WESTINGHOUSE CLASS 3

WCAP-10168

EVALUATION OF THE ACCEPTABILITY OF THE REACTOR VESSEL HEAD LIFT RIG, REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL, LOAD CELL LINKAGE AND REACTOR COOLANT PUMP MOTOR LIFT SLING TO THE REQUIREMENTS OF NUREG 0612

for

FLORIDA POWER AND LIGHT COMPANY
TURKEY POINT UNITS 3 AND 4

DECEMBER, 1982

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ABSTRACT

An evaluation of the Turkey Point Units 3 and 4 reactor vessel head and internal lift rigs, load cell, load cell linkage and reactor coolant pump motor lift sling was performed to determine the acceptability of these devices to meet the requirements of NUREG 0612. The evaluation consists of: (1) a comparison report of the ANSI N14.6 requirements and the requirements used in the design and manufacture of these devices; (2) a stress report in accordance with the design criteria of ANSI N14.6; and (3) a list of recommendations to enable these devices to demonstrate compliance with the intent of NUREG 0612 and ANSI N14.6.

58858:1/120382

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B. Stress Report - Reactor Vessel Head Lift Rig, Reactor Vessel Internals Lift Rig, Load Cell, Load Cell Linkage and Reactor Coolant Pump Motor Lift Sling for Florida Power and Light Company, Turkey Point Units 3 and 4.

REFERENCES

1. George, H., Control of Heavy Loads at Nuclear Power Plants Resolution of Generic Technical Activity A-36, NUREG 0612, July, 1980.

2. ANSI N14.6-1978 Special Lifting Devices for Shipping Containers Weighing 10,000 Pounds or More for Nuclear Material

3. ANSI B 30.9-1971. Slings, American National Standards Institute, New York, 1971

4. Westinghouse Drawing 685J249, Head Lifting Rig General Assembly

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ATTACHMENT A
to WCAP-10168

COMPARISON OF ANSI N14.6-1978 REQUIREMENTS FOR SPECIAL LIFTING DEVICES AND THE REQUIREMENTS FOR THE REACTOR VESSEL HEAD LIFT RIG, REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL, LOAD CELL LINKAGE AND REACTOR COOLANT PUMP MOTOR LIFT SLING

FOR

FLORIDA POWER AND LIGHT COMPANY
TURKEY POINT UNITS 3 AND 4

December 1982

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5888B:1/120682

ABSTRACT

The requirements used in the original design, fabrication, testing, maintenance and quality assurance were compared to the ANSI N14.6-1978 requirements for the Turkey Point Units 3 and 4 reactor vessel head and internals lift rig, load cell, load cell linkage and reactor coolant pump motor lift sling. A critical items list per section 3.1.2 has been prepared and a tabulation of ANSI N14.6 requirements that are, at present, incompatible with the Turkey Point Units 3 and 4 lifting devices has been prepared.

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3. Westinghouse Drawing AED-SK-618J644TXK Handling Sling Spreader for Motor SV-4M-A1 Pump General Assembly

SECTION 1
PURPOSE

The purpose of this report is to compare the requirements of the special lifting rigs used to lift the reactor vessel head, reactor vessel upper and lower internals, and the reactor coolant pump motor with the requirements contained in ANSI N14.6 for special lifting devices.

SECTION 1
INTRODUCTION

The Nuclear Regulatory Commission (NRC) issued NUREG 0612 "Control of Heavy Load at Nuclear Power Plants"[1] in 1980 to address the control of heavy loads to prevent and mitigate the consequences of postulated accidental load drops. NUREG 0612 imposes various training, design, inspection and procedural requirements for assuring safe and reliable operation for the handling of heavy loads. In the containment building, NUREG 0612 Section 5.1.1(4) requires special lifting devices to meet the requirements of ANSI N14.6-1978-"American National Standard for Special Lifting Devices for Shipping Containers Weighing 10,000 Pounds or More for Nuclear Materials"[2]. In general, ANSI N14.6 contains detailed requirements for the design, fabrication, testing, maintenance, and quality assurance of special lifting devices. In addition, ANSI N14.6 required that when wire rope or chain is used in the design of a special lifting device, the wire rope or chain shall be in conformance with ANSI B30.9-1971 "American National Standard Safety Standard for Slings."[3] The Turkey Point lifting devices which can be categorized as special lifting devices and which are contained in the scope of this report are:

1. Reactor vessel head lift rig

2. Reactor vessel internals lift rig

3. Load cell

4. Load cell linkage

5. Reactor coolant pump motor lift sling.

This report contains the evaluation performed on these lifting devices to determine the acceptability of these devices to meet the above requirements.

1.1 BACKGROUND

The reactor vessel head lift rig, the reactor vessel internals lift rig, load cell, load cell linkage and reactor coolant pump motor lift sling were designed and built for the Turkey Point Units 3 and 4 circa 1967-68. The actual design criteria is unknown for the lifting devices. It appears that for the head lift rig, internals lift rig, load cell, load cell linkage and the reactor coolant pump motor lift sling that in most cases the design criteria used was that the resulting stress in the load carrying members, when subjected to the total combined lifting weight, does not exceed one fifth (1/5) of the ultimate strength of the material. These items were not classified as nuclear safety components and requirements for formal documentation of design requirements and stress reports were not applicable. Thus, stress reports and design specifications were not formally documented. Westinghouse defined the design, fabrication and quality assurance requirements on detailed manufacturing drawings and purchase order documents. Westinghouse also issued field assembly and operating instructions, where applicable.

SECTION 2
COMPONENT DESCRIPTION

2.1 REACTOR VESSEL HEAD LIFT RIG

The reactor vessel head lift rig[4] (Figure 2-1) is a three legged carbon steel structure, approximately 35 feet high and 14 feet in diameter, weighing approximately 25,000 pounds. It is used to handle the assembled reactor vessel head.

The three vertical legs and Control Rod Drive Mechanism (CRDM) platform assembly are permanently attached to the reactor vessel head lifting lugs. The tripod assembly is attached to the three vertical legs and is used when installing and removing the reactor vessel head. During plant operation, the sling assembly is removed and the three vertical legs and platform assembly remain attached to the reactor vessel head.

2.2 REACTOR VESSEL INTERNALS LIFT RIG

The internals lifting rig[5] (Figure 2-2) is a three-legged carbon and stainless steel structure, approximately 29 feet high and 12 feet in diameter weighing approximately 13,000 pounds. It is used to handle the upper and lower reactor vessel internals packages. It is attached to the main crane hook for all lifting, lowering and traversing operations. A load cell linkage is connected between the main crane hook and the rig to monitor loads during all operations. When not in use, the rig is stored on the upper internals storage stand.

The rig attaches to the internals packages by means of three engaging screws which are screwed into tapped holes in the internals flanges. These screws are manually operated from the spreader assembly using the integral tools. The screws are normally spring retracted upward and are depressed to engage the tapped holes.

2.3 LOAD CELL AND LOAD CELL LINKAGE

The load cell is used to monitor the load during lifting and lowering the reactor vessel internals to ensure no excessive loadings are occurring. It is installed between the load cell linkage and the lifting device. The load cell is a strain gage (tension) type, rated at 300,000 pounds and built by W. C. Dillon and Co. The load cell linkage is an assembly of pins, plates and bolts which connect the polar crane main hook to the load cell.

2.4 REACTOR COOLANT PUMP MOTOR LIFT SLING

The reactor coolant pump motor lift sling [6] (Figure 2-3) consists of a six (6) foot triangular carbon steel spreader assembly weighing approximately one thousand (1,000) pounds. It has wire rope slings, shackles and turnbuckles attached which form a three point lift assembly. It is used to handle the reactor coolant pump motor.

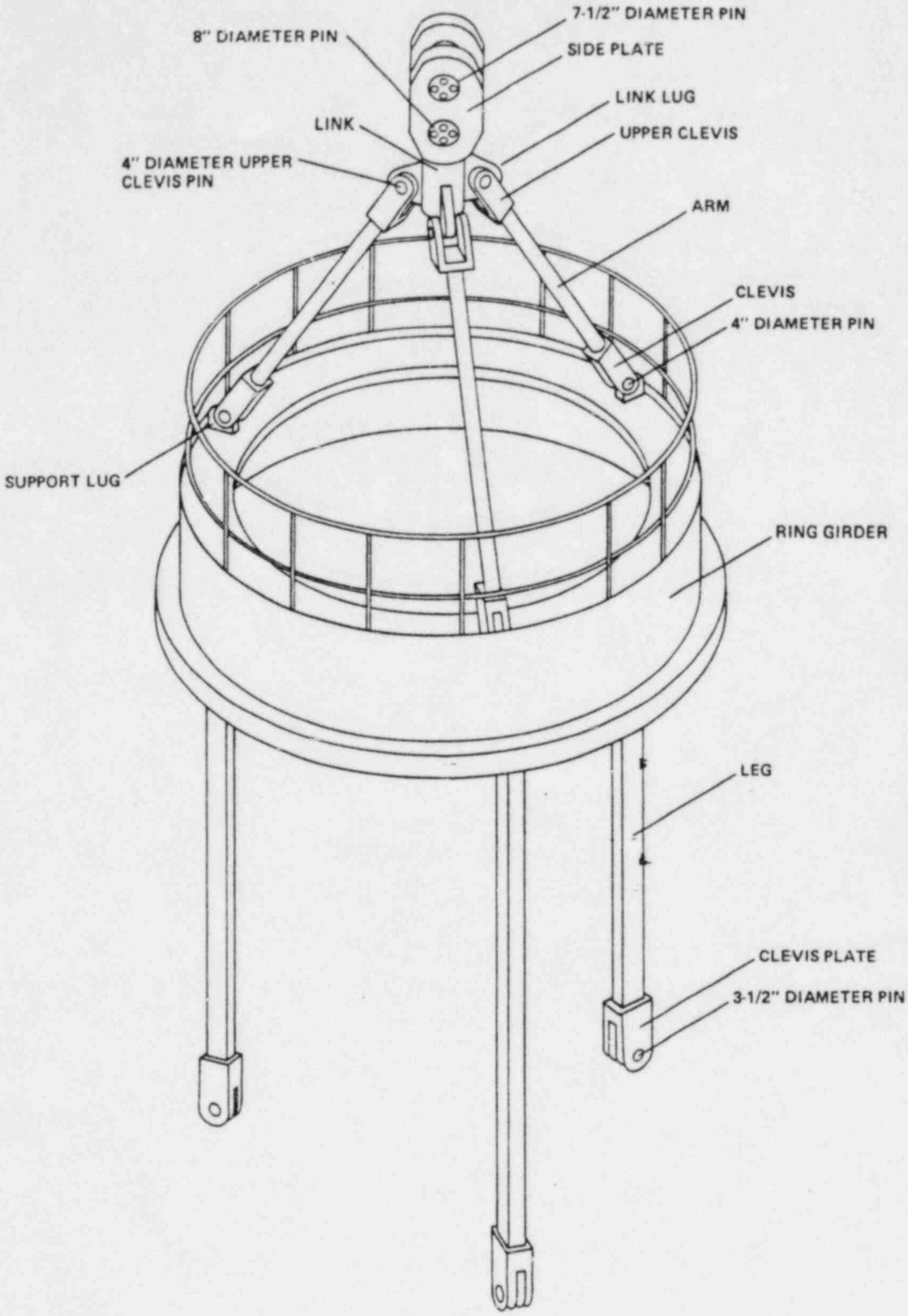

Figure 2-1. Reactor Vessel Head Lifting Rig

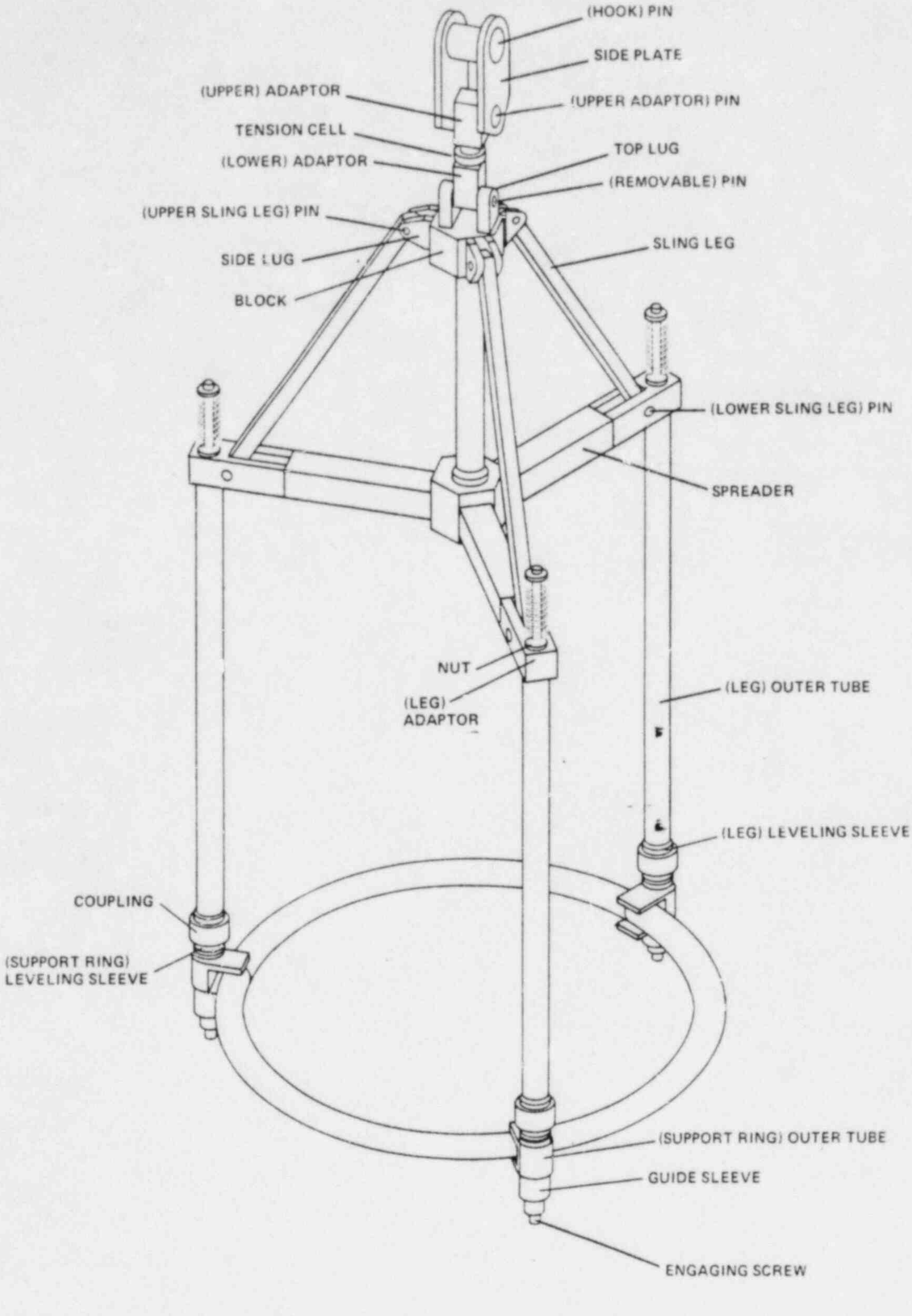

Figure 2-2. Reactor Vessel Internals Lifting Rig

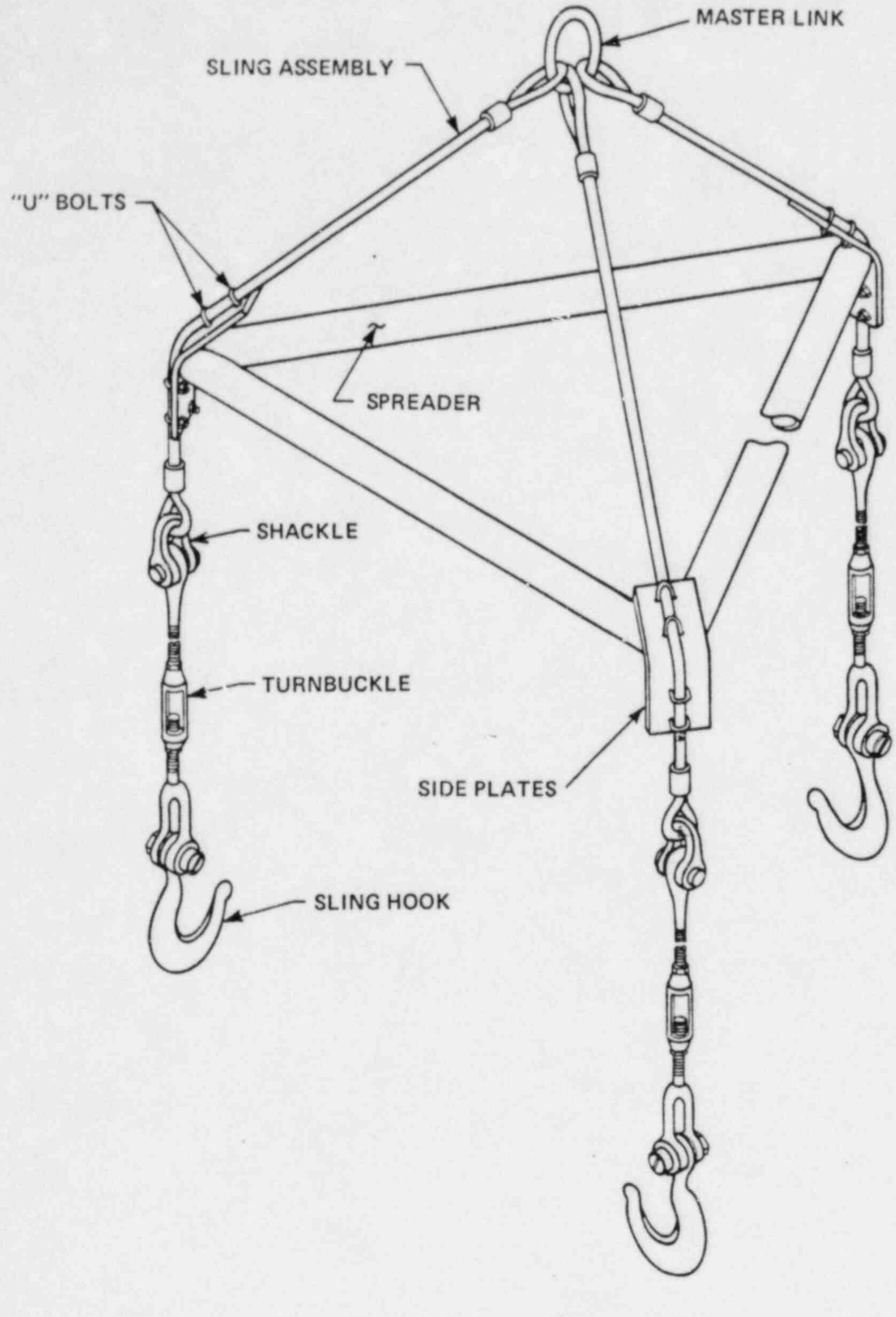

Figure 2-3. Reactor Coolant Pump Motor Lift Sling

SECTION 3
SCOPE OF EVALUATION

The evaluation of these lifting devices consists mainly of three parts:

1. A detailed review of the ANSI N14.6 requirements

2. Preparation of a stress report

3. Recommendations to demonstrate compliance with NUREG 0612, Section 5.1.1(4).

Discussion of these items follows.

3.1 REVIEW OF ANSI N14.6-1978

A detailed comparison was made of the information contained in ANSI N14.6 with the information that was used to design, manufacture, inspect and test these special lifting devices.* The detailed comparison is provided in three parts:

1. Overall item by item comparison of requirements

2. Preparation of a critical item list per ANSI N14.6 Section 3.1.2, and

3. Preparation of a list of nonconforming items.

This detailed analysis is contained in Attachment A to this report.

3.2 PREPARATION OF A STRESS REPORT

Section 3.1.3 of ANSI N14.6 and NUREG 0612 Section 5.1.1(4) require a stress report to be prepared. Special loads and allowable stress criteria are specified for this analysis. The stress report is Attachment B to this report.

3.3 RECOMMENDED ACTIONS

An obvious result from the previous evaluations is a list of items that can be performed to demonstrate to the NRC that these special lifting devices are in compliance with the guidelines of ANSI N14.6 and NUREG 0612 Section 5.1.1(4). These recommendations are identified in Section 6.

