increases the bolt load only 4%. If the tightening load had been equal to the yield or proof load, and this is a common specification, the increase due to the applied load would have been less. A rough "rule of thumb" used by some designers is to design bolts for the tightening load or the applied load, whichever is the larger.

VIII.5.2 Finite Element Analysis of Cradle Base Plate

Due to the symmetric geometry and loading case, only half of the cradle base plate is analyzed by SAP IV program. The finite element model of top cradle base plate and bottom cradle base plate are shown on Fig. VIII.5.2-1 and Fig. VIII.5.2-2 respectively. The two edge bolts are modeled as a boundary 3-D beam located at the center of these two bolts. The ribs are modeled as 3-D vertical beam (column) connecting two plates or horizontal beam supporting each plate. Concentrated loads are converted from distributed loads applied at node 41, 55, 67, 77, 87, 97, and 115.

VIII.5.2.1 Boundary Beam (Bolt)

The boundary beam (see Fig. VIII.5.2.1-1 and-2 converted from two edge bolts ($7/8 \phi$ bolt, $l = 3"$, $A = 0.462 \text{ in}^2$, $d = 0.767"$) is assumed to be fixed at top end with length 1.375" and area 0.924 in^2 .

Fig. VIII.5.2.1-1
Boundary Beam Model

$$A = 2 \times 0.462$$

$$= 0.924 \text{ in}^2$$

$$I_x = 2 \times \frac{\pi}{64} \times 0.767^4$$

$$= 0.034 \text{ in}^2$$

$$I_y = 2 \times \left[\frac{\pi}{64} \times 0.767^4 + 0.462 \times 2^2 \right]$$

$$= 3.73 \text{ in}^4$$

Fig. VIII.5.2-1 Finite Element Model of Top Plate

Fig. VIII.5.2-2 Finite Element Model of Bottom Plate

Fig. VIII.5.2.1-2 Two Edge Bolts

Fig. VIII.5.2.1-3

Section 1

$$A_1 = 1 \text{ in}^2 \text{ (cross section area)}$$

$$A_2 = 1 \text{ in}^2 \text{ (area against shear force in 2-directions)}$$

$$A_3 = 1 \text{ in}^2 \text{ (area against shear force in 3-directions)}$$

$$I_2 = \frac{1}{12} \times 2 \times \left(\frac{1}{2}\right)^3 = 0.0208 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 2^3 = 0.333 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 0.354 \text{ in}^4$$

Section 2 (column: 2, 3, 26, 27)

$$A_1 = 2 \text{ in}^2$$

$$A_2 = 2 \text{ in}^2$$

$$A_3 = 2 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 4 \times \left(\frac{1}{2}\right)^3 = 0.0417 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 4^3 = 2.666 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 2.708 \text{ in}^4$$

Fig. VIII.5.2.1-4

Section 2

Section 3 (column: 9, 19)

Fig. VIII.5.2.1-5

Section 3

$$A_1 = 1\frac{1}{2} \times \frac{3}{4} = 1.125 \text{ in}^2$$

$$A_2 = 1.125 \text{ in}^2$$

$$A_3 = 1.125 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times \frac{3}{2} \times \left(\frac{3}{4}\right)^3 = 0.0528 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{3}{4} \times \left(\frac{3}{2}\right)^3 = 0.211 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 0.264 \text{ in}^4$$

Section 4 (column: 10, 14, 18)

Fig. VIII.5.2.1-6

Section 4

$$A_1 = \frac{3}{4} \times 3 = 2.25 \text{ in}^2$$

$$A_2 = 2.25 \text{ in}^2$$

$$A_3 = 2.25 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 3 \times \left(\frac{3}{4}\right)^3 = 0.105 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{3}{4} \times 3^3 = 1.668 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 1.793 \text{ in}^4$$

Section 5 (beam: 28, 29, 30, 31, 38, 39, 40, 41, 49, 50, 54, 55, 62, 63, 64, 65, 72, 73, 74, 75)

Fig. VIII.5.2.1-7

Section 5

$$A_1 = 1 \text{ in}^2$$

$$A_2 = 1 \text{ in}^2$$

$$A_3 = 1 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 2 \times \left(\frac{1}{2}\right)^3 = 0.0208 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 2^3 + 1 \times \left(1 + \frac{1.25}{2}\right)^2 = 2.974 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 2.994 \text{ in}^4$$

Section 6 (beams: 76, 77, 78, 79, 80, 81, 84, 85, 87, 88, 90, 91, 92, 93, 94, 95)

Fig. VIII.5.2.1-8

Section 6

$$A_1 = 1 \text{ in}^2$$

$$A_2 = 1 \text{ in}^2$$

$$A_3 = 1 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 2 \times \left(\frac{1}{2}\right)^3 = 0.0208 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 2^3 + 1 \times \left(1 + \frac{0.625}{2}\right)^2 = 2.055 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 2.076 \text{ in}^4$$

Section 7 (beam: 45, 46, 53, 58)

Fig. VIII.5.2.1-9

Section 7

$$A_1 = 2 \times \frac{3}{4} = 1.5 \text{ in}^2$$

$$A_2 = 1.5 \text{ in}^2$$

$$A_3 = 1.5 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 2 \times \left(\frac{3}{4}\right)^3 = 0.0703 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{3}{4} \times 2^3 + 1.5 \left(1 + \frac{1.25}{2}\right)^2 = 4.461 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 4.531 \text{ in}^4$$

Section 8 (beam: 82, 83, 86, 89)

Fig. VIII.5.2.1-10

Section 8

$$A_1 = 2 \times \frac{3}{4} = 1.5 \text{ in}^2$$

$$A_2 = 1.5 \text{ in}^2$$

$$A_3 = 1.5 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 2 \times \left(\frac{3}{4}\right)^3 = 0.0703 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{3}{4} \times 2^3 + 1.5 \times \left(1 + \frac{0.625}{2}\right)^2 = 3.084 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 3.084 \text{ in}^4$$

Section 9 (beam: 33, 34, 36, 37, 43, 44, 47, 48, 51, 52, 56, 57, 60, 61, 67, 68, 70, 71)

$$A_1 = 4 \times \frac{1}{2} = 2 \text{ in}^2$$

$$A_2 = 2 \text{ in}^2$$

$$A_3 = 2 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 4 \times \left(\frac{1}{2}\right)^3 = 0.0417 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 4^3 + 2 \times \left(2 + \frac{1.25}{2}\right)^2 = 16.44 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 16.490 \text{ in}^4$$

Fig. VIII.5.2.1-11

Section 9

Section 10 (beam: 32, 35, 42, 59, 66, 69)

$$A_1 = 4 \times \frac{1}{4} = 1 \text{ in}^2$$

$$A_2 = 1 \text{ in}^2$$

$$A_3 = 1 \text{ in}^2$$

$$I'_2 = \frac{1}{12} \times 2 \times \left(\frac{1}{3}\right)^3 = 0.0208 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 2^3 + 1 \times \left(1 + \frac{0.625}{2}\right)^2 = 2.055 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 2.076 \text{ in}^4$$

Fig. VIII.5.2.1-12

Section 10

Section 11 (column: 1, 4, 5, 6, 7, 8, 11, 12, 13, 15, 16, 17, 20, 21, 22, 23, 24, 25)

(For Model A)

$$A_1 = 1.5 \text{ in}^2$$

$$A_2 = 1.5 \text{ in}^2$$

$$A_3 = 1.5 \text{ in}^2$$

$$I_2 = \frac{1}{12} \times 3 \times \left(\frac{1}{2}\right)^3$$

$$= 0.03125 \text{ in}^4$$

$$I_3 = \frac{1}{12} \times \frac{1}{2} \times 3^3$$

$$= 1.125 \text{ in}^4$$

$$I_1 = I_2 + I_3 = 1.156 \text{ in}^4$$

Fig. VIII.5.2.1-13 Section 11

Section 12 (boundary beam: 96, 97, 98, 99, 100, 101)

$$A_1 = 0.924 \text{ in}^2$$

$$A_2 = 0.924 \text{ in}^2$$

$$A_3 = 0.924 \text{ in}^2$$

$$I_1 = 3.764 \text{ in}^4$$

$$I_2 = 0.034 \text{ in}^4$$

$$I_3 = 3.73 \text{ in}^4$$

Fig. VIII.5.2.1-14 Section 12

SUMMARY

Model (A)