SECTION 4
DISCUSSION OF EVALUATIONS

4.1 STUDY OF ANSI N14.6-1978

A review of ANSI N14.6 identifies certain analyses to be performed and certain identifications that are required to be made to demonstrate compliance with this document. These are a preparation of a stress report in accordance with Section 3.2 and a preparation of a critical items list in accordance with Section 3.1.2. The stress report is Attachment B to this report. The critical items list has been prepared per Section 3.1.2 and is contained in Appendix A to Attachment A. This list identifies the critical load path parts and welds, the materials of these items, and the applied non-destructive volumetric and surface inspections that were performed. (Details of these non-destructive processes and acceptance standards are available at Westinghouse should they be needed.)

A detailed item by item comparison of all the requirements of ANSI N14.6 and those used for the design, manufacture and inspection of these lifting devices is contained as Table 2-1 of Attachment A. The comparison shows that these devices meet the intent of the ANSI document for design, fabrication and quality control. However, they do not meet the requirements of ANSI N14.6 for periodic maintenance, proof and functional testing. Thus, a tabulation of those ANSI N14.6 requirements that are incompatible with these lifting devices was prepared and is Appendix B to Attachment A. Included in Appendix B to Attachment A are recommended actions that may be used to demonstrate acceptability to the NRC.

4.2 STRESS REPORT

As part of the invoking of the ANSI N14.6 document, the NRC requested utilities to demonstrate their compliance with the stress criteria with some qualifying conditions. Attachment B is the stress report for these devices performed in accordance with the criteria of ANSI N14.6. A discussion is included which responds to the NRC qualifying conditions of NUREG 0612. All of the tensile and shear stresses with the exception of the tensile stress at the minimum section of the internals engaging screw, item 23, meet the design criteria of Section 3.2.1.1 of ANSI N14.6, requiring application of stress design factors of three and five with the accompanying allowable stress limits of yield and ultimate strength, respectively. In the engaging screw, the tensile stress slightly exceeds the criteria that three times the calculated stress must be less than the yield stress. However, the conservative criterion that five times the calculated stress must be less than the ultimate is met.

Application of Section 3.2.1.1 of ANSI N14.6 criteria to pins subject to bending, structural members subject to buckling and bending loads, and various parts subject to bearing loads, result in some stresses exceeding this criteria. However, when using more appropriate criteria, the resulting stresses are acceptable.

4.3 RECOMMENDATIONS

The recommendations identified in Section 6 require a review of the existing Turkey Point maintenance and operating instructions to ensure that they contain information relative to the identification, maintenance and periodic testing required by ANSI N14.6. The extent of the periodic testing is also addressed and the recommendations identify procedures which are intended to fully meet the intent of NUREG 0612 and ANSI N14.6 with the least amount of perturbation to the refueling sequence.

SECTION 5
CONCLUSIONS

The following conclusions are apparent as a result of this evaluation:

1. The ANSI N14.6 requirements for design, fabrication and quality assurance are generally in agreement with those used for these special lift devices.

2. The ANSI N14.6 criteria for stress limits associated with certain stress design factors for tensile and shear stresses are adequately satisfied with the exception noted in Section 4.2.

3. These devices are not in strict compliance only with the ANSI N14.6 requirements for acceptance testing, maintenance and verification of continuing compliance. Recommendations are included to identify actions that should enable these devices to be considered in compliance with the intent of ANSI N14.6.

4. The application of the ANSI N14.6 criteria for stress design factor of 3 and 5 are only for shear and tensile loading conditions. Other loading conditions are to be analyzed to other appropriate criteria.

SECTION 6
RECOMMENDATIONS

The following recommendations address the areas of ANSI N14.6 which are incompatible with the present lifting devices and which are considered most important in demonstrating the continued reliability of these devices. They consist of suggestions and proposed responses to identify compliance to the NRC and future considerations.

6.1 Review the existing operating procedures to include consideration of ANSI N14.6 Sections 5.1.3 through 5.1.8. These sections include requirements for: scheduled periodic testing; special identification and marking; maintenance, repair, testing and use. Westinghouse remarks on addressing these sections are listed in Attachment A, Appendix B, Items 7, 8, and 9.

6.2 A proposed response to the requirement of ANSI N14.6 Section 5.2.1 requiring an initial acceptance load test prior to use equal to 150 percent of the maximum load is as follows:

The head lift rig was load tested and inspected at assembly to approximately 100 percent of the load. The internals lift rig and the R.C. pump motor lift sling were not required to be load tested.

6.3 A proposed response to ANSI N14.6 Section 5.3 which requires, annually, either a 150 percent maximum load test or dimensional, visual and non-destructive testing of major load carrying welds and critical areas follows. (Since the 150 percent load test is very impractical, the approach identified in the following recommendation is to perform a minimum of non-destructive testing.)

a. Reactor Vessel Head Lift Rig:

Prior to use and after reassembly of the head lifting rig sling assembly to the platform assembly, visually check the clevis to leg fillet welds at the bottom end of the legs and the support lug to ring girder fillet welds on the platform. Raise the vessel head slightly above its support and hold for 10 minutes. During this time, continue to visually inspect these welds. If no problems are apparent, continue to lift.

b. Reactor Vessel Internals Lift Rig

Prior to use, visually inspect the rig components and welds while on the storage stand for signs of cracks or deformation. Check all bolted joints to ensure that they are tight and secure. After connection to the upper or lower internals, raise the assembly slightly off its support and hold for 10 minutes. If no problems are apparent, continue to lift, monitoring the load cell readout at all times.

c. Reactor Coolant Pump Motor Lift Sling

Prior to use, visually inspect the rig components and welds for signs of cracks, deformation, kinks, or frayed ends. Check all bolted joints to ensure that they are tight and secure. After connection to the pump motor, raise the assembly slightly off its support and hold for 10 minutes. During this time visually check the spreader welds. If no problems are apparent, continue to lift.

The above actions do not include a non-destructive test of these welds because:

a. Access to the welds for surface examination is difficult. These rigs are in containment and some contamination is present.

b. All tensile and shear stresses in the welds are well within the allowable stress with the exception of the internals engaging screw.

c. The items that are welded remain assembled and cannot be misused for any other lift other than their intended function.

d. To perform non-destructive tests would require:

(1) Removal of paint around the area to be examined which is contaminated.

(2) Performance of either magnetic particle inspection or liquid penetrant inspection and

(3) Repainting after testing is completed.

(4) Cleanup of contaminated items.

Performing non-destructive tests on these welds every refueling would increase the critical path refueling time.

Dimensional checking is not included since these structures are large (about 14 ft. dia. by 35 ft. high) and the results of dimensional checking would always be questionable. Other checks on critical load path parts such as pins, are also not included since an examination of these items would require disassembly of the special lift devices.

6.4 Recommend that a periodic non-destructive surface examination of critical welds and/or parts be performed once every ten years as part of an inservice inspection outage.

6.5 Recommend that a modification be made to the load cell to be adaptable to both the head and internals lift rig. Thus, monitoring of these loads during movement can be accomplished at all times.

6.6 Recommend that no changes be made to the reactor vessel internals lift rig should the stresses, discussed in Attachment B, be considered excessive by others because:

a. The design weight used in the stress calculations is based on the weight of the lower internals. The lower internals are only removed when a periodic inservice inspection of the vessel is required (once/10 years).

b. Prior to removal of the lower internals, all fuel is removed. Thus the concern for handling over fuel is non-existant in this particular case.

c. Normal use of the rig is for moving the upper internals, which weigh less than 1/2 of the lower internals. The design weight is based on lifting the lower internals. Thus all the stresses could be reduced by approximately 50 percent and considered well within the ANSI N14.6 criteria for stress design factors.

6.7 Recommended that the internals lift rig sling block be changed to a forged block with welded side lugs to reduce the number of welds, should the NRC require yearly surface inspection of welds.

6.8 Recommended that plates be added to the sides of the spreader (item 14) to reinforce it. Plates should be of sufficient size to satisfy the emperical equations for structures loaded in combined compression and bending, as described in the ASME Boiler and Pressure Vessel Code Section III, Appendix XVII.

SECTION 2
INTRODUCTION

ANSI N14.6-1978-"American National Standard for Special Lifting Devices for Shipping Containers Weighing 10,000 Pounds or More for Nuclear Materials" contains detailed requirements for the design, fabrication, testing, maintenance and quality assurance of special lifting devices. NUREG 0612 "Control of Heavy Load at Nuclear Power Plants", paragraph 5.1.1(4), specifies that special lifting devices should satisfy the guidelines of ANSI N14.6-1978. Subsequently the Nuclear Regulatory Commission (NRC) has requested operating plants to demonstrate compliance with NUREG 0612. To demonstrate compliance with this document, a detailed comparison of the original design, fabrication, testing, maintenance and quality assurance requirements with those of ANSI N14.6 is necessary.

Thus, the ANSI N14.6 document has been reviewed in detail and compared to the requirements used to design and manufacture the reactor vessel head lift rig, the reactor vessel internals lift rig, load cell, load cell linkage and the reactor coolant pump motor lift sling. This comparison is listed in Table 2-1.

2.1 BACKGROUND

The reactor vessel head lift rig, the reactor vessel internals lift rig, load cell, load cell linkage and reactor coolant pump motor lift sling were designed and built for the Turkey Point Units 3 and 4, circa 1967-63. The actual design criteria is unknown for the lifting devices. It appears that for the head lift rig, internals lift rig, load cell, load cell linkage and the reactor coolant pump motor lift sling that in most cases the design criteria used was that the resulting stress in the load carrying members, when subjected to the total combined lifting weight, does not exceed one fifth (1/5) of the ultimate strength of the material. Westinghouse also required non-destructive tests and inspections on most critical load path parts and welds both as raw material and as finished items. These requirements of design, manufacturing and quality assurance were identified on detailed manufacturing drawing and purchasing documents.

Westinghouse also issued field assembly and operating instructions, where applicable.

2.2 COMPONENT DESCRIPTION

2.2.1 Reactor Vessel Head Lift Rig

The reactor vessel head lift rig[1] is a three legged carbon steel structure, approximately 35 feet high and 14 feet in diameter, weighing approximately 25,000 pounds. It is used to handle the assembled reactor vessel head.

The three vertical legs and control rod drive mechanism (CRDM) platform assembly are permanently attached to the reactor vessel head lifting lugs. The tripod sling assembly is attached to the three vertical legs and is used when installing and removing the reactor vessel head. During plant operations, the sling assembly is removed and the three vertical legs and platform assembly remain attached to the reactor vessel head.

2.2.2 Reactor Vessel Internals Lift Rig

The reactor vessel internals lift rig[2] is a three-legged carbon and stainless steel structure, approximately 29 feet high and 12 feet in diameter weighing approximately 13,000 pounds. It is used to handle the upper and lower reactor vessel internals packages. It is attached to the main crane hook for all lifting, lowering and traversing operations. A load cell linkage is connected between the main crane hook and the rig to monitor loads during all operations. When not in use, the rig is stored on the upper internals storage stand.

The rig attaches to the internals packages by means of three engaging screws which are screwed into tapped holes in the internals flanges. These screws are manually operated from the spreader assembly using the integral tools. The screws are normally spring retracted upward and are depressed to engage the tapped holes.

2.2.3 Load Cell and Load Cell Linkage

The load cell is used to monitor the load during lifting and lowering the reactor vessel internals to ensure no excessive loadings are occurring. It is installed between the load cell linkage and the lifting device. The load cell is a strain gage (tension) type, rated at 300,000 pounds, built by W. C. Dillon and Co. The load cell linkage is an assembly of pins, plates and bolts which connect the polar crane main hook to the load cell.

2.2.4 Reactor Coolant Pump Motor Lift Sling

The reactor coolant pump motor lift sling[3] consists of a six (6) foot triangular carbon steel spreader assembly weighing approximately one thousand (1,000) pounds. It [illegible] wire rope slings, shackles and turnbuckles attached which form a three point lift assembly. It is used to handle the reactor coolant pump motor.

TABLE 2-1

COMPARISON OF THE REQUIREMENTS OF ANSI N14.6 AND TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
1 1.1 to 1.3 2	Scope and Definitions - These sections define the scope of the document and include pertinent definitions of specific items	These sections are definitive, and not requirements.
3 3.1 3.1.1 to 3.1.4	Design Designer's Responsibilities - This section contains requirements for preparing a design specification and its' contents, stress reports; repair procedures; limitations on use with respect to environmental conditions; marking and nameplate information; and critical items list.	A. No design specification was written concerning these specific requirements. However, assembly and detailed manufacturing drawings and purchasing documents contain the following requirements: (1) Material specification for all of the critical load path items to ASTM, ASME specifications or special requirements listed. (2) All welding, weld procedures and welds to be in accordance with ASME Boiler and Pressure Vessel Code - Section IX or Westinghouse specifications (3) Special non-destructive testing for specific critical load path items to be performed to written and approved procedures in accordance with ASTM or specified requirements

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
		(4) All coatings to be performed to strict compliance with specified requirements.
		(5) Letters of compliance for materials and specifications were required for verification with original specifications.
		B. A stress report was not originally required but has been prepared and is Attachment B.
		C. Repair procedures were not identified.
		D. No limitations were identified as to the use of these devices under adverse environments.
		E. Markings and nameplate information was not addressed except for the reactor coolant pump motor lifting which identifies MAXCAPACITY & TOOL (Drawing) Number.
		F. Critical item lists have been prepared for each device and are identified as Appendix A to this Attachment A.

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
3.2 3.2.1 to 3.2.6	<u>Design Criteria</u> <u>Stress Design Factors</u> - These sections contain requirements for the use of stress design factors of 3 and 5 for allowable stresses of yield and ultimate respectively for maximum shear and tensile stresses; high strength material stress design factors; special pins; wire rope and slings to meet ANSI B30.9-1971; and drop-weight tests and Charpy impact test requirements	1. The actual design criteria is unknown for the lifting devices. It appears that for the head lifting rig, internals lift rig, load cell, and the reactor coolant pump motor lift sling that in most cases the design criteria used was that the resulting stress in the load carrying members, when subjected to the total combined lifting weight, does not exceed one fifth of the ultimate strength of the material. A stress report (Attachment B) has been generated which addresses the capability of these rigs to meet the ANSI design stress factors. 2. High strength materials are used in some of these devices (mostly for pins). Although the fracture toughness was not determined, the material was selected based on it's fracture toughness characteristics. However, the stress design factors of ANSI N14.6 Section 3.2.1 of 3 and 5 were used in the analysis and the resulting stresses are acceptable. 3. Where necessary, the weight of pins was considered for handling.

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
		4. Wire rope is used only in the design of the R.C. pump motor lift sling. However, the wire rope is of a special design and thus this device is considered a special lift device. 5. Drop weight and Charpy impact tests were not required nor performed.
3.3 3.3.1 to 3.3.8	Design Considerations - These sections contain considerations for; materials of construction, lamellar tearing; decontamination effects; remote engagement provisions; equal load distribution; lock devices; position indication of remote actuators; retrieval of device if disengaged; and nameplates.	Decontamination was not specifically addressed. Lamellar tearing was not considered but the designs of these devices are not susceptible to this type of joint deterioration. Even distribution of the load is evident from these designs. Locking plates, pins, etc. are used throughout these special lifting devices. Remote actuation is only used when engaging the internals lift rig with the internals, however, no position indication of engagement was provided. However, all these items were considered and the designs reflect these requirements.
3.4 3.4.1 to 3.4.6	Design Considerations to Minimize Decontamination Efforts in Special Lifting Device Use - These sections contain fabrication, welding, finishes, joint and machining requirements to permit ease in decontamination.	Decontamination was not specifically addressed. However, the design and manufacture included many of these items, i.e. lock devices, pins, etc.

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
3.5 3.5.1 to 3.5.10	Coatings - These sections contain provisions for ensuring proper methods are used in coating carbon steel surfaces and for ensuring non-contamination of stainless steel items.	The requirements for coating carbon steel surfaces are contained in a Westinghouse process specification referenced on the assembly and detail drawings except for the reactor coolant pump motor lift sling. (The coating requirements for the reactor coolant pump motor lift sling are contained in purchasing documents and require proper preparation and application of an epoxy paint.) This specification requires a proven procedure, proper cleaning, preparation, application and final inspection of the coating. These requirements meet the intent of 3.5.1 through 3.5.8. No provisions were included in these designs for consideration of decontamination materials or the use of non-contaminating contact materials for use in stainless steel parts.
3.6 3.6.1 to 3.6.3	Lubricants - These sections contain requirements for special lubricants to minimize contamination and degradation of the lubricant and contacted surfaces or water pools	No specific lubrication requirements have been identified. However, neolube is recommended for use with the engaging screws in the internals lift device which are under water and silicone grease for the load cell pins which are out of water.

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
4 4.1 4.1.1 to 4.1.12	Fabrication Fabricators Responsibilities -These sections contain specific requirements for proper quality assurance, document control, deviation control, procedure control, material identification and certificate of compliance.	A formal quality assurance program for the manufacturer was not required. However, all the manufacturers welding procedures and non-destructive testing procedures were reviewed by Westinghouse prior to use. All critical load carrying members require letters of compliance for material requirements. Westinghouse performed certain checks and inspections during various steps of manufacturing. Final Westinghouse review includes visual, dimensional, procedural, cleanliness, personnel qualification, etc. and issuance of a quality release* to ensure conformance with drawing requirements.
4.2 4.2.1 to 4.2.5	Inspectors Responsibilities -These sections contain requirements for a non-supplier inspector.	Westinghouse Quality Assurance personnel performed some inprocess and final inspections similar to those identified in these sections. (Also see comments to Section 4.1 above)
4.3 4.3.1 to 4.3.3	Fabrication Considerations -These sections contain special requirements for ease in decontamination or control of corrosion.	General good manufacturing processes were followed in the manufacture of these devices. However, the information defined in these sections was not specifically addressed.

* These remarks are not applicable to the Reactor Coolant Pump Motor Lift Sling

TABLE 2-1 (cont)

COMPARISON OF REQUIREMENT OF THE ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
5 5.1 5.1.1 to 5.1.8	<u>Acceptance Testing Maintenance, and Assurance of Continued Compliance Owner's Responsibilities</u> - Sections 5.1.1 and 5.1.2 require the owner to verify that the special lifting devices meet the performance criteria of the design specification by reviewing records and witness of testing.	There wasn't any design specification for these rigs and functional load testing was not originally required, or performed. *However, the Westinghouse Quality Release may be considered an acceptable alternate to verify that the criteria for letters of compliance for materials and specifications required by the Westinghouse drawings and purchasing documents was satisfied. *Although proof and functional testing was not required, the site assembly instructions require, after initial assembly on site, the following for the reactor vessel head lifting: Raising the rig, assembled to it's respective attachment, slightly above the supporting surface to be free hanging for one-half hour. Lowering the rig to its support and performing visual inspection and the appropriate nondestructive testing.

* These remarks are not applicable to the Reactor Coolant Pump Motor Lift Sling

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
	Section 5.1.3 requires periodic functional testing	Since maintenance and inspection procedures are written, if these procedures do not contain this requirement, then these procedures should be revised to include a visual check of critical welds and parts during lifting to comply with this requirement for functional testing.
	Section 5.1.4 requires operating procedure	Operating instructions for the reactor vessel internals lift rig were furnished to the utility and operating procedures were prepared and are used.
	Sections 5.1.5, 5.1.5.1 and 5.1.5.2 require special identification and marking to prevent misuse.	It is obvious from their designs that these rigs are special lifting devices and can only be used for their intended purpose. Specific identification of the rig can be made by marking, with stencils, the rig name and rated capacity, preferably on the spreader assembly.
	Sections 5.1.6, 5.1.7 and 5.1.8 require the owner to provide written documentation on the maintenance, repair, testing and use of these rigs.	Since operating instructions and maintenance instructions have been written by the owner, these should be reviewed to assure that they contain the requirements to address maintenance logs, repair and testing history, damage incidents etc.

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
5.2 and 5.3 5.2.1 to 5.2.3 and 5.3.1 to 5.3.8	<u>Acceptance Testing and Testing to Verify Continuing Compliance</u> - These paragraphs require the rigs to be initially tested at 150 percent maximum load followed by non-destructive testing of critical load bearing parts and welds and also annual 150 percent load tests or annual non-destructive tests and examinations; qualification of replacement parts.	The head lift rig was load tested at field assembly. It is suggested that a check of critical welds and parts be included in the maintenance procedures for all three devices. Preferably, since these devices have been in use at least once per year for over ten years, that a visual check during initial lift when possible should be acceptable. Further note that with the use of the load cell for the internals, lifting and lowering is monitored at all times. However, the load cell, which is used to monitor the internal lift rig loads at all times, cannot exceed the rated load by 20 percent without being inaccurate. This would preclude monitoring of a load test with the present equipment. Replacement parts should be in accordance with the original or equivalent requirements.

TABLE 2-1 (cont)

COMPARISON OF THE REQUIREMENT OF ANSI N14.6 AND

TURKEY POINT SPECIAL LIFT DEVICES

ANSI N14.6 Section	Description of ANSI N14.6 Requirement	Actual Special Lift Device Requirements
5.4 5.4.1 to 5.4.2	Maintenance and Repair - This section requires any maintenance and repair to be performed in accordance with original requirements and no repairs are permitted for bolts, studs and nuts.	Maintenance and repair procedure should contain, as much as possible, requirements that were used in the original fabrication. The critical items list of Appendix A contains the original type of non-destructive testing. The procedure should also define bolts, studs and nuts as non-repairable items.
5.5 5.5.1 to 5.5.2	Non-destructive Testing Procedures, Personnel Qualifications, and Acceptance Criteria - This section requires non-destructive testing to be performed in accordance with the requirements of the ASME Boiler and Pressure Vessel Code	Liquid penetrant, magnetic particle, ultrasonic and radiograph inspections were performed on identified items. These were in accordance with ASTM specifications, the ASME Code, and Westinghouse process specifications or as noted on detailed drawings and provide similar results to the requirement of the ASME Code.
6 6.1 6.2 6.3	Special Lifting Devices for Critical Loads - These sections contain special requirements for items handling critical loads.	It is assumed that compliance with NUREG 0612, Section 5.1 has been demonstrated and therefore this section is not applicable to these devices.

SECTION 3
DISCUSSION

The reactor vessel head and internals lift rigs, load cell, load cell linkage and reactor coolant pump motor lift sling generally meet the intent of the ANSI N14.6 requirements for design and manufacture. However, they are not in strict compliance with the ANSI N14.6 requirements for acceptance testing, maintenance and verification of continuing compliance.

Although no specific design specification was written, the assembly and detailed manufacturing drawings and purchase order documents contain equivalent requirements. A stress report has been prepared for these devices and the design criteria is considered satisfied. These devices, for the most part, were manufactured under Westinghouse surveillance with identified hold points, procedure review and personnel qualification which adequately meet these related ANSI requirements. Acceptance testing was not performed. However, an initial lift test was conducted for the head lift rig followed by the appropriate non-destructive testing following site assembly. Although the reactor coolant pump motor lift sling is called a sling, it is not a standard catalog item. Thus, it cannot be considered to be reviewed in accordance with the requirements of ANSI B30.9-1971.

It is anticipated that a 100 percent load test, performed on each device, followed by a visual check of critical welds would be sufficient to demonstrate compliance. This may require modification of existing Turkey Point operating and maintenance procedures.

SECTION 4
CONCLUSIONS

The review of the ANSI N14.6 requirements and comparison with the original Westinghouse requirements has shown that these items are generally in agreement for the design, fabrication and quality assurance of the lifting devices. However, the lifting devices are not in strict compliance with the ANSI N14.6 requirements for acceptance testing, maintenance and verification of continuing compliance. These specific requirements that are incompatible with the lifting devices are discussed in Appendix B with suggested actions. Westinghouse's objective was to provide a quality product and this product was designed, fabricated, assembled and inspected in accordance with internal Westinghouse requirements. In general, Westinghouse requirements meet the intent of ANSI N14.6 but not all the specific detailed requirements.

APPENDIX A

CRITICAL ITEMS LIST PER ANSI N14.6-1978

1. GENERAL

Section 3.1.2 of ANSI N14.6-1978 specifies that the design specification shall include a critical items list, which identifies critical components and defines their critical characteristics for material, fabrication, non-destructive testing and quality assurance.

"Critical items list" is further defined in ANSI N14.6, Section 2 as:

> "critical items list. A list that specifies the items of a special lifting device and their essential characteristics for which specified quality requirements shall apply in the design, fabrication, utilization, and maintenance of the device."

Load carrying members and welds of these special lifting devices are considered to be the critical items.

Tables A-1, A-2, A-3, A-4, and A-5 are the critical items list of parts and welds for the reactor vessel head lift rig, the reactor vessel internals lift rig, load cell and load cell linkage, and the reactor coolant pump motor lift sling respectively. These tables include the material identification, and the applicable volumetric and surface inspections that were performed in the fabrication of these special lifting devices. In some instances, non-destructive testing was not specified since the material selection and strength result in very low tensile stresses and thus, non-destructive testing was not justified.

The material selection for critical load path items was made to ASTM, ASME or special material requirements. However, the non-designed items of the reactor coolant pump motor lift sling were selected based on their load carrying capacities. The material requirements were supplemented by Westinghouse imposed non-destructive testing, and/or special heat treating requirements for almost all of the critical items. Westinghouse required all welding, welders, and weld procedures to be in accordance with ASME Boiler and Pressure Vessel Code Section IX or Westinghouse specifications. Westinghouse required certificates, or letters of compliance that the materials and processes used by the manufacturer were in accordance with the purchase order and drawing requirements. Westinghouse also performed final inspection on these devices and issued quality releases for the internals and head lifting rigs.

TABLE A-1

REACTOR VESSEL HEAD LIFT RIG

CRITICAL ITEMS LIST OF PARTS

PER ANSI N14.6-1978

Item[a]	Description	Material	Non-destructive Testing	
			Material	Finished
1,7,10,13 14	Pins	ASTM A434 Class BD	Ultrasonic	Magnetic Particle
2	Clevis Plate	ASTM A515 Grade 70	Ultrasonic Magnetic Particle	
3	Leg	ASTM A36		
4	Ring Girder	ASTM A285 Grade C		
5	Support Lug	ASTM A515 Grade 70	Ultrasonic Magnetic Particle	
6,9	Clevis	ASTM A237 Class A	Ultrasonic	Magnetic Particle
8	Arm	ASTM A306 Grade 70	Ultrasonic	Magnetic Particle
11,12	Sling Assembly Link and Lug	ASTM A105 Class 2	Ultrasonic	Magnetic Particle
15	Lifting Plate	ASTM A514	Ultrasonic	Magnetic Particle

[a] See figure A-1

TABLE A-2

REACTOR VESSEL HEAD LIFT RIG

CRITICAL ITEMS LIST OF WELDS

PER ANSI N14.6-1978

Item[a]	Weld Description	Non-destructive Testing	
		Root Pass	Final
2,3	Clevis Plate to Leg (fillet)	Visual	Magnetic Particle
4,5	Ring Girder to Support Lug (fillet)		Magnetic Particle
11,12	Link Lugs to Link (full penetration)		Radiograph Magnetic Particle

[a] See figure A-1.

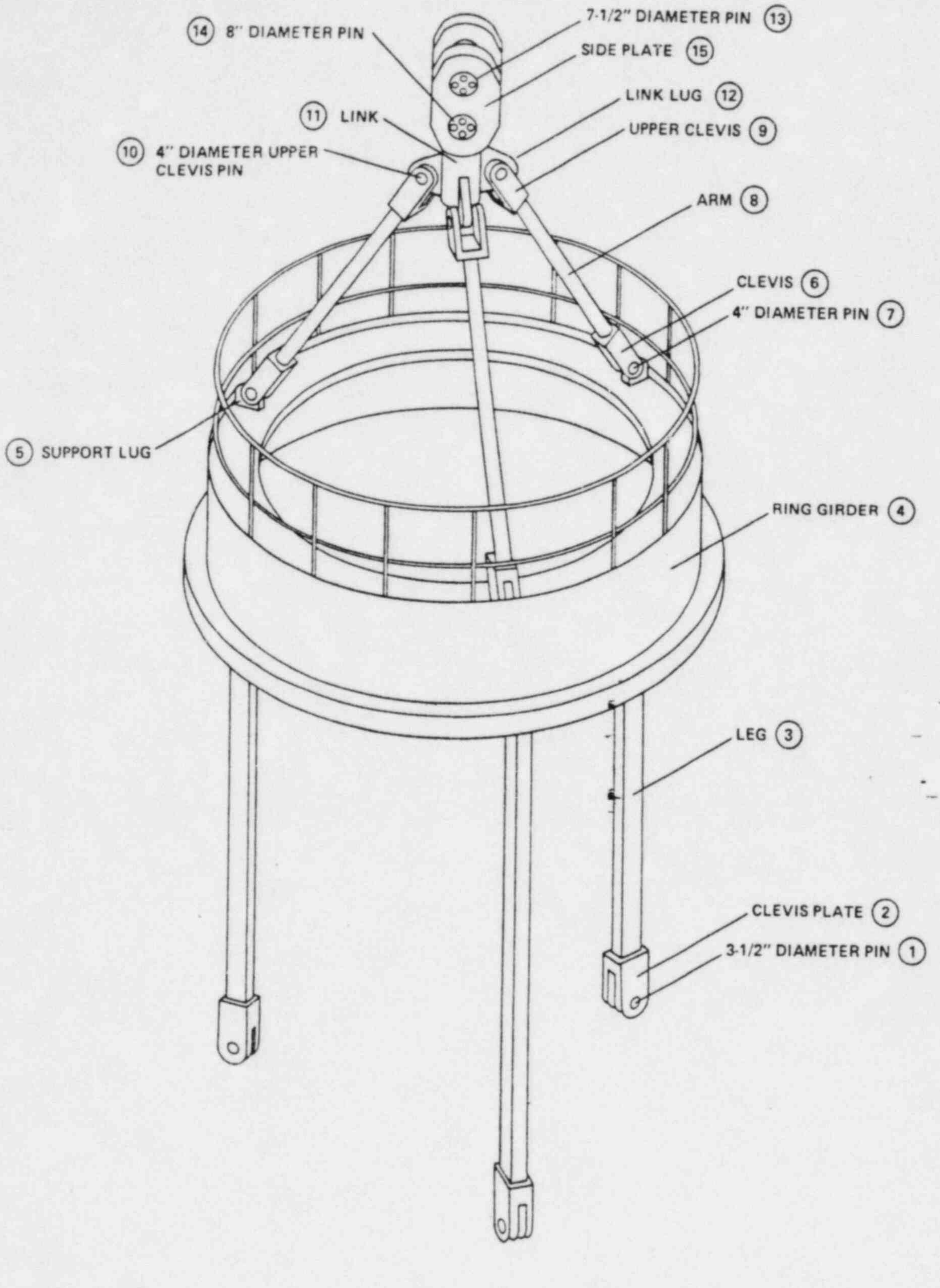

Figure A-1. Reactor Vessel Head Lift Rig

TABLE A-3

REACTOR VESSEL INTERNALS LIFT RIG,

LOAD CELL AND LOAD CELL LINKAGE

CRITICAL ITEMS LIST OF PARTS

PER ANSI N14.6-1978

Item[a]	Description	Material	Non-destructive Testing	
			Material	Finished
1,	Hook Pin	ASTM A434 Class BD	Ultrasonic	Magnetic Particle
3	Upper Adapter Pin			
7	Removable Pin			
2	Side Plate	ASTM A515 Grade 70 or ASTM A516 Grade 70	Magnetic Particle	
8	Top Lugs			
12	Sling Leg			
10	Side Lugs			
4	Upper Adapter	ASTM A540 Grade B24	Ultrasonic	Magnetic Particle or Liquid Penetrant
6	Lower Adapter			
5	Tension Cell	17-4 pH H-1100	Ultrasonic	Liquid Penetrant
9	Block	SA-105 Class 1 or 2 or SA-266 Class 1 or 2 or SA-508 Class 1 or 2	Ultrasonic	Magnetic Particle

(a) See figure A-2

TABLE A-3 (Cont)

REACTOR VESSEL INTERNALS LIFT RIG,

LOAD CELL AND LOAD CELL LINKAGE

CRITICAL ITEMS LIST OF PARTS

PER ANSI N14.6-1978

Item[a]	Description	Material	Non-destructive Testing	
			Material	Finished
11	Upper Sling Leg Pin	ASTM A276 Type 304, Center Ground Condition A		
13	Lower Sling Leg Pin			
14A	Spacer Block	SA-105 Class 1 or 2 or SA-266 Class 1 or 2 or SA-508 Class 1 or 2	Ultrasonic	Magnetic Particle
14B,C,D	Spreader Assembly	ASTM A515 Grade 70 or ASTM A516 Grade 70		
15	Nut	ASTM A276 Type 304 HR and PKLD, Condition A		
18	Leg Leveling Sleeve	ASTM A276 Type 304 HR and PKLD, Condition A		
20	Supporting Ring Leveling Sleeve	ASTM A276 Type 304 HR and PKLD, Condition A		
22	Guide Sleeve	ASTM A276 Type 304 HR and PKLD, Condition A		

(a)See figure A-2

TABLE A-3 (Cont)

REACTOR VESSEL INTERNALS LIFT RIG,

LOAD CELL AND LOAD CELL LINKAGE

CRITICAL ITEMS LIST OF PARTS

PER ANSI N14.6-1978

Item[a]	Description	Material	Non-destructive Testing	
			Material	Finished
16	Leg Adapter	ASTM A276 Type 304 HR and PKLD, Condition A		
23	Engaging Screw	ASTM A276 Type 304 HR and PKLD, Condition A		
19	Coupling	ASTM A312 Type 304		
21	Support Ring Outer Tube	ASTM A312 Type 304 Seamless, Cold Finish, HT TR		
17	Leg Outer Tube	ASTM A312 Type 304 Seamless, Cold Finish, HT TR		

[a]See figure A-2

TABLE A-4

REACTOR VESSEL INTERNALS LIFT RIG,
LOAD CELL AND LOAD CELL LINKAGE
CRITICAL ITEMS LIST OF WELDS
PER ANSI N14.6-1978

Item[a]	Weld Description	Non-destructive Testing	
		Root Pass	Final
8,9	Top Lugs to Sling Block (full penetration)	Magnetic Particle	Magnetic Particle
9,10	Side Lugs to Sling Block (full penetraton)	Magnetic Particle	Magnetic Particle
14	Spreader Assembly (fillet)	Visual	Magnetic Particle
16,17	Leg Outer Tube to Leg Adapter (full penetration)	Liquid Penetrant	Liquid Penetrant
17,18	Leg Outer Tube to Leg Leveling Sleeve (full penetration)	Liquid Penetrant	Liquid Penetrant
21,22	Support Ring Outer Tube to Guide Sleeve (full penetration)	Liquid Penetrant	Liquid Penetrant

[a] See figure A-2.

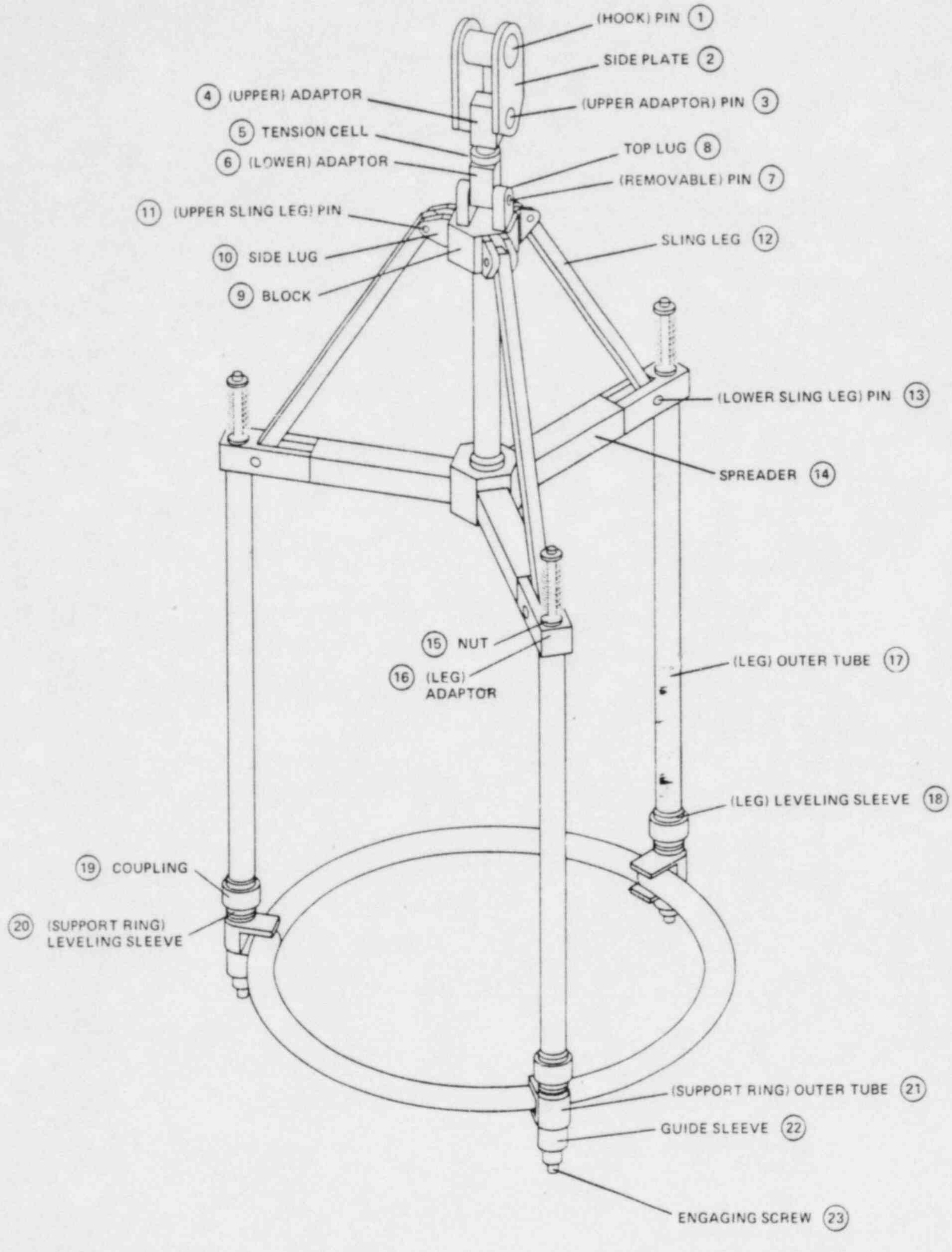

Figure A-2. Reactor Vessel Internals Lift Rig

TABLE A-5

REACTOR COOLANT PUMP MOTOR LIFT RIG

CRITICAL ITEMS LIST OF PARTS AND WELDS

PER ANSI N14.6-1978

Item[a]	Description	Material	Non-destructive Testing	
			Material	Finished
1	Spreader	ASTM A106 Grade B		Magnetic Particle on Welds Only
2	Side Plate	ASTM A106 Grade B		
5	Master Link	Alloy Steel Forging	Radiograph	
6	Sling Assembly	Improved Plow Steel Grade		
7	Shackle	Alloy Steel Forging		
8	Turnbuckle	Alloy Steel Forging		
9	Eye Hook	Alloy Steel Forging		