(1) Axial Force in Boundary Beam (see Fig. VIII.5.2.1-15)

The location and magnitude of external concentrated load and axial force in boundary element obtained from SAP IV program are shown in Fig. VIII.5.2.1-15. It is found that the max. axial force is 18.1^K . The sum of those axial forces is 50.00^K which is exactly checked with the total external load. Note that each boundary element consists of two bolts. So the axial force in each bolt is half value of the axial force in boundary element.

(2) Shear Force at End of Vertical Beam (see Fig. VIII.5.2.1-16)

The shear forces at end of vertical beams (columns) are shown in Fig. VIII.5.2.1-16. From these shear forces, the weld stresses in the top plate and in the bottom plate were calculated. Figure VIII.5.2.1-17 shows the weld stress in top plate. Figure VIII.5.2.1-18 shows the weld stress in bottom plate.

Top Plate Stress

3" Column Case

$$\tau = \frac{VQ/I}{2 \times \frac{1}{4} \times 0.707} = \frac{V \times 0.75 \times 0.75 / 1.125}{2 \times 0.25 \times 0.707}$$

$$= 1.41 V$$

Fig. VIII.5.2.1-19
3" Weld Case

Fig. VIII.5.2.1-15 Axial Force in Boundary Beam (kips)

Fig. VIII.5.2.1-16 Shear Force at End of Vertical Beam (kips)

Fig. VIII.5.2.1-17 Weld Stress on Top Plate (ksi)

Fig. VIII.5.2.1-18 Weld Stress on Bottom Plate (ksi)

4" Column Case

$$\tau = \frac{VQ/I}{2 \times \frac{1}{4} \times 0.707} = \frac{V \times 1 \times 1/2.666}{2 \times 0.25 \times 0.707}$$

$$= 1.06 V$$

Fig. VIII.5.2.1-20
4" Weld Case

Where

τ = weld shear stress

V = shear force in column end

Bottom Plate Weld Stress3" Column Case

$$\tau = \frac{V}{3 \times \frac{1}{4}} = 1.3 V$$

4" Column Case

$$\tau = \frac{V}{4 \times \frac{1}{4}} = 1.0 V$$

(3) Stresses in Plate Element

Stress in top face of top plate (see Fig. VIII.5.2.1-21)

Stress in bottom face of bottom plate (see Fig. VIII.5.2.1-22)

(4) Deflection (see Fig. VIII.5.2.1-23)

MODEL A

Fig. VIII.5.2.1-21 Stress in Top Face of Top Plate (ksi)

MODEL A

Fig. VIII.5.2.1-22 Stress in Bottom Face of Bottom Plate (ksi)

Fig. VIII.5.2-1-23 Deflection of Model A (in $\times 10^{-2}$)

Model (B)

- (1) Axial Force in Boundary Beam (see Fig. VIII.5.2.1-24)
- (2) Shear Force at End of Vertical Beam (see Fig. VIII.5.2.1-25)

Weld stresses of top plate (see Fig. VIII.5.2.1-26)

Weld stresses of bottom plate (see Fig. VIII.5.2.1-27)

Top Plate Weld Stress2" Column Case

$$\tau = \frac{VQ/I}{2 \times \frac{1}{4} \times 0.707} = \frac{V \times 0.5 \times 0.5/0.333}{2 \times 0.25 \times 0.707}$$

$$= 2.1 V$$

Fig. VIII.5.2.1-28

2" Weld Case

(3) Stresses in Plate Element

Stresses in top face of top plate
(Fig. VIII.5.2.1-29)

Stresses in bottom face of bottom plate
(Fig. VIII.5.2.1-30)

4" Column Case

Same as Model (A)

Bottom Plate Weld Stress(4) Deflection (Fig. VIII.5.2.1-31)2" Column Case

$$\tau = \frac{V}{4 \times \frac{1}{4}} = 2.0 V$$

4" Column Case

Same as Model (A)