(a) See figure A-3

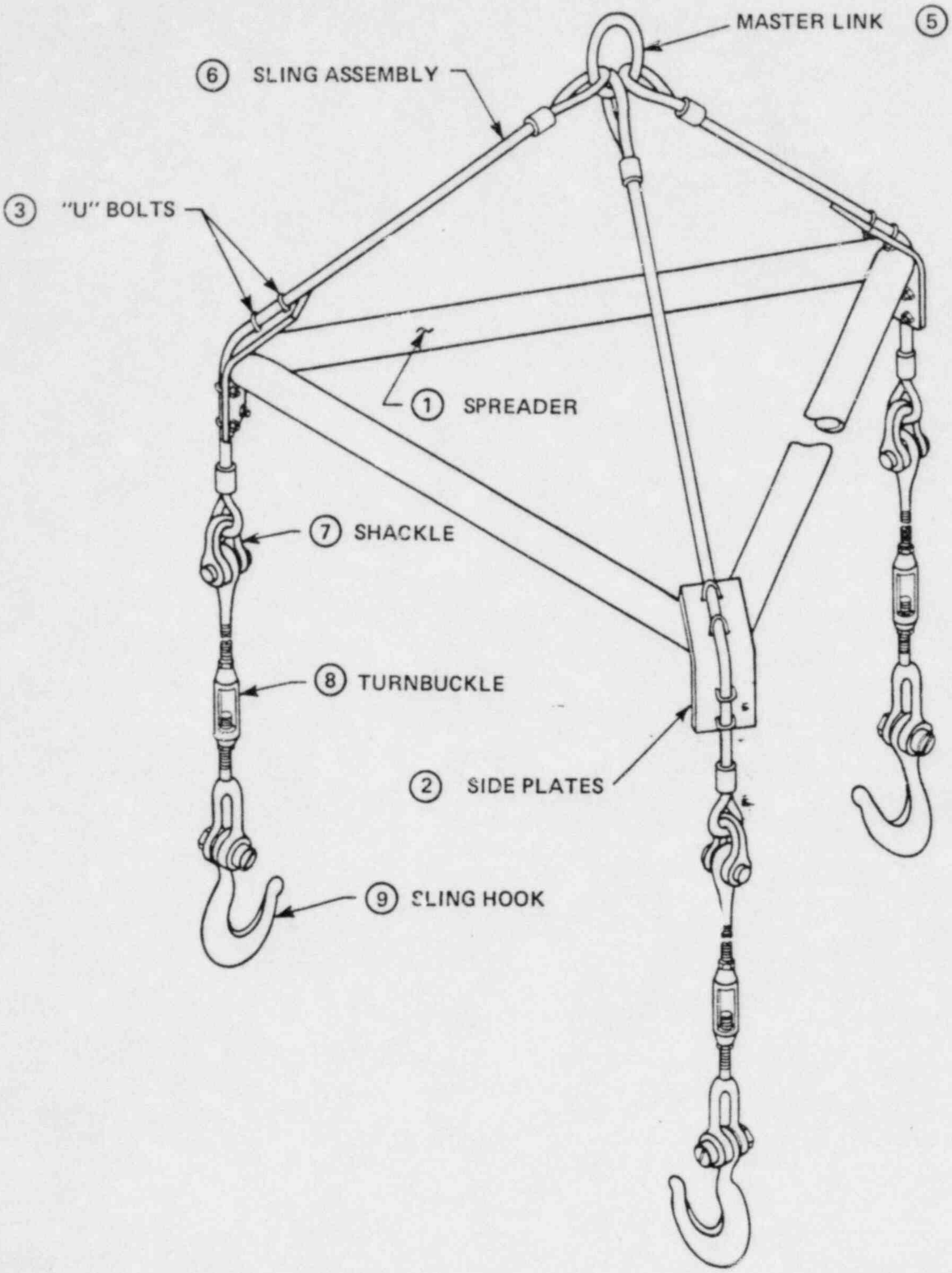

Figure A-3. Reactor Coolant Pump Motor Lift Sling

APPENDIX B

TABULATION OF ANSI N14.6-1978 REQUIREMENTS INCOMPATIBLE WITH THE TURKEY POINT LIFTING DEVICES

1. GENERAL

The comparison of the various ANSI N14.6 requirements and those of these lifting devices has shown that these devices are not in strict compliance with all the ANSI N14.6 requirements. Listed below is a tabulation of those sections of ANSI N14.6 considered most important in demonstrating the continued load handling reliability of these special lifting devices. Associated Westinghouse remarks are also listed and could be used as suggested actions and/or responses to demonstrate compliance to the NRC.

1a. Requirement:
Para. 3.1.4 - requires the designer to indicate permissible repair procedures and acceptance criteria for the repair.

1b. Remarks:
Any repair to these special lifting devices is considered to be in the form of welding. Should pins, bolts or other fasteners need repair, they should be replaced, in lieu of repair, in accordance with the original or equivalent requirements for material and non-destructive testing. Weld repairs should be performed in accordance with the requirements identified in NF-4000 and NF-5000 (Fabrication and Examaination) of the ASME Boiler and Pressure Vessel Code, Section III, Division 1 Subsection NF.

2a. Requirement:
Para. 3.2.1.1 - requires the design, when using materials with yield strengths above 80 percent of their ultimate strengths,

to be based on the material's fracture toughness and not the listed design factors.

2b. Remarks:

High strength materials are used in these devices. Although the fracture toughness was not determined, the material was selected based on it's fracture toughness characteristics. However, in lieu of a different stress design factor, the stress design factors listed in 3.2.1 of 3 and 5 were used in the analysis and the resulting stresses are considered acceptable.

3a. Requirement:

Para. 3.2.6 requires material for load-bearing members to be subjected to drop-weight or Charpy impact tests.

3b. Remarks:

Fracture toughness requirements were not identified for the material used in these special lifting devices. However, the material selection was based on its fracture toughness characteristics.

4a. Requirement:

Para. 3.3.6 requires an indication that an actuating mechanism is engaged.

4b. Remarks:

The reactor vessel internals lift rig employs a long handled tool to engage the rig and the internals. The tool depresses a spring loaded tube and turns the engaging screw into the internals. Although no specific position indication is identified, the visual difference in the top of the spring loaded tube is considered sufficient indication that the internals are engaged.

5a. Requirement:

Para. 4.1.6 requires a formal quality assurance program for the manufacturer and para 4.1.7 requires certification and identification of materials.

5b. Remarks:

A formal quality assurance program for the manufacturer was not required. However, the manufacturers welding procedures and non-destructive testing procedures were reviewed by Westinghouse prior to use. All critical load carrying members require letters of compliance for material requirements. Westinghouse performed certain checks and inspections during various steps of manufacturing. Final Westinghouse review includes visual, dimensional, procedural, cleanliness, personnel qualification, etc. and issuance of a quality release to ensure conformance with drawing requirements. No information that a quality release was issued for the reactor coolant pump motor lift sling has been found, although Westinghouse performed the final inspection.

6a. Requirement:

Para. 5.1 lists <u>Owner Responsibilities</u> and 5.1.2 requires the owner to verify that the special lifting devices meet the performance criteria of the design specification by records and witness of testing.

6b. Remarks:

There wasn't any design specification for these rigs and load testing was not originally required or performed. However, the Westinghouse Quality Release may be considered an acceptable alternate to verify that the criteria for letters of compliance for materials and specifications required by the Westinghouse drawings and purchasing document were satisfied. Although

proof and functional testing was not required, the site assembly instructions require, after initial assembly on site, the following for the reactor vessel head lifting:
Raising the rig, assembled to it's respective attachment, slightly above the supporting surface to be free handing for one-half hour. Lowering the rig to its support and performing visual inspection and the appropriate nondestructive testing. No other checks of welds and/or dimensions after assembly was required of the other special lifting devices.

7a. Requirement:
Para. 5.1.3 requires periodic functional testing and a system to indicate continued reliable performance.

7b. Remarks:
Since maintenance and inspection procedures are available if these procedures do not contain this requirement, then these procedures should be revised to include a visual check of critical welds and parts when possible during lifting to comply with this requirement for functional testing.

8a. Requirement:
Para. 5.1.5, 5.1.5.1 and 5.1.5.2 require special identification and marking to prevent misuse.

8b. Remarks:
It is obvious, from their designs, that these rigs are specific lifting devices and can only be used for their intended purpose and parts are not interchangeable. Specific identification of the rig can be made by marking with stencils, the rig name and rated capacity, preferably on the spreader assembly.

9a. Requirement:

Para. 5.1.6, 5.1.7 and 5.1.8 require the owner to provide written documentation on the maintenance, repair, testing and use of these rigs.

9b. Remarks:

Since operating instructions and maintenance instructions have been written by the owner, these should be reviewed to assure that they contain the requirements to address maintenance logs, repair and testing history, damage incidents and other items mentioned in these paragraphs.

10a. Requirement:

Para 5.2.1 requires the rigs to be initially tested at 150 percent maximum load followed by non-destructive testing of critical load bearing parts and welds.

10b. Remarks

The head lift rig was load tested and inspected at assembly to approximately 100 percent of the load. The internals lift rig and the R.C. pump motor lift sling were not required to be load tested.

11a. Requirement:

Para 5.2.2 requires replacement parts to be individually qualified and tested.

11b. Remarks

Replacement parts, should they be required, should be made of identical (or equivalent) material and inspections as originally required. Only pins, bolt and nuts are considered replacement parts for the reactor vessel head and internal lift rigs. Some of the items comprising the reactor coolant pump motor lift sling are catalog items and should they need replacement should be as identified in table B-1.

12a. Requirement:

Para 5.3 requires testing to verify continuing compliance and annual 150 percent load tests or annual non-destructive tests and examinations to be performed.

12b. Remarks

These special lifting devices are used during plant refueling which is approximately once per year. During plant operation these special lifting devices are inaccessable since they are permanently installed and/or remain in the containment. They cannot be removed from the containment unless they are disassembled and no known purposes exist for disassembly. Load testing to 150 percent of the total weight before each use would require special fixtures and is impractical to perform. Crane capacity could also be limiting. It is suggested that a check (visual) of critical welds and parts be conducted at initial lift when possible prior to moving to full lift and movement for all three devices. Preferably, since these devices have been in use at least once per year for over ten years, that a visual check during initial lift would be acceptable. Further note that with the use of the load cell for the internals lift rig, all lifting and lowering is monitored at all times.

2. SUMMARY

The requirements for periodic checking and functional load testing appear to be the ANSI N14.6 requirements that are most difficult to demonstrate compliance. It is almost impractical to perform the 150 percent load test prior to each use. It is suggested that the proposal to the NRC include a 100 percent load test, similar to the original check-out test to be performed with a minimum of nondestructive testing, (visual-only) in the critical parts and welds.

TABLE B-1

BILL OF MATERIALS FOR THE NON-DESIGNED ITEMS OF THE REACTOR COOLANT PUMP MOTOR LIFT SLING

No.(1)	Item	Description		Rated(6) Load Value (Pounds)
		Drawing(2)	Catalog(7)	
3	"U" Bolts	7/8" "U" Bolts + Cable Saddle	(5) "U" Bolts - G-450 1-1/2" Dia. Wire 7/8" Dia. "U" Bolt Wire Rope Clips	-----
5	Alloy Oblong Link	2-3/4" x 9" x 18"	(3) 2-3/4" x 9" x 16"	211,500
6	Sling Assembly	1-1/2" x 11'4" (Total inc. turn-buckle, shackle, etc.) 6 x 37 improved plow I.W.R.C. 3 Bridle Sling (7'6" sling length only incl. eye)	----(4) 1-1/2" x 11'4" 6 x 37 Improved Plowsteel I.W.R.C. 3 Bridle Sling with Mechanical Connections	81,000(2)
7	Shackle	Anchor Safety Shackle	(5) 2" Size G-213 or S-213	70,000
8	Turnbuckle	2" x 6" Jaw & Eye Turnbuckle	(5) G-227 Jaw & Eye Turnbuckle 2" Dia. thd., 24" take-up Class 8	37,000
9	Sling Hook	TAYCO A-73	(3) Sling Hook No. A-73 1" Dia. Chain with 2-1/2" Dia. Eye	38,750

NOTES:

(1) See figure A-3 for identification of item numbers

(2) Description is from Westinghouse drawing AED SK 618J644 TXK SUB 5

(3) Taylor Chain Co., Inc., Alloy Steel Chain Assemblies, Attachments, Revised 4/80

(4) Pennsylvania Sling Co.

(5) Crosby Group, 950 General Catalog, June 1981.

(6) Rated load value: The maximum recommended load that should be exerted on the item. The following terms are also used for the term Rated Load: "SWL", "Safe Working Load", "Working Load Limit", and the "Resultant Safe Working Load". All rated load values are for in-line pull with respect to the centerline of the item. Information is from catalogs identified in (3), (4), (5).

(7) Ordering Information

Sling assembly per Westinghouse Drawing AED SK 618J644 TXK, SUB 5 composed of the items in table B-1 including the following requirements:

a. The safe working load of this sling assembly is 81,000 lb. and a safety factor of 5:1".

b. Perform lift test at assembly of 121,500 pounds (61 tons).

c. The master link is to be magnetic particle inspected after load test to the requirements of ASME Boiler & Pressure Vessel Code. Section V Article 7. Acceptance standards are to be to ASME B&PV Code Section III Subsection NF 5341. Radiograph inspection may be substituted for magnetic particle inspection to ASME B&PV Code Section V, Article 2 with acceptance standard to ASME B&PV Code Subsection NF 5320.

d. A certification is required for load testing, non-destructive testing and material used in this assembly.

ATTACHMENT B to
WCAP-10168

STRESS REPORT
REACTOR VESSEL HEAD LIFT RIG
REACTOR VESSEL INTERNALS LIFT RIG,
LOAD CELL, LOAD CELL LINKAGE
AND REACTOR COOLANT PUMP
MOTOR LIFT SLING

FOR

FLORIDA POWER AND LIGHT COMPANY
TURKEY POINT UNITS 3 AND 4

December, 1982

H. H. Sandner, P.E.

Approved: [signature]
R. J. Leduc, P. E., Manager
Component Handling Equipment

ABSTRACT

A stress analysis of the Turkey Point Units 3 and 4 reactor vessel head and internal lift rigs, load cell, load cell linkage and reactor coolant pump motor lift sling was performed to determine the acceptability of these devices to meet the design requirements of ANSI N14.6.

ACKNOWLEDGMENT

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M. F. Hankinson, P.E.

F. C. Peduzzi

J. Urban

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SECTION 1

INTRODUCTION

The Nuclear Regulatory Commission (NRC) issued NUREG 0612 "Control of Heavy Load at Nuclear Power Plants"[1] in 1980 to address the control of heavy loads to prevent and mitigate the consequences of postulated accidental load drops. NUREG 0612 imposes various training, design, inspection and procedural requirements for assuring safe and reliable operation for the handling of heavy loads. In the containment building, NUREG 0612 requires special lifting devices to meet the requirements of ANSI N14.6-1978 "American National Standard for Special Lifting Devices for Shipping Containers Weighing 10,000 Pounds or More for Nuclear Materials".[2] In general, ANSI N14.6 contains detailed requirements for the design, fabrication, testing, maintenance and quality assurance of special lifting devices. In addition, ANSI N14.6 requires that when wire rope or chain is used in the design of a lifting device, the wire rope or chain shall be in conformance with ANSI B30.9-1971 "American National Standard Safety Standard for Slings".[3] The NRC has requested operating plants to demonstrate compliance with these requirements.

This report contains the stress analysis performed on the Turkey Point reactor vessel headlift rig, reactor vessel internals lift rig, load cell, load cell linkage and reactor coolant pump motor lift sling to determine the acceptability of these devices to meet these requirements.

1.1 BACKGROUND

The reactor vessel head lift rig, the internals lifting rig, load cell, load cell linkage, and reactor coolant pump motor lift sling were designed and built for the Turkey Point Units 3 and 4, circa 1967-68. The actual design criteria is unknown for the lifting devices. It appears that for the head lift rig, internals lift rig, load cell, cell linkage and the reactor coolant pump motor lift that in most cases the design criteria used was that the resulting stress in the load members, when subjected to the total combined lifting weight, does not exceed one fifth (1/5) of the ultimate strength of the material. These items were not classified as nuclear safety components and thus requirements for formal documentation of design requirements and stress reports were not applicable. Thus, stress reports and design specifications were not formally documented. Westinghouse defined the design, fabrication and quality assurance requirements on detailed manufacturing drawings and purchase order documents. Westinghouse also issued field assembly and operating instructions, where applicable.

SECTION 2
COMPONENT DESCRIPTION

2.1 REACTOR VESSEL HEAD LIFT RIG

The reactor vessel head lift rig[4] is a three legged carbon steel structure, approximately 35 feet high and 14 feet in diameter, weighing approximately 25,000 pounds. It is used to handle the assembled reactor vessel head.

The three vertical legs and control rod drive mechanism (CRDM) platform assembly are permanently attached to the reactor vessel head lifting lugs. The tripod sling assembly is attached to the three vertical legs and is used when installing and removing the reactor vessel head. During plant operations, the sling assembly is removed and the three vertical legs and platform assembly remain attached to the reactor vessel head.

2.2 REACTOR VESSEL INTERNALS LIFT RIG

The reactor vessel internals lift rig[5] is a three-legged carbon and stainless steel structure, approximately 29 feet high and 12 feet in diameter weighing approximately 13,000 pounds. It is used to handle the upper and lower reactor vessel internals packages. It is attached to the main crane hook for all lifting, lowering and traversing operations. A load cell linkage is connected between the main crane hook and the rig to monitor loads during all operations. When not in use, the rig is stored on the upper internals storage stand.

The rig attaches to the internals packages by means of three engaging screws which are screwed into tapped holes in the internals flanges. These screws are manually operated from the spreader assembly using the internal tools. The screws are normally spring retracted upward and are depressed to engage the tapped holes.

2.3 LOAD CELL AND LOAD CELL LINKAGE

The load cell is used to monitor the load during lifting and lowering the reactor vessel internals to ensure no excessive loadings are occurring. It is installed between the load cell linkage and the lifting device. The load cell is a strain gage (tension) type, rated at 300,000 pounds, built by W. C. Dillon and Co. The load cell linkage is an assembly of pins, plates and bolts which connect the polar crane main hook to the load cell.

2.4 REACTOR COOLANT PUMP MOTOR LIFT SLING

The reactor coolant pump motor lift sling[6] consists of a six (6) foot triangular carbon steel spreader assembly weighing approximately one thousand (1000) pounds. It has wire rope slings, shackles and turnbuckles attached which form a three point lift assembly. It is used to handle the reactor coolant pump motor.

SECTION 3

DESIGN BASIS

3.1 DESIGN CRITERIA

NUREG 0612, paragraph 5.1.1(4) states that special lifting devices should satisfy the guidelines of ANSI N14.6. Further, NUREG 0612, 5.1.1(4) states: "In addition, the stress design factor stated in Section 3.2.1.1 of ANSI N14.6 should be based on the combined maximum static and dynamic loads that could be imparted on the handling device based on characteristics of the crane which will be used. This is in lieu of the guideline in Section 3.2.1.1 of ANSI N14.6 which bases the stress design factor on only the weight (static load) of the load and of the intervening components of the special handling device".

It can be inferred from this paragraph that the stress design factors specified in Section 3.2.1.1 of ANSI N14.6 (3 and 5) are not all inclusive. Also, it can be inferred that the specified ANSI N14.6 stress design factors should be increased by an amount based on the crane dynamic characteristics. The dynamic characteristics of the crane would be based on the main hook and associated wire ropes holding the hook. Most main containment cranes use sixteen (16) or more wire ropes to handle the load. Should the crane hook suddenly stop during the lifting or lowering of a load, a shock load could be transmitted to the connected device. Because of the elasticity of the sixteen or more wire ropes, the dynamic factor for a typical containment crane is not much larger than 1.0. The maximum design factor that is recommended by most design texts[7,8,9] is a factor of 2 for loads that are suddenly applied. The stress design factors required in Section 3.2.1.1 of ANSI N14.6 are:

$3 \text{ (weight)} < \text{Yield Strength}$

$5 \text{ (weight)} < \text{Ultimate Strength}$

The factor of 3 specified, certainly, includes consideration of suddenly applied loads for cases where the dynamic impact factor may be as high as 2.0. Thus, we feel that the use of the design criteria in ANSI N14.6 satisfies the NUREG requirement.

To provide flexibility on stress design factor, the summary tables list the stresses with stress design factors of 1, 3 and 5. Thus, any stress design factor may be easily applied to satisy any concerns.

3.2 DESIGN WEIGHTS

The following design weights were used in the analysis of the lifting devices:

3.2.1 Reactor Vessel Head Lift Rig

(A) Design Weight for Lower Clevis and Pin (Items 14 and 15) is 253,000 pounds

(B) Design Weight for rest of Lift Rig is 278,000 pounds

3.2.2 Reactor Vessel Internals Lift Rig, Load Cell, and Load Cell Linkage

The design weight is 260,000 pounds which is the total weight of the lifting device and the lower internals.

3.2.3 Reactor Coolant Pump Motor Lift Sling

The design weight is 81,000 pounds which is larger than the total weight of the lifting device and the reactor coolant pump motor. It is the safe working load identified on the assembly drawing.

SECTION 4
MATERIALS

4.1 MATERIAL DESCRIPTION

The materials and material properties for the reactor vessel head lift rig, the reactor vessel internals lift rig, load cell, load cell linkage and reactor coolant pump motor lift sling are listed in Tables 4-1, 4-2 and 4-3.

TABLE 4-1

REACTOR COOLANT HEAD LIFT RIG MATERIAL AND MATERIAL PROPERTIES

Item(a)	Description	Materials	Yield Strength S_y (ksi)	Ultimate Strength S_{ult} (ksi)
1 7 10 13 14	3 1/2" Diameter Pin 4" Dia. Bottom Clevis Pin 4" Dia. Upper Clevis Pin 7 1/2" Dia. Pin 8" Dia. Pin	ASTM A434 Class BD	110 110 105 100 100	140 140 135 130 130
2,5,15	Side Plate Support Lug	ASTM A515 or GR70	38	70
3	Leg	ASTM A36	36	58
4	Ring Girder	ASTM A285 GR. C	30	55
6,9	Upper Clevis Clevis	ASTM A237 CL A	50	80
8	Arm	ASTM A306 Gr. 70	35	70
11, 12	Clevis Plate Link Link Lug	ASTM A 105 CL 2	36	70

(a)See figure 5-1

TABLE 4-2

REACTOR VESSEL INTERNALS LIFT RIG MATERIAL AND MATERIAL PROPERTIES

Item[a]	Description	Materials	Yield Strength S_y (ksi)	Ultimate Strength S_{ult} (ksi)
1	Hook Pin	ASTM A434 CL BD	100	130
3	Upper Adaptor Pin	ASTM A434 CL BD	105	135
7	Removable Pin			
11	Upper Sling Leg Pin	ASTM A 276 Type 304 Center Ground Cond. A	30	75
13	Lower Sling Leg Pin			
2	Side Plate	ASTM A-515 Gr. 70 or ASTM A-516 Gr. 70	38 38	70 70
8	Top Lugs			
10	Side Leg			
12	Sling Leg			
14,B,C,D	Spreader Assembly			
4	Upper Adaptor	ASTM A-540 Gr. B-24	120	135
6	Lower Adaptor			
5	Tension Cell	17-4 pH H1100	115	140

(a)See figure 5-2

TABLE 4-2 (Cont)

REACTOR VESSEL INTERNALS LIFT RIG MATERIAL AND MATERIAL PROPERTIES

Item[(a)]	Description	Materials	Yield Strength S_y (ksi)	Ultimate Strength S_{ult} (ksi)
9	Block	SA-105 CL 1 or 2 or SA-266 CL 1 or 2 or SA-508 CL 1 or 2	36 30 35	70 60 70
14A	Spacer Block			
15	NUT	ASTM A-276 Type 304 HR+PKLD, Cond. A	38	82
18	Leg Leveling Sleeve			
20	Support Ring Leveling Sleeve			
22	Guide Sleever			
16	Leg Adaptor	ASTM A276 Type 304 HR+PKLD, Cond. A	43.9	86.5
23	Engaging Screen	ASTM A276 HR+PKLD, Cond. A	36.8	80.9
19	Coupling	ASTM A312, Type 304	30	75
17	Leg Outer Tube	ASTM A312 Type 304 Seamless, Cold Fanish and Heat Treat	38.0	83.4
21	Support Ring Outer Tube			

(a) See figure 5-2

TABLE 4-3

REACTOR COOLANT PUMP MOTOR LIFT SLING MATERIALS AND MATERIAL PROPERTIES

Item[a]	Description	Materials	Yield Strength S_y (ksi)	Ultimate Strength S_{ult} (ksi)
1	Spreader (pipe)	ASTM A106 Grade B	35	60
2	Side plates	ASTM 212 Grade A	36	58

[a] See figure 5-3

SECTION 5
SUMMARY OF RESULTS

Tables 5-1, 5-2, and 5-3 summarize the stresses on each of the parts which make up the reactor vessel head and internals lift rig, load cell, load cell linkage and reactor coolant pump motor lift sling, respectively. All of the tensile and shear stresses, with the exception of the tensile stress at the minimum section of the internals lifting rig engaging screw, item (23), meet the design criteria of section 3.2.1.1 of ANSI N14.6, requiring application of stress design factors of three and five with the accompanying allowable stress limits of yield and ultimate strength, respectively. Application of the ANSI N14.6 criteria to pins subject to bending, structural members subject to buckling and bending loads, and various parts subject to bearing loads result in some stresses exceeding this criteria. However, when using more appropriate criteria, the resulting stresses are acceptable.

5.1 DISCUSSION OF RESULTS

5.1.1 Application of ANSI N14.6 Criteria

The design criteria of section 3.2.1.1 of ANSI N14.6, requiring application of stress design factors of three and five with the accompanying allowable stresses, are to be used for evaluating load bearing members of a special lifting device when subjected to loading conditions resulting in shear or tensile stresses. Application of these design load factors to other loading conditions is not addressed in ANSI N14.6. However, these two stress design factors have been used to determine the stresses of the load carrying members when subject to other loading conditions, viz. bearing, bending, buckling. This is an extremely conservative approach and in some cases the resulting stresses exceed the accompanying allowable stress limit.

In the internals lift rig engaging screw (item 23) the tensile stress slightly exceeds the criteria that three times the calculated stress must be less than the yield stress. However, the conservative criteria that five times the calculated stress must be less than the ultimate is met and thus the engaging screw is considered acceptable.

For items 11, 13, 15, 22 and 23 of the internals lifting rig, the bearing stress exceed the criteria that three times the calculated stress must be less than the yield stress. However, all the bearing stresses meet the conservative criteria that five times the calculated stress must be less than the ultimate stress and thus the parts are considered acceptable for bearing.

The bending stresses in the internals lifting rig pins (items 3, 7, 11 and 13) were calculated using the conservative approach shown in the Machine Design Fastening and Joining Issue. This approach, coupled with the use of the ANSI 14.6 stress factor for this condition, results in bending stresses exceeding the allowables. However, the acceptance criterion for pin design is shear stresses and the results of all shear stress calculations are below acceptable limits. The bending stress calculations are included for reference.

The bending stress in the internals lifting rig spacer blocks (item 14-A) does not meet the criteria of section 3.2.1.1. However, this is a local fiber stress. Even if the fiber stress reached the yield stress, and it doesn't, the rest of the cross-section could assume the additional load. This localized stress can be considered under section 3.2.1.2 which states that the stress design factors of 3.2.1.1 are not intended to apply to situations where high load stresses are relieved by slight yielding. The shear stress in the block is extremely low and well within the section 3.2.1.1 criteria.

Structural elements loaded in compression and bending are analyzed by the empirical equations of the ASME and AISC rules. Buckling stresses are expressed as the ratios of actual stresses to the allowable stresses with the acceptable ratio being $<$ 1.0. These equations do not determine the limiting stresses of members in buckling but indicate whether or not the calculated stress is or is not within the allowable values. Instead, the ultimate load carrying capability of the member is the determining factor in the structural member's acceptability. Timoshenko[12] notes that the ultimate load for short struts is equal to the material yield point. Calculation of the ultimate load results in this load being 1.3 times the nominal design load and thus, these members are considered acceptable.

5.1.2 Pin Bending

Several of the internal lifting rig pins do not meet the ANSI N14.6 criteria (3 and 5) when analyzed for bending stresses. In calculating the bending stresses in pins, it is assumed that loads from the outside lugs are linerally distributed and from the inside eye are uniformly distributed along the pin. Further, the assumption is made that span to diameter ratios are large enough such that the assumptions inherent to simple beam theory are valid (Neither condition is met, however, by the actual joint geometry).

Pin deflections and local yielding of the pin, lugs, and eye cause the loads to be non-uniform and their centroids to shift towards the interfaces between the lugs and eye, i.e., the shear planes in the pins. This concentration of load near the shear planes reduces the effective bending arms at which loads are applied thereby reducing the bending moment.

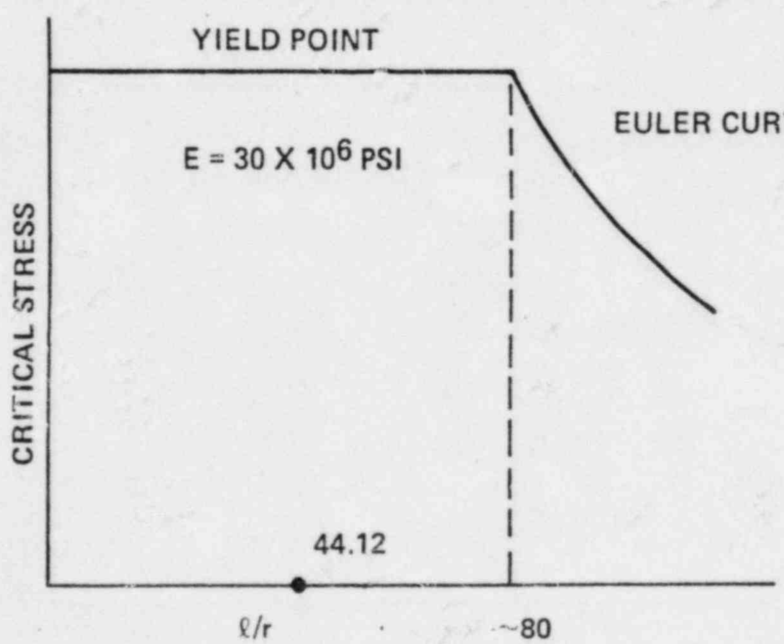
The calculated bending stresses are thus over estimated and are tabulated for reference. Shear stresses in the pins are the governing parameter for pin strength. Using shear stresses as the criteria for pin design results in all pins meeting three times the calculated stress being less than the yield stress and the ANSI criteria of 5 times the calculated stress being less than the ultimate stress. Thus all the above pins are considered acceptable.

5.1.3 Structures Loaded in Compression and Bending

The spreader assemblies of the reactor vessel internals lift rig and the reactor coolant pump motor lift sling do not meet all the ANSI N14.6 criteria (3W and 5W) when analyzed for combined axial compression and bending (buckling) or bending loadings. Structures loaded in combined compression and bending are analyzed by the empirical equations of the [10] ASME Boiler and Pressure Vessel Code Section III, Appendix XVII. (Same as the AISC [11] Part 5 rules). Ratios less than 1.0 indicate the calculated stress is within the allowable values; greater than 1.0, the stress is greater than the allowables.

When comparing the spreader arm to the above criteria, the acceptable ratio < 1.0 is not satisfied. However, these empirical equations do not identify the maximum stress for this loading condition. When the ratio is >1.0, these equations identify an unacceptable design condition. If we were designing a new structure, the material and member size would be changed to ensure these ratios would be satisfied for all loading conditions. However, these calculations are being applied to an existing structure and since these conditions are not satisfied then the ultimate load carrying capability must be determined.

The column under consideration is relatively short ($\frac{k\ell}{r} = 44.12$). Timoshenko[12] states that experiments show that short columns buckle when the compressive strength reaches the material yield point. (The horizontal line on the figure below).



COLUMN BUCKLING

Therefore the total stress

$$\sigma = \frac{P}{A} + \frac{M}{Z} = \text{Direct Stress + Bending Stress}$$

must be less than or equal to the material yield stress.

Since the column is loaded eccentrically (i.e., M is proportional to P, so the two could be equivalently replaced with a load P displaced from the column centerline by a distance e) as the column bends the effective lever arm of the load increases. For this condition

$$\sigma = \frac{P}{A} + \frac{Pe}{Z} \text{ secant } \sqrt{\frac{P}{EI}}\frac{\ell}{2}$$

For this particular column (internals lift rig spreader)

the expression $\left[\text{secant } \sqrt{\frac{P}{EI}}\frac{\ell}{2}\right]$

is essentially 1.0 for even 5 times the nominal load and the equation reverts to

$$\sigma = \frac{P}{A} + \frac{Pe}{Z} = \frac{P}{A} + \frac{M}{Z}$$

For the case of the internals lift rig spreader;

$\frac{P}{A}$ = 3290 psi and $\frac{M}{Z}$ = 25,135 psi

Then the total stress is:

σ_{total} = 3290 + 25,135 = 28,425 psi

which is less than the material yield strength (S_y)

Then to find the ultimate column load, let $\sigma_{max} = S_y = 38{,}000$ psi

Then the maximum column load is the ratio of

$$\sigma_{max}/\sigma_{total} = \frac{38{,}000}{28{,}425} = 1.337$$

Thus the ultimate column load is 1.3 times the nominal value.

The internals lift rig spreader members are considered acceptable for this condition of combined stresses for axial compression and bending.

Similarly, the ultimate load for the reactor coolant pump motor lift sling is 3.7 times the nominal value and also considered acceptable.

5.1.4 Rated Load Values of the Reactor Coolant Pump Motor Lift Sling

Since most of the components that comprise the reactor coolant pump motor lift sling are non-designed components application of the criteria of section 3.2.1.1 of ANSI N14.6 to these components are not appropriate. Therefore, Table 5-4 has been prepared from catalog information and list the various load conditions. Noting that the safety factors are based on the ratio of the ultimate strength of the material to the rated load value, the sling is acceptable for five times the design load.

5.2 CONCLUSION

Application of the ANSI N14.6 criteria of (3 and 5) to these special lifting devices results in acceptable stress limits for tensile and shear stresses with the exception of the internals lifting rig engaging screw. Application of this criteria to all structural members subject to other types of loadings tend to result in oversimplified conservatism and with some stresses exceeding the accompanying allowable limits. However, when using the more appropriate criteria for those cases not addressed by the ANSI N14.6 criteria the stresses are within the appropriate allowable limits.

TABLE 5-1

SUMMARY OF RESULTS

REACTOR VESSEL HEAD LIFT RIG

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value W[b]	Value 3W	Value 5W	Material Allowable (ksi) S_y[c]	Material Allowable (ksi) S_{ult}[d]
1	3-1/2" dia. pin ASTM A434 Class BD	Shear	4.8	14.4	24.0	110	140
		Bearing on Head Clevis	8.8	26.4	44.0		
		Bearing on Clevis Plate	8.3	24.9	41.5		
		Bending	17.0	51.0	85.0		
2	Clevis Plate ASTM A515 Grade 70 Q&T	Shear Tear-Out at Pin Hole	4.1	12.3	20.5	38	70
		Bearing on Pin	8.3	24.9	41.5		
		Tension at Pin Hole	4.1	12.3	20.5		
3	Leg ASTM A36	Fillet Weld on Clevis Plate to Leg - Shear	4.0	12.0	20.0	weld allowable = 18[e]	
		Tension	7.8	23.4	39.0	36	58
4	Ring Girder ASTM A285 GRC Firebox	Total Shear	2.4	7.2	12.0	30	55
		Combined Tensile	3.5	10.5	17.5		
		Weld of Ring Girder to Support Lug	2.4	7.2	12.0		

(a) See figure 5-1 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)
(e) Stress limit for fillet weld from ASME Boiler and Pressure Vessel Code, Section III Division 1 - Subsection NF 1980 Edition, Table NF-3292 1-1, page 43

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TABLE 5-1 (cont)
SUMMARY OF RESULTS
REACTOR VESSEL HEAD LIFT RIG

Item[a] No.	Part Name And Material	Calculated Stresses (ksi)				Material Allowable (ksi)	
		Designation	Value				
			W[b]	3W	5W	S_y[c]	S_{ult}[d]
5	Support Lug ASTM A515 GR70 Normalized and Tempered	Tension at Pin Hole	5.8	17.4	29.0	38	70
		Shear-Tear-Out at Pin Hole	5.0	15.0	25.0		
		Bearing on Pin	7.5	22.5	37.5		
6	Clevis ASTM A237 Class A	Thread Shear	3.4	10.2	17.0	50	80
		Bearing on Pin	6.7	20.1	33.5		
		Shear Tear-Out at Pin Hole	4.3	12.9	21.5		
		Tension at Pin Hole	4.3	12.9	21.5		
		Tension at Thd. Relief	2.1	6.3	10.5		
7	4" Dia Pin ASTM A434 Class BD Min. Tempering Temp. = 1100°F	Shear	4.1	12.3	20.5	105	135
		Bearing on Support Lug	7.5	22.5	37.5		
		Bearing on Lower Clevis	6.7	20.1	33.5		
		Bending	14.3	42.9	71.5		
8	Arm ASTM A306 GR 70	Thread Shear	3.4	10.2	17.0	35	70
		Thread Tension	9.2	27.6	46.0		

(a) See figure 5-1 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

TABLE 5-1 (cont)
SUMMARY OF RESULTS
REACTOR VESSEL HEAD LIFT RIG

Item[(a)] No.	Part Name And Material	Designation	Calculated Stresses (ksi) Value W[(b)]	3W	5W	Material Allowable (ksi) S_y[(c)]	S_{ult}[(d)]
9	Upper Clevis	Thread Shear Bearing on Pin Shear Tear-Out at Pin Hole Tension at Pin Hole Tension at Thd. Relief	3.4 6.7 4.3 4.3 2.1	10.2 20.1 12.9 12.9 6.3	17.0 33.5 21.5 21.5 10.5	50	80
10	4" Dia Upper Clevis Pin ASTM A434 Class BD Min. Tempering Temp. = 1100°F	Shear Bearing on Link Lug Bearing on Upper Clevis Bending	4.1 7.3 6.7 14.1	12.3 21.9 20.1 42.3	20.5 36.5 33.5 70.5	105	135
11	Link A105 Class 2	Bearing on Hook Pin Tension at Pin Hole Shear Tear-Out at Pin Hole Tension at Cylindrical Section	4.3 4.0 4.0 5.3	12.9 12.0 12.0 15.9	21.5 20.0 20.0 26.5	36	70

(a) See figure 5-1 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

TABLE 5-1 (cont)
SUMMARY OF RESULTS
REACTOR VESSEL HEAD LIFT RIG

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value $W^{(b)}$	3W	5W	Material Allowable (ksi) $S_y^{(c)}$	$S_{ult}^{(d)}$
12	Link Lug A105 Class 2	Tension at Pin Hole	4.9	14.7	24.5	36	70
		Shear Tear-Out at Pin Hole	4.9	14.7	24.5		
		Bearing at Pin Hole	7.3	21.9	36.5		
		Shear at Root of Lug	1.6	4.8	8.0		
		Combined Tension at Lug Root	3.7	11.1	18.5		
13	7-1/2" Dia. Pin ASTM A434 Class BD	Shear	3.2	9.6	16.0	100	130
		Bearing on Hook	4.6	13.8	23.0		
		Bearing on Side Plate	6.2	18.6	31.0		
		Bending	10.7	32.1	53.5		
14	8" Dia. Pin ASTM A434 Class BD	Shear	2.8	8.4	14.0	100	130
		Bearing on Link	4.3	12.9	21.5		
		Bearing on Side Plate	5.8	17.4	29.0		
		Bending	8.8	26.4	44.0		
15	Side Plate ASTM A514	Bearing at 7-1/2" Hole	6.2	18.6	31.0	90	100
		Tension at 8" Hole	5.5	16.5	27.5		
		Shear Tear-Out at 8" Hole	5.5	16.5	27.5		