Fig. VIII.5.2.1-24 Axial Force in Boundary Beam (kips)

Fig. VIII.5.2.1-25 Shear Force at End of Vertical Beam (kips)

Fig. VIII.5.2.1-27 Weld Stress of Bottom Plate (ksi)

MODEL B

Fig. VIII.5.2.1-29 Stresses in Top Face of Top Plate (ksi)

MODEL B

Fig. VIII.5.2.1-30 Stresses in Bottom Face of Bottom Plate (ksi)

Fig.VIII.5.2.1-3a Deflection of Model B (in $\times 10^{-2}$)

VIII.5.2.2 Vertical Beam (column) and Horizontal Beam Element

Figures VIII.5.2.2-1 and -2 show the real weld location between rib and top, bottom plate, respectively. It is noted that the transverse rib (x-direction) is welded with both top and bottom plate, but the longitudinal rib (y-direction) is only welded with top plate. There is no welding between longitudinal rib and bottom plate. Figure VIII.5.2.2-3 is the combination of Figures VIII.5.2.2-1 and -2. Figure VIII.5.2.2-4 shows the two modified welding models from Fig. VIII.5.2.2-3 used in this analysis. Model A has the weld length 3 in., Model B has the weld length 2 in.

The top plate and bottom plate are connected by vertical beams (columns). Besides these vertical beams, there are horizontal beams supporting each plate. In transverse direction (see Fig. VIII.5.2.2-5) the rib is divided into two equal parts. They are modeled as transverse beams belonging to top and bottom plate, respectively. In longitudinal direction, see Fig. VIII.5.2.2-6), due to lack of welding between rib and bottom plate, the whole cross section of rib is modeled as a beam belonging to the top plate. In all horizontal beams, the principle bending moment of inertia is taken about the center line of the corresponding plate. Figures VIII.5.2.2-7 and -8 show the finite element model of the column and beam of the top plate, and bottom plate, respectively. The data of section properties of those columns and beams are listed below.

Section 1 (Column: 1, 4, 5, 6, 7, 8, 11, 12, 13, 15, 16, 17, 20,
21, 22, 23, 24, 25) (For Model B)

Fig. VIII.5.2.2-1 Real Weld Location on Top Plate

Fig.VIII.5.2.2-2 Real Weld Condition of Bottom Plate

Fig. VIII.5.2.2-3 Combination of Top and Bottom Plate Weld

Fig. VIII.5.2.2-4 Two Modified Welding Models Used in Analysis

Fig. VIII.5.2.2-5 Transverse Direction Rib

Fig. VIII.5.2.2-6 Longitudinal Direction Rib

Fig. VIII.5.2.2-7 Finite Element Model of Column and Beam of Top Plate

Fig. VIII.5.2.2-8 Finite Element Model of Column and Beam of Bottom Plate

VIII.5.3 Conclusion

From the summary, one can find that due to the unsymmetric distribution of the boundary element, a very high tensile force occurs at the highest boundary beam.

The highest weld stress of Model A is 14.5 ksi. In Model B, the high weld stress is 16.6 ksi. The stresses of top and bottom plate elements of both models have been plotted. By examining, it is seen that the bending stress σ_x at the top plate changes from negative to positive in the heavy line cross section. In other words, the bending moment changes from negative to positive in the heavy line cross section. This yields a maximum shear stress in this cross section as shown in both models.

VIII.6 STRESS ANALYSIS OF TIE-DOWN MECHANISMS

Each of the two tie-down mechanisms that are used to clamp the cask to the cradle consists of a strap and two tie-down clamp assemblies as shown in Figs. V.3.1.6-1 and V.3.1.6-2. The mechanisms are required to withstand a 2g loading.

Fig. VIII.6-1 Tie Down Mechanisms on LHM

The locations of the tie-down mechanisms, relative to the center of gravity of the LHM, are shown above. The load F_2 at mechanism 2 for the LHM weight of 91808 and a 2g acceleration is

$$F_2 = \frac{91808 \times 2 \times 96}{127} = 138,796 \text{ lb}$$

TIE-DOWN STRAPS

From Fig. V.3.1.6-2 the area of a strap is $3.75 \times 0.5 = 1.875 \text{ in}^2$. Because it encircles the cask, the load on the strap will be $138,796/2 = 69,398 \text{ lb}$ and the tensile stress

$$\sigma_T = \frac{69,398}{1.875} = 37,012 \text{ psi}$$

This is greater than the yield point of 30,000 psi for this stainless steel.

The total length of weld on the strap is (approximately)

$$L = \frac{2(11.5-4.25)}{\cos 30^\circ} + 3 = 19.7 \text{ in}$$

For this fillet weld the effective area in shear is

$$A = 0.707 \times 19.7 \times 0.375 = 5.22 \text{ in}^2$$

$$\tau = \frac{69,398}{5.22} = 13,295 \text{ psi}$$

Assuming a shear yield point of $\frac{2}{3} \times 30,000 = 20,000 \text{ psi}$

$$SF = \frac{20,000}{13,295} = 1.5$$

TIE-DOWN CLAMP

From Fig. V.3.1.6-1 part (A) is subjected to a load of 69,398 lb from the strap and reactions from each of the connecting rods (B) of $69,398/2 = 34,699$ lb. The maximum bending moment on the 2 in by 2 in section will be

$$M = 34,699 \times 1.25 = 43,374 \text{ in lb}$$

$$\frac{I}{c} = \frac{2^3}{6} = 1.33 \text{ in}^3$$

The bending stress is

$$\sigma = \frac{43,374}{1.33} = 32,612 \text{ psi}$$

The shear stress is

$$T = \frac{3}{2} \frac{43,374}{4} = 16,266 \text{ psi}$$

For a tensile yield point of 36,000 psi and a shear yield point of 22,000 psi

$$\text{SF based on tension} = \frac{36,000}{32,612} = 1.1$$

$$\text{SF based on shear} = \frac{22,000}{16,266} = 1.4$$

The connecting rods (B) are threaded at both ends with a 1-12 UNF thread having a stress area of 0.663 in^2 . Then

$$\sigma = \frac{34,699}{0.663} = 52,336$$

For an ultimate strength of 105,000 psi for the bolts

$$\text{SF} = \frac{105,000}{52,336} = 2.0$$

The parts (C), at the lower end of the rods, have an area across the 4.5 in hole equal to

$$A = 2 \times 2.5 \times 1.25 = 6.25 \text{ in}^2$$

Then the direct tensile stress is

$$\sigma_T = \frac{34,699}{6.25} = 5552 \text{ psi}$$

and

$$SF = \frac{36,000}{5,552} = 6.5$$

The end sections of shaft (D), of 2.25 in diameter, will experience a bending moment at the step equal to

$$\begin{aligned} M &= 34,699 \times (1.125 + 0.375) \\ &= 52048 \text{ in lb} \end{aligned}$$

$$\frac{I}{c} = \frac{\pi \times 2.25^3}{32} = 1.118 \text{ in}^3$$

At the step, the 0.125 in radius requires a stress concentration factor of approximately 2.2. Then

$$\sigma_{\max} = \frac{2.2 \times 52,048}{1.118} = 102,420 \text{ psi}$$

The shaft is made of 17-4 DH stainless steel in the -A- condition with a yield strength of 110,000 psi. Then

$$SF = \frac{110,000}{102,420} = 1.07$$

The pillow blocks (E) have an area across the hole equal to

$$A = 2 \times 1.125 \times 2.25 = 5.06 \text{ in}^2 \text{ and a tensile stress}$$

$$\sigma_T = \frac{34,699}{5.06} = 6858 \text{ psi}$$

$$SF = \frac{36,000}{6858} = 5.2$$

The section in direct shear on the bolting lugs has a shear area of

$$A = 2 \times 1.75 \times 2.25 = 7.88 \text{ in}^2$$

and a maximum direct shear stress of

$$\tau = \frac{3}{2} \times \frac{34,699}{7.88} = 6605 \text{ psi}$$

The two 3/4 - 10 UNC bolts have stress areas of 0.334 in² and body areas of $\pi/4 \times (3/4)^2 = 0.442$ in². Because the bolts and the bolting lugs have the same length and modulus of elasticity, the stiffnesses of the bolts and the joints will be proportional to their areas: For the lugs the area $2-1/4 \times 2-1/8 = 0.442 = 4.34$ in² will be used. The $K_J/K_B = 4.34/0.442 = 9.82$ and $K_B/K_J = 0.10$.