(a) See figure 5-1 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

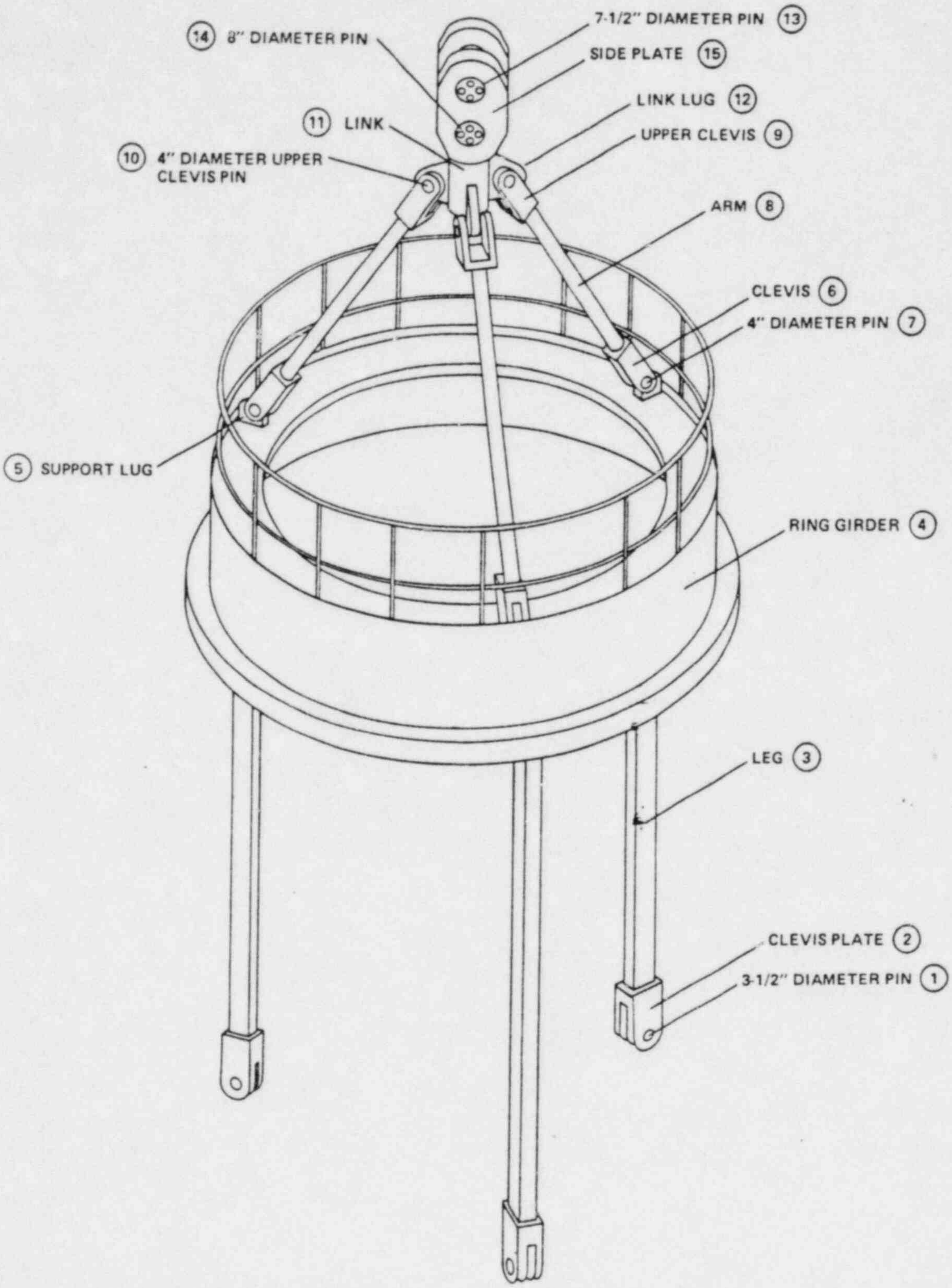

Figure 5-1. Reactor Vessel Head Lifting Rig

TABLE 5-2
SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Designation	Calculated Stresses (ksi) Value W[b]	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
1	(Hook) Pin ASTM A434 Class BD AISI 4340 Hot Rolled and Quenched & Tempered	Shear	3.0	9.0	15.0	100	130
		Bearing on Hook	4.4	13.2	22.0		
		Bearing on Side Plates	10.1	30.3	50.5		
		Bending	9.3	27.9	46.5		
2	Side Plate ASTM A515 GR 70 or ASTM A516 GR 70 Q&T	Tension at 7.515 Dia. Hole	10.1	30.3	50.5	ASTM A515	
		Bearing at 7.515 Dia. Hole	10.1	30.3	50.5	38	70
		Shear Tear-out at 7.515 Dia. Hole	10.1	30.3	50.5		
						ASTM A516	
		Tension at 4.385 Dia. Hole	9.2	27.6	46.0	38	70
		Shear Tear-out at 4.385 Dia. Hole	9.2	27.6	46.0		
		Bearing at 4.385 Dia. Hole	10.6	31.8	53.0		
3	(Upper Adaptor) Pin ASTM A434, Class BD, AISI 4340 Hot Rolled and Q&T	Shear	8.8	26.4	44.0	105	135
		Bearing on (Upper) Adaptor	9.6	28.8	48.0		
		Bearing on Side Plate	10.6	31.8	53.0		
		Bending	42.4	127.2	212.0		

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG,

LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value W[b]	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
4	(Upper) Adaptor ASTM A540 Grade B-24	Tension at 4.387 Dia. Hole Bearing at 4.387 Dia. Hole Tension at Thread Relief Thread Shear Shear Tear-out at 4.387 Dia. Hole	10.6 9.5 7.0 11.2 9.1	31.8 28.5 21.0 33.6 27.3	53.0 47.5 35.0 56.0 45.5	120	135
5	Tension Cell 17-4 pH ss H-1100°	Tension at Threads Thread Shear	19.4 11.2	58.2 33.6	97.0 56.0	115	140
6	(Lower) Adaptor	Tension at 4.387 Dia. Hole Bearing at 4.387 Dia. Hole Tension at Thread Relief Thread Shear Shear Tear-out at 4.387 Dia. Hole	10.6 9.5 7.0 11.2 9.1	31.8 28.5 21.0 33.6 27.3	53.0 47.5 35.0 56.0 45.5	120	135
7	(Removable) Pin ASTM A434 Class BD, AISI 4340, Hot Rolled, Q&T	Shear Bearing on Lower Adaptor Bearing on Top Lugs Bending	8.8 9.6 10.0 42.4	26.4 28.8 30.0 127.2	44.0 48.0 50.0 212.0	105	135

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

5887B:1/010483

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG,

LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value W[b]	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
8	Top Lugs	Bearing on Pin	10.0	30.0	50.0	ASTM A515	
	ASTM A515 GR 70	Tension at Pin Hole	9.8	29.4	49.0	38	70
	or						
	ASMT A516 GR 70	Shear Tear-out at Pin Hole	9.8	29.4	49.0	ASTM A516	
	Q&T	Tension at Weld	4.9	14.7	24.5	38	70
9	Block	Tension in Block	1.5	4.5	7.5	SA 105	
	SA 105 CL 1 or 2					36	70
	or						
	ASTM A266 CL 1					ASTM A266	
						CL1 30	60
						CL 2 35	70
	or						
	SA 508 CL 1 or 2					SA 508	
						CL 2 35	70
						CL 2 50	80
10	Side Lug	Tension at Pin Hole	5.6	16.8	28.0	ASTM A515	
	ASTM A515 GR 70	Shear Tear-out at Pin Hole	5.6	16.8	28.0	38	70
	or	Bearing on Pin	9.1	27.3	45.5		
	ASTM A516 GR 70	Shear at Lug Root Weld	1.9	5.7	9.5	ASTM A516	
	Q&T CL 2	Combined Bending and Tension at Lug Root Weld	5.2	16.5	26.0	38	70

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

5887B:1/010683

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value W[b]	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
11	(Upper Sling-Leg) Pin ASTM A276 Type 304 Cent. Grd. Cond. A	Shear Bearing on Sling Leg Bearing on Side Lug Bending	7.5 11.8 9.1 28.8	22.5 35.4 27.3 86.4	37.5 59.0 45.5 144.0	30	75
12	Sling Leg ASTM A515, GR 70 or ASTM A516, GR 70	Bearing on Pins Tension in Leg Tension at Pin Hole Shear Tear-out at Pin Hole	11.8 5.9 12.0 12.0	35.4 17.7 36.0 36.0	59.0 29.5 60.0 60.0	ASTM A515 38 ASTM A516 38	 70 70
13	(Lower Sling Leg) Pin ASTM A276 Type 304 Cent. Grd. Cond. A	Shear Bearing on Sling Leg Bearing on Spreader Bending	7.5 11.8 6.1 35.3	22.5 35.4 18.3 105.9	37.5 59.0 30.5 176.5	30	75

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Designation	Calculated Stresses (ksi) Value W[b]	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
14	SPREADER					SA 105 36	 90
14-A	Spacer Block SA 105 CL 1 or 2 Q&T, or ASTM A266 CL 1 or 2 or SA 508 CL 1 or 2 Q&T	Bending Shear Bearing	16.3 3.4 6.1	48.9 10.2 18.3	81.5 17.0 30.5	ASTM A266 30 CL1 35 CL2 SA508 35 CL1 50 CL2	 60 CL1 70 CL2 70 CL2 80 CL2
14-B, C, D	Spreader Assembly ASTM A515 GR 70 or ASTM A516 GR 70	Nominal Compression Buckling Welds (max. shear stress)	3.3 1.2 11.1	9.9 NA 33.3	16.5 NA 55.5	38 Ratio 18[f]	70 < 1.0[g]

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)
(f) Stress limit for fillet weld from ASME Boiler & Pressure Vessel Code, Section III, Division 1 - Subsection NF 1980 Edition, Table NF - 3292.1-1 page 50
(g) Stress limit ratio's for buckling from ASME Boiler & Pressure Vessel Code, Section III, Division 1, Appendices, Article XVII-2215, 1980 Ed.

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value W[b]	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
15	NUT ASTM A276 Type 304 HR & PKLD. Cond. A	Thread Shear Bearing on Spacer Block	11.9 14.5	35.7 43.5	59.5 72.5	38[e]	82[e]
16	(Leg) Adaptor ASTM A276 Type 304 HR & PKLD. Cond. A	Thread Shear Tension at Threads	11.9 10.2	35.7 30.6	59.5 51.0	43.9[e]	86.5[e]
17	(Leg) Outer Tube ASTM A312 Type 304 SMLS CF and HT. TR.	Tension	10.3	30.9	51.5	38.0[e]	83.4[e]
18	(Leg) Leveling Sleeve ASTM A276 Type 304 HR & PKLD. Cond. A.	Tension at 6.60 Dia. Thread Shear	3.2 4.0	9.6 12.0	16.0 20.0	30	75

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)
(e) These are actual S_y and S_{ult} taken from the material certification

5887B:1/010483

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Calculated Stresses (ksi)				Material Allowable (ksi)	
			Value				
		Designation	W[b]	3W	5W	S_y[c]	S_{ult}[d]
19	Coupling ASTM A312 Type 304 SMLS CF & HT Tr.	Thread Shear Tension at Thread Relief	4.0 4.5	12.0 13.5	20.0 22.5	30	75
20	(Support Ring) Leveling Sleeve ASTM A276 Type 304 HR & PLKD. Cond. A.	Thread Shear on 7.000-8UN Thread Tension at THD. Relief Thread Shear on 5.500-12UN Thread	4.0 5.6 4.9	12.0 16.8 14.7	20.0 28.0 24.5	30	75
21	(Support Ring) Outer Tube ASTM A312 Type 304 SMLS CF & HTTR	Thread Shear Tension at Thd. Relief Tension at Weld	4.9 8.4 6.5	14.7 25.2 19.5	24.5 42.0 32.5	30	75

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)

5887B:1/010483

TABLE 5-2 (cont)

SUMMARY OF RESULTS REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LOAD CELL LINKAGE

Item[a] No.	Part Name And Material	Calculated Stresses (ksi) Designation	Value $W^{(b)}$	3W	5W	Material Allowable (ksi) S_y[c]	S_{ult}[d]
22	Guide Sleeve ASTM A276 Type 304 HR & PKLD. Cond. A	Bearing on Engaging Screw Nominal Compression Below Engaging Screw	13.6 8.3	40.8 24.9	68.0 41.5	30	75
23	Engaging Screw ASTM A276 Type 304	Bearing on Guide Sleeve Tension at Minimum Section Thread Shear	13.6 13.5 5.1	40.8 40.5 15.3	68.0 67.5 25.5	36.8[e]	80.9[e]

(a) See figure 5-2 for location of item number and section
(b) W is the total static weight of the component and the lifting device
(c) S_y is the yield strength of the material (ksi)
(d) S_{ult} is the ultimate strength of the material (ksi)
(e) These are actual S_y and S_{ult} taken from the material certification

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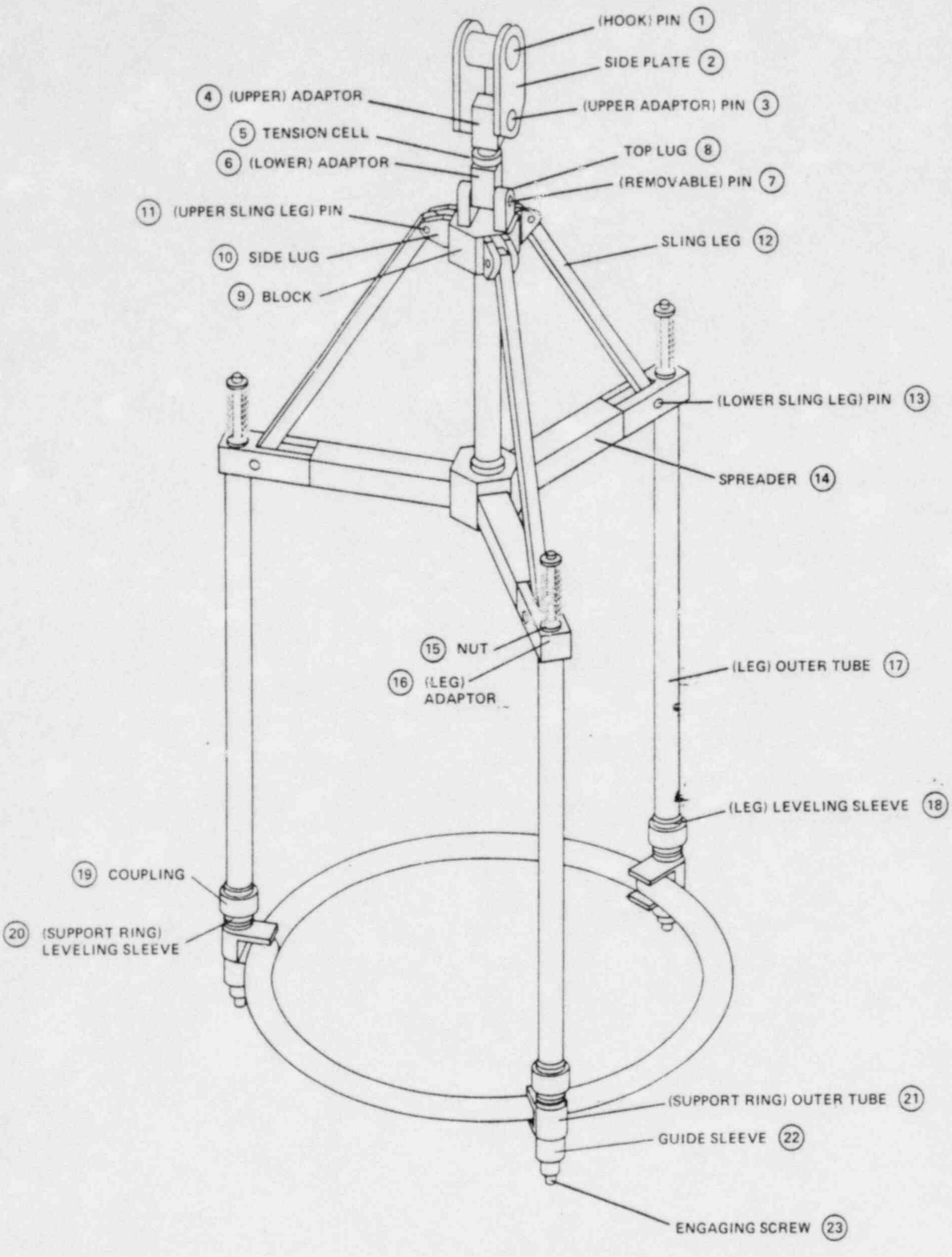

Figure 5-2 Reactor Vessel Internals Lifting Rig

TABLE 5-3

SUMMARY OF RESULTS

REACTOR COOLANT PUMP MOTOR LIFT RIG

Item[(a)] No.	Part Name And Material	Calculated Stresses (ksi)				Material Allowable (ksi)	
		Designation	Value				
			W[(b)]	3W	5W	S_y [(c)]	S_{ult} [(d)]
1	Spreader	Compressive Buckling	5.2	15.4	25.7	F_c [(e)] = 19.35 ksi	
		Stress on Tube-to-Tube Weld	2.9	8.6	14.3	35	60
		Stress on Tube-to-Plate Weld	7.1	21.3	35.5		
2	Plate	Tension	1.9	5.6	9.3	36	58

(a) See Figure 5-3 for location of item number and section

(b) W is tne total static weight of the component and the lifting device

(c) S_y is the yield strength of the material (ksi)

(d)- S_{ult} is the ultimate strength of the material (ksi)

(e) F_c is the compressive buckling strength of the material (ksi)

5887B:1/120682

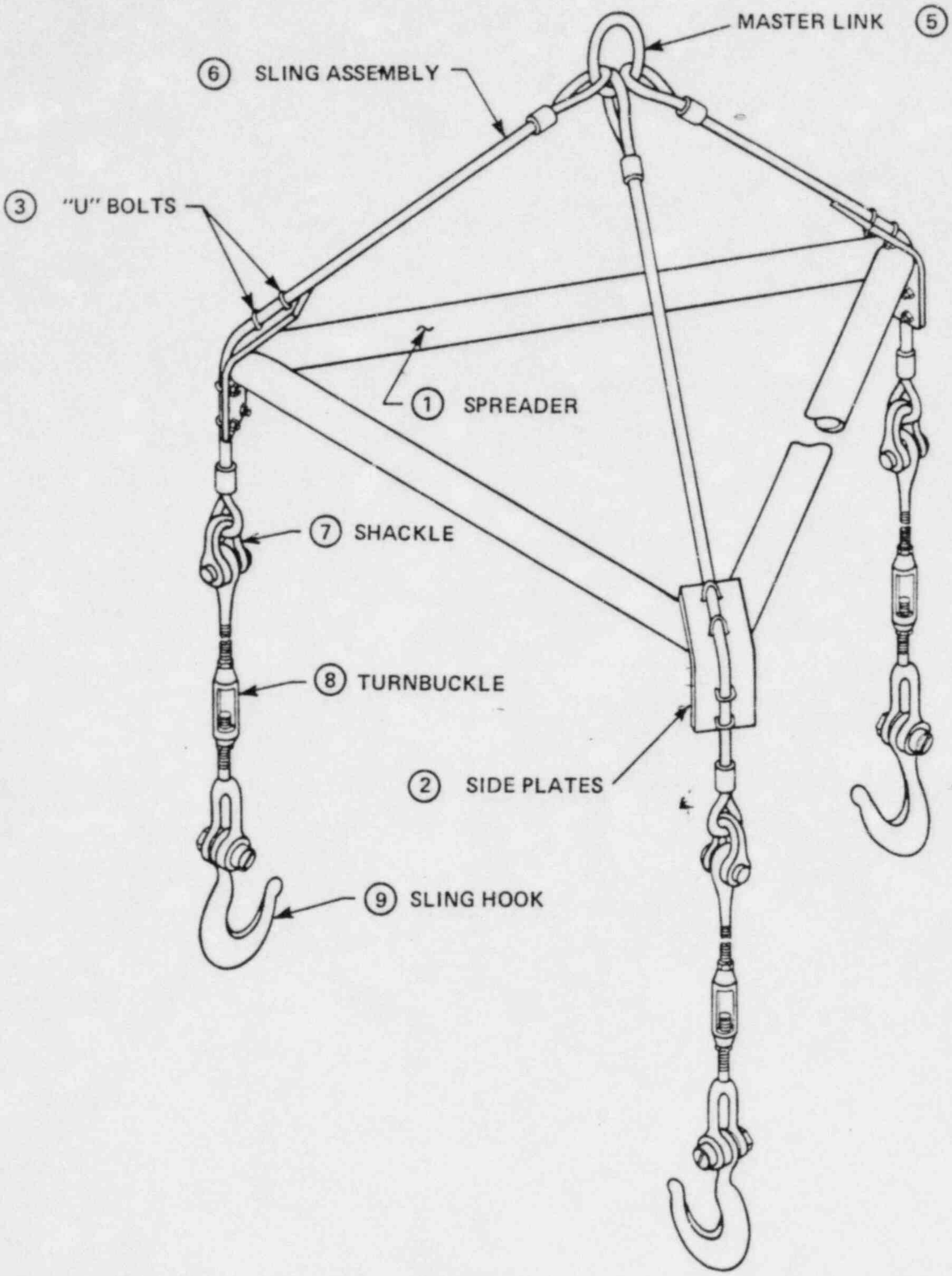

Figure 5-3. Reactor Coolant Pump Motor Lift Sling

TABLE 5-4

COMPARISON OF DESIGN LOADS

AND RATED LOAD VALUES OF THE NON-DESIGNED

ITEMS OF THE R.C. PUMP MOTOR LIFT SLING

No.(7)	Item	Loads (Pounds)(6)				Safety Factor(4)
		Design	Rated(1) Load Value	Proof(2) Load	Ultimate(3) Load	
5	Master Link	81,000	160,000	320,000	544,000	3.4:1
6	Sling	81,000	81,000(5)	94,000(5)	405,000	5:1(5)
7	Shackle	27,000	70,000	154,000	420,000	6:1
8	Turnbuckle	27,000	37,000	74,000	185,000	5:1
9	Hook	27,000	38,750	77,500	131,750	3.4:1

NOTES:

(1) RATED LOAD VALUE - The maximum recommended load that should be exerted on the item. The following terms are also used for the term Rated Load: "SWL", "Safe Working Load", "Working Load", "Working Load Limit", and the "Resultant Safe Working Load." All rated load values, are for in-line pull with respect to the centerline of the item.