If F_J and F_B are the loads on the joint and bolt respectively, P is the applied load and F_O the tightening load then

$$F_J = F_O - \frac{P}{1 + \frac{K_B}{K_J}}$$

The joint will separate when $F_J = 0$ or

$$F_O = \frac{P}{1 + \frac{K_B}{K_J}} = \frac{34,699/2}{1.1} = 15,772 \text{ lb}$$

Assume $F_O = 16,000$ lb. Then

$$F_B = F_O + \frac{P}{1 + \frac{K_J}{K_B}} = 16,000 + \frac{34,699/2}{10.82} = 17,603 \text{ lb}$$

$$\sigma_T = \frac{17,603}{0.334} = 52,703 \text{ psi}$$

$$SF = \frac{100,000}{52,703} = 1.9$$

VIII.7 STRESS ANALYSIS OF HORIZONTAL HOLD-DOWN CLAMPS

Figure V.3.1.7-1 shows one of the four horizontal hold-down clamps with a 2g (50000lb) load applied. This load was arrived at by assuming that the LHM weighs 100,000 lb instead of 91,808 lb. For a 2g loading carried by four clamps, $100,000 \times \frac{2}{4} = 50,000$ lb.

BOLT STRESSES

$$\begin{aligned} 1\frac{1}{8} - 7 \text{ UNC bolts, stress area} &= 0.763 \text{ in}^2 \\ \text{body area} &= \frac{\pi}{4} \times 1\frac{1}{8}^2 = 0.994 \text{ in}^2 \end{aligned}$$

$$\begin{aligned} \frac{3}{4} - 10 \text{ UNC bolts, stress area} &= 0.334 \text{ in}^2 \\ \text{Body area} &= \frac{\pi}{4} \times 0.75^2 = 0.442 \text{ in}^2 \end{aligned}$$

Assume SF = 2.1, then

$$\sigma_W = \frac{105,000}{2.1} = 50,000 \text{ psi}$$

Initial tightening loads F_o

$$\text{For } 1\frac{1}{8} \text{ bolts, } F_o = 50,000 \times 0.763 = 38,150 \text{ lb}$$

$$\text{For } \frac{3}{4} \text{ bolts, } F_o = 50,000 \times 0.334 = 16,700 \text{ lb}$$

Figure VIII.7-1(a) shows one of the bearing areas indicated in Fig. V.3.1.7-1. Because the bolts and the plates through which they extend (hereafter called the joints, subscript J) have the same length and the same modulus of elasticity, their stiffnesses are proportional to their areas; in this case the areas abcd and defg are utilized.

$$\begin{aligned} \text{For the } 1\frac{1}{8} \text{ in bolt, } K_B &= 0.994 \text{ in}^2, K_J = \text{area abcd} = 2.5^2 - 0.994 \\ &= 5.256 \text{ in}^2. K_J/K_B = 5.29, K_B/K_J = 0.189. \end{aligned}$$

$$\begin{aligned} \text{For the } 3/4 \text{ in bolt, } K_B &= 0.442, K_J = \text{area efgh} = 3.5 \times 2.5 - 0.442 \\ &= 8.31 \text{ in}^2. K_J/K_B = 18.8, K_B/K_J = 0.053. \end{aligned}$$

Fig. VIII.7-1 Bolt Bearing Areas in Fig. V.3.1.7-1

The 50,000 lb load in Fig. V.3.1.7-1 will be divided equally between the two bearing areas. Then direct load on each bolt area = $25,000/2 = 12,500$ Bending moment = $4.25 \times 25,000 = 106,250$ in lb. On each bolt area, load = $106,250/4.5 = 23611$ lb, tension at $1\frac{1}{8}$ in bolt area, compression at $\frac{3}{4}$ in bolt area. The net external loads on the bolt areas are shown in Fig. VIII.7-1(a).