(2) PROOF LOAD - The average force to which an item may be subjected before visual permanent deformation occurs or a force that is applied in the performance of a proof test.

(3) ULTIMATE LOAD - The average load or force at which item fails or no longer supports a load.

(4) SAFETY FACTOR - An industry term denoting theoretical reserve capability. Usually computed by dividing the catalog stated ultimate load by the catalog stated working load limit and generally expressed as a ratio, for example 5 to 1.

TABLE 5-4 (Cont)

(5) This information is as stated on Westinghouse drawing AED SK 618J644 TXK Sub 5 as follows: "Safe working load of this sling assembly is 81,000 lb and a safe factor of 5:1." Catalog information is not applicable.

(6) The rated load value, proof load, ultimate load and safety factor information was obtained from the following vendor catalogs:

a. S.G. Taylor Chain Co., Inc., Bulletin AS-67 Alloy Steel Chain Assemblies, Attachments for items 5 and 9.

b. Pennsylvania Sling Co. for item 6.

c. Crosby Group, 950 General Catalog, June 1981 for items 7 and 8.

(7) Refer to figure 5-3 for identification of items.

APPENDIX A

DETAILED STRESS ANALYSIS - REACTOR VESSEL HEAD LIFT RIG

This appendix provides the detailed stress analysis for the Turkey Point Units 3 and 4 Reactor Vessel Head Lift Rig in accordance with the requirements of ANSI N14.6. Acceptance criteria used in evaluating the calculated stresses are based on the material properties given in section 5.

SKETCH SHEET
WESTINGHOUSE FORM 54202

S.O.	PROJECT	PAGE
FJIP-93447	Turkey Point Units 3 and 4	1 OF 26

TITLE	CALCULATIONS NO.
R. V. Head Lift Rig Assembly	PDC -

AUTHOR & DATE	CHECKED BY & DATE
J. Urban [signature] 12/82	J. Richard [signature] 12/82

PURPOSE AND RESULTS:

1. The purpose of this analysis is to determine the acceptability of this rig to the requirements of ANSI N14.6.

2. The results show that all stresses are within the allowable stresses.

COMMONWEALTH OF PENNSYLVANIA
REGISTERED PROFESSIONAL
JOSEPH W. RICHARD
ENGINEER
NO. 037107-E

J W Richard

REVISION NO.	DATE	DESCRIPTION	BY
		Original Issue	J. Urban

RESULTING REPORTS, LETTERS OR MEMORANDA:

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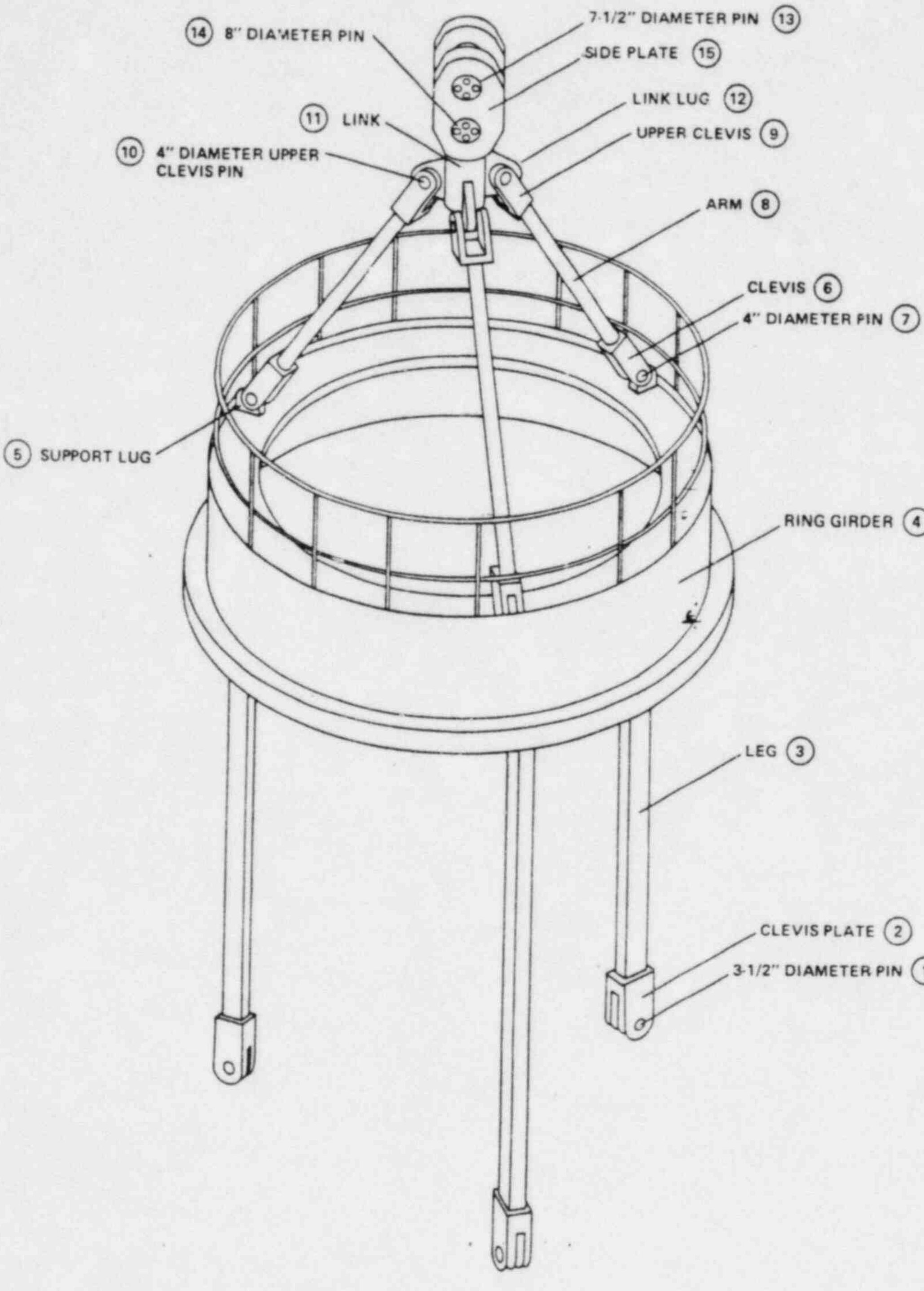
HEAD LIFT RIG			2 OF 26
PROJECT FPL / FLA	AUTHOR [signature] 1/83	CHK'D. BY JW Richard 1/83	CHK'D. BY
S.O. FJIP-93447	CALC. NO.	FILE NO.	GROUP CHF



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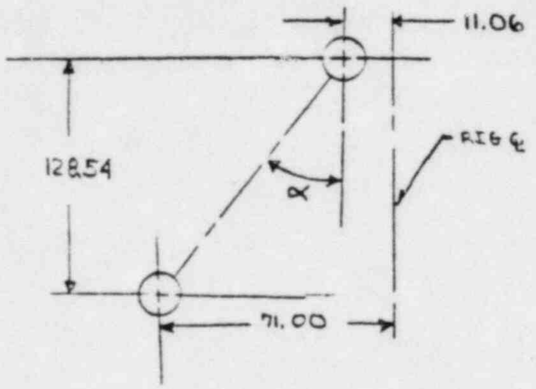
W = DESIGN WEIGHT
= WEIGHT OF ASSEMBLED R.V. HEAD AND ACCESSORIES + WEIGHT OF LIFT RIG
= 278,000 LBS

SLING LEG ANGLE



$\tan\alpha = (71.00 - 11.06)/128.54$

$\alpha = 25.00°$

T = TENSION IN SLING LEG

$T\cos\alpha = (W/3)$

$T = W/3/\cos 25.00°$

$T = W(.3678)$

K = HORIZONTAL COMPONENT OF T

$\tan\alpha = K/(W/3)$

$K = \tan\alpha * W/3$

$= \tan 25.00° * W/3$

$K = W(.15544)$

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HEAD LIFT RIG			4 of 26
PROJECT: FPL	AUTHOR / DATE: [signature] 12/82	CHK'D. BY / DATE: JW Bushard 12/22	CHK'D. BY / DATE:
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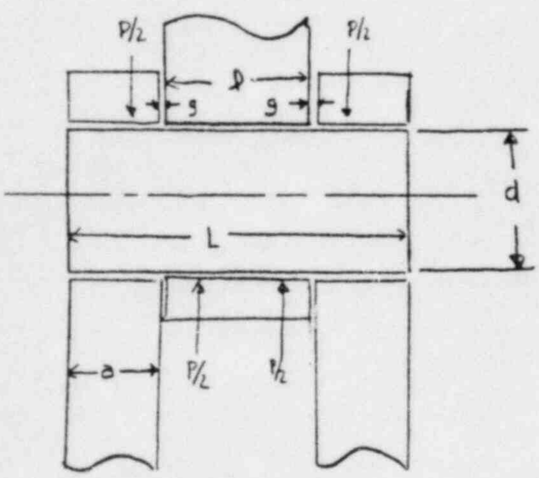
3½" DIA PIN (HEAD CLEVIS PIN) ①

MAT'L
ASTM A434 CLASS BD



P = FORCE ACTING ON ASSEMBLY, LB.
d = DIAMETER OF PIN, IN.
l = LENGTH OF BEARING SURFACE OF CENTER BODY, IN.
a = LENGTH OF BEARING SURFACE ON ONE SIDE OF OUTER BODY, IN
g = GAP BETWEEN BEARING SURFACES, IN.
L = TOTAL ACTIVE LENGTH OF PIN, IN
= l + 2(a+g)

P = W/3 lb
d = 3.5025 in
l = 3.00 nom
a = 1.75 - 2(.08) = 1.59 in
g = .265 in
= [3.37 + 2(.08) - 3.00]/2

SHEAR STRESS

$f_v = P/2A_v = (W/3)/2/A_v$

$A_v = \pi d^2/4 = \pi(3.5025)^2/4 = 9.6349 in^2$

$f_v = W/3/2/9.6349$
$= W(.017298)$

$f_v = 4809$ PSI

BEARING STRESSES

$f_c = F/A_c = (W/3)/A_c$

INNER

$A_c = dl = 3.5025 * 3.00$
$= 10.508 in^2$

$f_c = W/3/10.508$

$f_c = W(.03172)$

$f_c = 8819$ PSI

OUTER

$A_c = 2ad = 2 * 1.59 * 3.5025$
$= 11.138 in^2$

$f_c = W/3/11.138$
$= W(.02993)$

$f_c = 8320$ PSI

BENDING STRESS

$f_b = 16P\left(\frac{a}{3} + g + \frac{l}{4}\right)/(\pi d^3)$

$= 16 * (W/3) * \left(\frac{1.59}{3} + .265 + \frac{3.00}{4}\right)$
$/(\pi * 3.5025^3)$

$= W(.06104)$

$f_b = 16,970$ PSI

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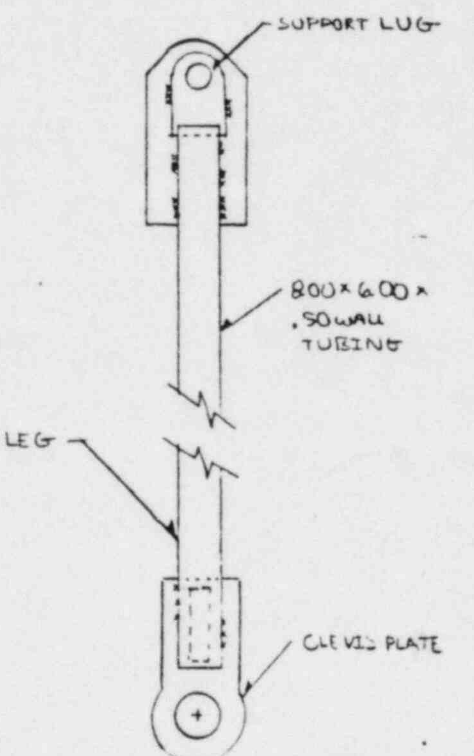
S.O.: FJIP-93447	CALC NO.:	FILE NO.:	GROUP: CHE

LEG ASSEMBLY (2) (3)

MAT'L:

CLEVIS PLATE — ASTM A-515 GRADE 70 Q&T

LEG — ASTM A36



LEG (3)

TENSION IN LEG TUBING

$f_t = P/A_t$

$P = W/3$

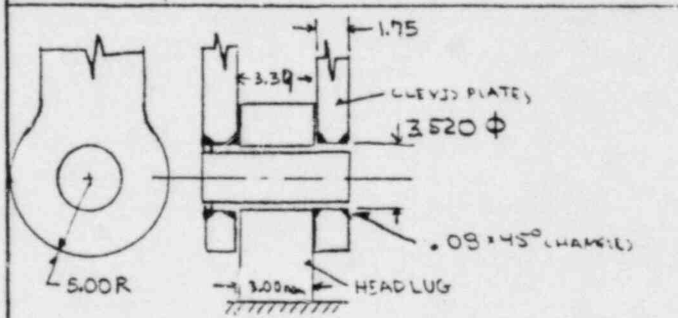
$A_t = 11.9\,in^2$ (AISC)

$f_t = W/3/11.9$

$= W(.02801)$

$\underline{f_t = 7787\ PSI}$

CLEVIS PLATE (2)



TENSION @ PIN-HOLE

$f_t = P/A_t = (W/3)/A_t$

$A_t = (10.00 - 3.520)(1.75)(2.0) - 2(.08)^2 = 22.67$

$f_t = W/3/22.67$

$= W(.014706)$

$\underline{f_t = 4088\ PSI}$

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CLEVIS PLATE (2)

SHEAR TEAR OUT @ PIN-HOLE

$f_v = P/2A_v = (W/3)/2A_v$

$A_v = (5.0 - \frac{3.500}{2})(1.75)(2) - .08^2$

$= 11.334\ in^2$

$f_v = W/3/2/11.334\ in^2$

$= W(.014706)$

$f_v = 4088$ PSI

tension above at straight side
$(1.75)(7) = 24.5\ in^2$
$= 22.74$

BEARING STRESS

THE BEARING STRESS IS THE SAME AS THE OUTER BEARING STRESS ON THE HEAD CLEVIS PIN

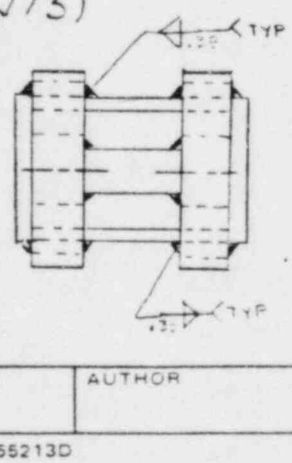
$f_c = W(.02993)$

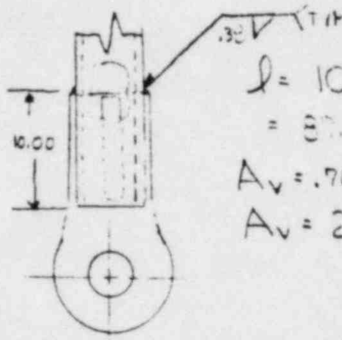
$f_c = 8320$ PSI

WELD OF CLEVIS PLATE TO LEG

$f_v = P/A_v$

$P = (W/3)$





$l = 10(8) + 7(1.75)$

$= 87.0\ in$

$A_v = .707(.38)*87.0$

$A_v = 23.373\ in^2$

$f_v = W/3/23.373$

$= W(.014261)$

$f_v = 3965$ PSI

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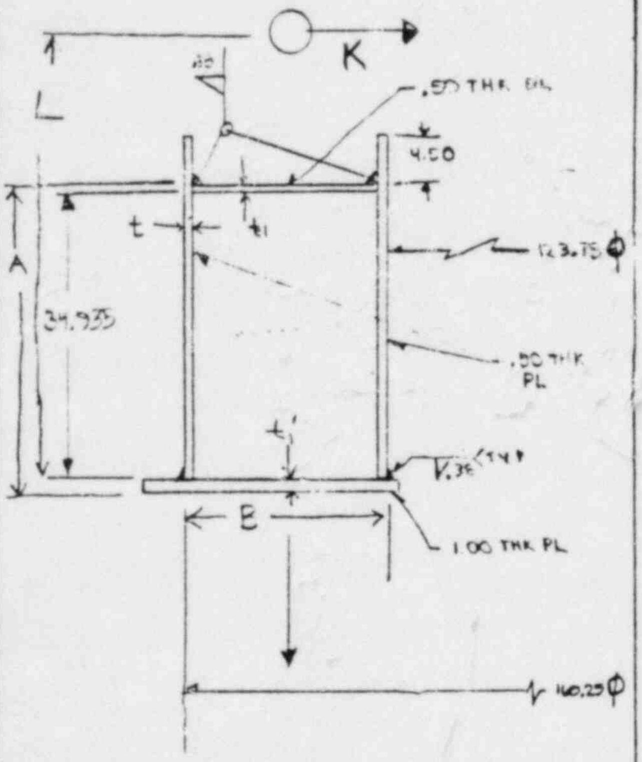
RING GIRDER (4)

MAT'L
ASTM-A285 GRC
FIREBOX



TORSIONAL SHEAR

CONSERVATIVELY LETTING $t_1' = t_1$
$= .50$ in

$$f_{torsion} = T_m / (2t[A-t][B-t_1])$$

FROM ROARK, 3RD EDITION
PAGE 176 CASE 12
NEGLECTING BRACE PLATE AND

$t = .50$

$t_1 = .50$

$A = 34.935 + 2(.50) = 35.935$ in

$B = (160.25 - 123.75)/2 =$

$= 18.25$ in

$T_m = K\,l$

$L = 30.50 + 11.0 = 41.50$

$l = L - A/2$

$= 41.50 - 35.935/2 = 23.533$ in

$K = .15544\,W$

$T_m = .15544\,W\,(23.533)$

$= W(3.6579)$ in-lb

$$f_{torsion} = \frac{3.6579\,W}{2(.50)[35.935 - .5][18.25 - .50]}$$

$= W(.005816)$

$f_{torsion} = 1617$ PSI

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S.O.: FJIP-93447	CALC. NO.	FILE NO.	GROUP: CHE

RING GIRDER (4)

SHEAR

$f_v = P/A_v$

$P = K = .15544W$

$A_v = [34.935 + 2(.50)] * [160.25 - 123.75]/2 - 34.935 * [\frac{160.25-123.75}{2} - 2(.50)]$

$= 53.185 \, in^2$

$f_v = .15544W / 53.185$

$= W(.002923)$

$f_v = 812$ PSI

TOTAL SHEAR

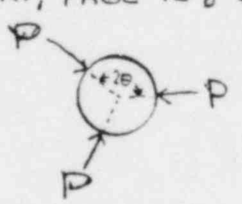
$f_v = f_v + f_{v\,torsion}$

$1617 + 812$

$f_v = 2429$

BENDING STRESS

ROARK, PAGE 158 CASE 9



$s = \sin\theta$

all angles in radians

$M_{max} = \frac{1}{2} PR \left(\frac{1}{s} - \frac{1}{\theta}\right)$

BETWEEN LOADS

$\theta = 2\pi/6 = \pi/3$ radians

$M_{max} = \frac{1}{2} K \left(\frac{123.75 + 160.25}{4}\right) * \left(\frac{1}{\sin \pi/3} - \frac{1}{(\pi/3)}\right)$