For the tensile load at $1\frac{1}{8}$ in bolt area the resultant load on the bolt (refer to Fig. VIII.7-1(b)).

$$F_B = F_o + \frac{P_T}{1 + \frac{K_J}{K_B}} = 38,150 + \frac{36,111}{6.29} = 43,891 \text{ lb}$$

$$\sigma = \frac{43,891}{0.763} = 57,524 \text{ psi}$$

$$SF = \frac{105,000}{57,524} = 1.83$$

$$F_J = F_o - \frac{P_T}{1 + \frac{K_B}{K_J}} = 38,150 - \frac{36,111}{1.189}$$

$$= + 7779 \text{ lb } (\therefore \text{ JOINT HAS NOT SEPARATED})$$

For the compression load at $\frac{3}{4}$ in bolt area, the resultant load on the bolt

$$F_B = F_o - \frac{P_c}{1 + \frac{K_J}{K_B}} = 16,700 - \frac{1111}{19.8} = 16,644 \text{ lb}$$

$$\sigma = \frac{16,644}{0.334} = 49,832 \text{ psi}$$

$$SF = \frac{105,000}{49,832} = 2.1$$

$$F_J = F_o + \frac{P_c}{1 + \frac{K_B}{K_J}} = 16,700 + \frac{1111}{1.053} = 17,755 \text{ lb}$$

For cross-section AB in Fig. V.3.1.7-3

$$A = 5\frac{11}{16} \times 2\frac{1}{4} = 12.8 \text{ in}^2$$

$$M = 5 \times 10^4 \times \frac{1}{2} = 2.5 \times 10^4 \text{ in. lb}$$

$$\frac{I}{c} = \frac{1}{6} \times 5\frac{11}{16} \times 2\frac{1}{4}^2 = 4.8 \text{ in}^4$$

$$\tau_{\max} = \frac{3}{2} \times \frac{5 \times 10^4}{12.8} = 5859 \text{ psi}$$

Nominal bending stress

$$\sigma = \frac{2.5 \times 10^4}{4.8} = 5208 \text{ psi}$$

Because of the 3/32 in. radius at the step, a stress concentration factor will be applied.

$$\text{For } \frac{r}{d} = \frac{3}{32 \times 2.25} = 0.0417, \quad \frac{D}{d} = \frac{3\frac{15}{16}}{2\frac{1}{4}} = 1.75$$

$$K_t = 2.4$$

$$\sigma_{\max} = 2.4 \times 5208 = 12,499 \text{ psi}$$

Based on normal stress

$$SF = \frac{50,000}{12,499} = 4$$

$$\text{Assuming } \tau_y = \frac{2}{3} \times 50,000 = 33,333 \text{ psi}$$

Based on shear stress

$$SF = \frac{33,333}{5859} = 5.7$$

Under the 50,000 lb load the two tongues in Fig. V.3.1.7-3 will be twisted, producing a torsional shear stress on the cross-sections CD. In this state the tongues are loaded like a twisted bar with a rectangular cross-section

Fig. VIII.7-2 Section CD in Fig. V.3.1.7-3

$$T = 25,000 \times 3\frac{3}{4} = 93,750 \text{ in. lb}$$

From Theory of Elasticity

$$\tau_{\max} = \frac{T}{abc^2} \quad \text{for} \quad \frac{b}{c} = 3.25, \alpha = 0.271$$

$$= \frac{93,750}{0.271 \times 6.5 \times 4} = 13,305 \text{ psi}$$

This stress occurs at the center of the long sides, points A and B.

There is also a direct shear stress which is zero at A and B and has a maximum value on line EF of

$$\tau = \frac{3}{2} \times \frac{25,000}{6.5 \times 2} = 2885 \text{ psi}$$

$$SF = \frac{33,333}{13,305} = 2.5$$

In addition to twisting the tongues in Fig. V.3.1.7-3 the 50,000 lb load will also cause bending and tension in the section AB adjacent to the tongue grooves in Fig. V.3.1.7-2. The stressed cross-section ABCD and the maximum tensile stress are shown in Fig. VIII.7-3.

Fig. VIII.7-3 Section AB in Fig. V.3.1.7-2

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