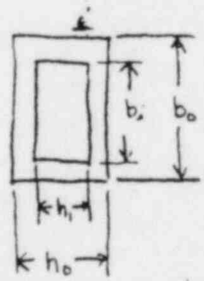
$= [.15544W/2] * 14.184$

$= 1.1024 W$ in-lb

$-M_{max} = -\frac{1}{2} PR \left(\frac{1}{\theta} - \cot\theta\right)$

$-M_{max} = -\frac{W}{2}(.15544)\left(\frac{123.75+160.25}{4}\right) * \left[\frac{1}{(\pi/3)} - \frac{1}{\tan 60°}\right]$

$= -2.0837 W$ in-lb



$h_o = \frac{160.25 - 123.75}{2} = 18.25$ in

$h_i = h_o - 2(.50) = 17.25$ in

$b_i = 34.935$ in

$b_o \approx b_i + 1 + .5 = 36.435$ in

$Z = I/c = \frac{b_o h_o^3 - b_i h_i^3}{6 h_o}$

$= \frac{36.435 * 18.25^3 - 34.935 * 17.25^3}{6 * 18.25}$

Z = 3[illegible]0 in^3

$f_b = M_{max}/Z = \frac{-2.0837W}{384.90}$

$= W(.005414)$

$f_b \sim 1787$ PSI

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RING GIRDER (4)

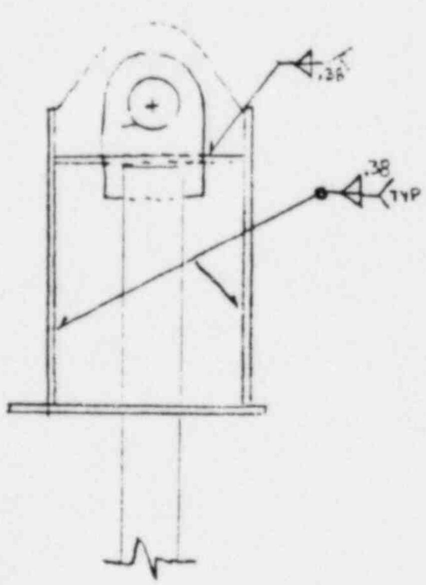
MAXIMUM TENSILE STRESS

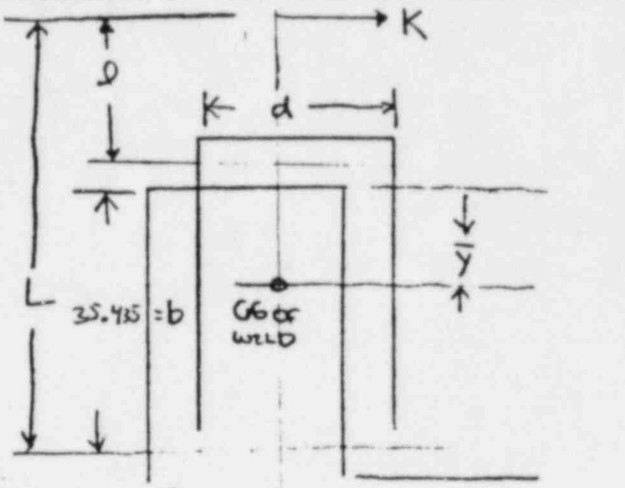
$$f_{max} = \frac{f_b}{2} + \sqrt{\left(\frac{f_b}{2}\right)^2 + f_v^2}$$

$$= \frac{1787}{2} + \left(\left(\frac{1787}{2}\right)^2 + 2429^2\right)^{1/2}$$

$$f_{max} = 3482 \text{ PSI}$$

WELD OF SUPPORT LUG TO RING GIRDER





$$l = (\overbrace{30.50 + 11.0}^{=L}) - 34.935 - .50 = 6.065''$$

$$34.935 + .50 = 35.435 \text{ in}$$

$$\bar{x} = [(160.25 - 123.75)/2 - 2(.50)]/2$$
$$= 8.625 \text{ in}$$

$$\bar{y} = \frac{b^2}{2b+d} = \frac{35.435^2}{2(35.435) + 8.625(2)}$$
$$= 14.249 \text{ in}$$

$$d = 8.625(2) = 17.25 \text{ in}$$

$$M = K(\bar{y} + l)$$
$$= .15549W(14.249 + 6.065)$$
$$= W(3.1576) \text{ in-lb}$$

$$J_u = \frac{8b^3 + 6bd^2 + d^3}{12} - \frac{b^4}{2b+d}$$

$$J_u = [8 * 35.435^3 + 6 * 35.435 * 17.25^2 + 17.25^3]/12 - 35.435^4/(2 * 35.435 + 17.25)$$
$$= 17,470 \text{ in}^3$$

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$J = [.707(.38) J_u] * 2$
$= 9387 \text{ in}^4$

$A_w = .707(.38)[17.25 + 2(35.435)]2$
$= 47.34 \text{ in}^2$

$\tau_x' = \frac{K}{A_w} = \frac{.15544}{47.34} W$

$= W(.003283)$

$= \underline{913 \text{ PSI}}$

FROM RING GIRDER WEIGHT

$\tau_y' = \frac{w}{A_w} = \frac{5000}{47.34} = \underline{105 \text{ PSI}}$

$\tau_y'' = \frac{M r_x}{J} = \frac{3.1576 W (17.25/2)}{9387}$

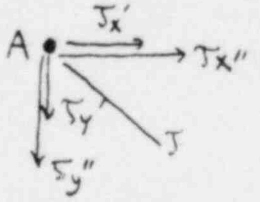
$= W(.002901)$

$\underline{\tau_y' = 807 \text{ PSI}}$

$\tau_x'' = \frac{M r_y}{J} = \frac{3.1576 W (14.24)}{9387}$

$= W(.004790)$

$\underline{\tau_x'' = 1331 \text{ PSI}}$



RESULTANT STRESS IN WELD

$\tau = \sqrt{\tau_y^2 + \tau_x^2}$

$= [(105 + 807)^2 + (913 + 1331)^2]^{1/2}$

$\underline{\tau = 2422 \text{ PSI}}$

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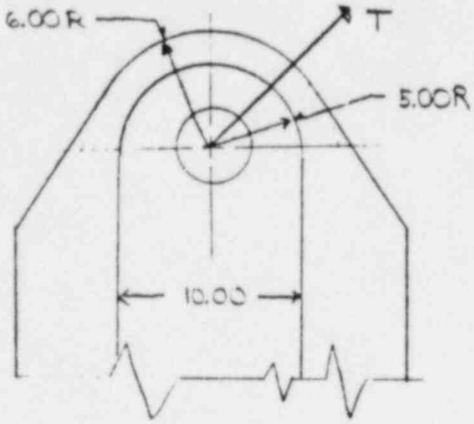
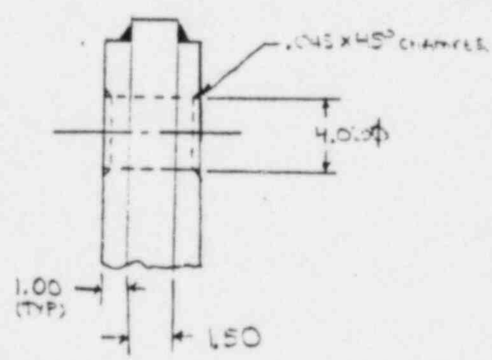
SUPPORT LUG (5)





MAT'L:
ASTM-A-515 GR 70
NORMALIZED & TEMPERED

TENSION @ PIN-HOLE

$f_t = P/A_t$

$P = T = .3678W$

TENSION @ PIN HOLE

CONSERVATIVELY

$A_t \simeq (1.50 + 2(1.00)) * (10.00 - 5.00) - 2(.045)^2 = 17.495 \text{ in}^2$

$f_t = .3678W/17.495$

$f_t = W(.02102)$

$f_t = 5844$ PSI

SHEAR TEAR-OUT @ PIN HOLE

$f_v = P/2A_v$

$P = T = .3678W$

$A_v = [1.00 * (5 - 5.00/2)] * 2 + 1.50 * (6 - 5.00/2) - .045^2$

$A_v = 10.248 \text{ in}^2$

$f_v = .3678W/2/10.248$

$= W(.017945)$

$f_v = 4989$ PSI

BEARING ON PIN

THE BEARING STRESS IS THE SAME AS THE BEARING ON THE LOWER CLEVIS PIN (INNER)

$f_c = W(.026948)$

$f_c = 7492$ PSI

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UPPER CLEVIS (9) AND CLEVIS (6)

MAT'L:
ASTM A237 CLASS A

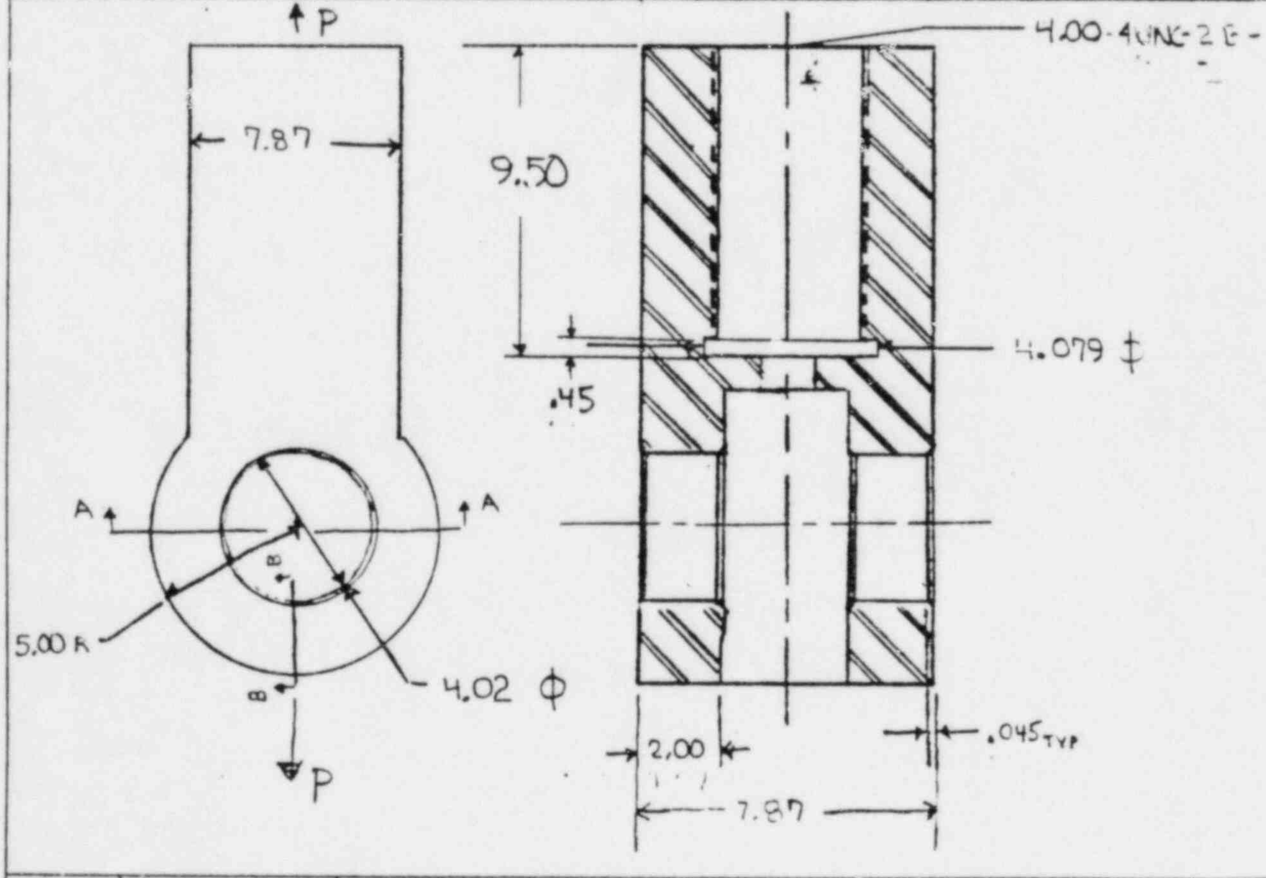
THREAD SHEAR

THE THREAD SHEAR IS THE SAME AS FOR THE ARM LEG

$$f_v = \frac{P}{(D_{pitch}\, \pi\, l/2)}$$

$$f_v = W(.012203)$$

$$\underline{f_v = 3392 \text{ PSI}}$$



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UPPER AND BOTTOM CLEVIS (6) (9)

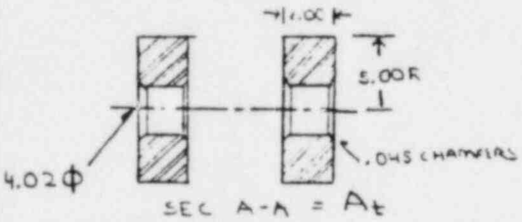
BEARING STRESS

THE BEARING STRESS IS THE SAME AS THE OUTER BEARING STRESS ON THE CLEVIS PINS

$f_c = W(.02406)$

$f_c = 6687$ PSI

TENSION @ PIN-HOLE



$f_t = P/A_t$

$P = .3678$

$A_t = [2.00(5.00*2) - 2(4.02) - 2(.045)^2] * 2$

$= 23.91\ in^2$

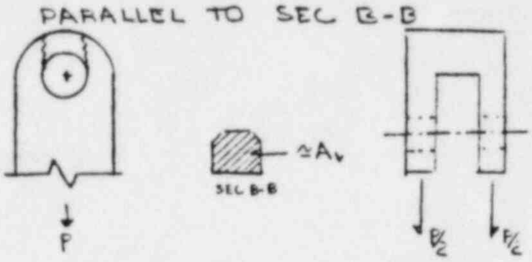
$f_t = W(.3678)/23.91$

$= W(.015383)$

$f_t = 4276$ PSI

SHEAR TEAR-OUT

PARALLEL TO SEC B-B



$f_v = P/4A_v$

$P = W(.3678)$

$A_v = 5.00(2.00) - (\frac{4.02}{2})(2.00) - .045^2$

$= 5.978\ in^2$

$f_v = W(.3678)/4/5.978$

$= W(.015383)$

$f_v = 4276$ PSI

TENSION @ THREAD RELIEF

$f_t = F/A_t$

$F = W(.3678)$

$A_t = bh - \pi d^2/4$

$= 7.87 * 7.87 - \pi * 4.079^2/4$

$= 48.87\ in^2$

$f_t = W(.3678)/48.87$

$f_t = W(.007526)$

$f_t = 2092$ PSI

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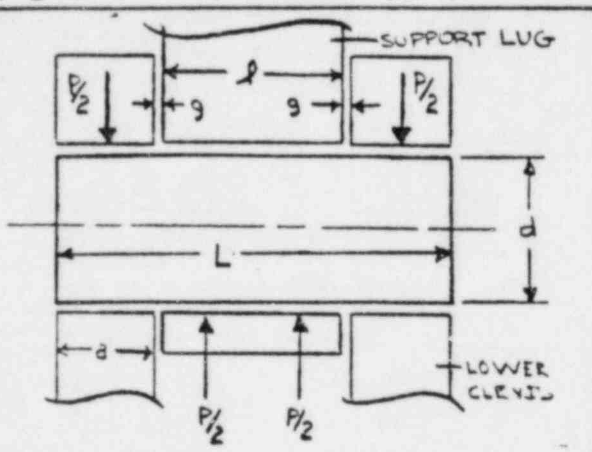
(LOWER CLEVIS PIN) (7)

4" DIA PIN

MAT'L: ASTM A434 CLASS BD
MINIMUM TEMPERING TEMP = 1100°F



P = force acting on assembly, lb.
d = diameter of pin, in.
l = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
= $l + 2(a+g)$

$P = .3678\,W = T$ lb.
$d = 4.0025$ in.
$l = 3.41$ in. $= 1(2) + 1.50 - 2(.045)$
$a = 1.91$ in $= 2.00 - 2(.045)$
$g = 0.275$ in
$= [7.87 - 2(2.00) + 2(.045) - 3.41]/2$

SHEAR STRESS

$f_v = P/2A_v$
$P = .3678\,W$
$A_v = \pi d^2/4$
$A_v = \pi(4.0025)^2/4$
$A_v = 12.582$ in²

$f_v = (.3678\,W)/(2 * 12.582)$
$= W(.014616)$
$f_v = 4063$ PSI

BEARING STRESS

$f_c = P/A_c$
$P = .3678\,W$

INNER

$A_c = d\,l = (4.0025)(3.41)$
$= (13.649)$ in²
$f_c = (.3678\,W)/(13.649)$
$= W(.026948)$
$f_c = 7492$ PSI

OUTER

$A_c = 2ad = 2(1.91)(4.0025)$
$= (15.290)$ in²
$f_c = (.3678\,W)/(15.290)$
$= W(.02406)$
$f_c = 6687$ PSI

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(LOWER CLEVIS PIN)

4" DIA PIN

(7)

BENDING STRESS(1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$

$$I = \pi d^4/64$$

$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$

$$= 16P\left(\tfrac{1}{3}a + g + \tfrac{1}{4}l\right)/(\pi d^3)$$

$$= 16(.3678W)/\pi/4.0025^3 *$$

$$\left(\frac{1.91}{3} + .275 + \frac{3.41}{4}\right)$$

$$= W(.051538)$$

$$\underline{\underline{f_b = 14,328 \text{ PSI}}}$$

(1) ADAPTED FROM FASTENING AND JOINING, 4th Ed, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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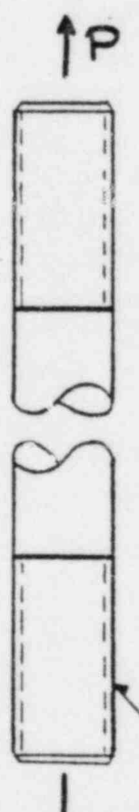
ARM (LEG) ⑧

MAT'L:

ASTM A-306 GR70



THREAD SHEAR

$f_v = P/A_v$

$A_v = D_{pitch} \pi \ell / 2$

FOR AN EXTERNAL THREAD

$D_{pitch} = D_{nom} - .64952/n$

$D_{pitch} = (4.00) - .64952/4.)$

$= (3.8376)$ in.

D_{nom} = major diameter of external thread

n = number of threads/in

ℓ = length of thread engagement

$= 5.00_{min}^{(1)}$ ($7.00_{nominal}$)

THEREFORE $A_v = (30.141)$ in²

$P = W(.3678)$

$f_v = W(.012203)$ /in²

$f_v = 3392$ PSI

THREAD TENSION

$f_t = P/A_t$

FOR EXTERNALLY THREADED PARTS

$A_t = \frac{\pi}{4}(D_{nom} - .9743/n)^2$

$= \frac{\pi}{4}(4.0 - .9743/4)^2 = 11.083$ in²

$f_t = W(.3678)/(11.083)$

$= W(.033186)$

$f_t = 9226$ PSI

(1) PER DWG NOTE OF SLING ASS'Y

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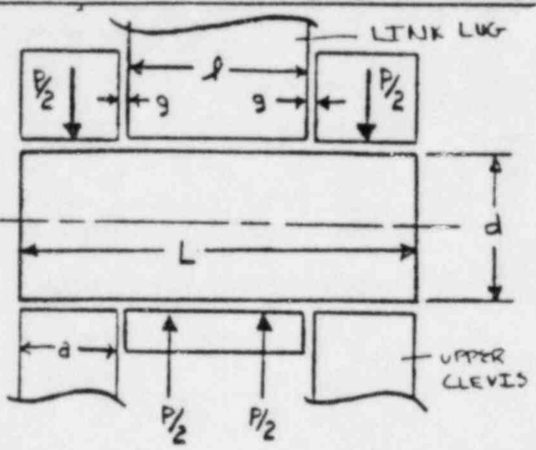
4" φ UPPER CLEVIS: PIN (10)

MAT'L: ASTM A434 CLASS BD
MINIMUM TEMPERING TEMP = 1100°F



P = force acting on assembly, lb.
d = diameter of pin, in.
ℓ = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
= ℓ + 2(a + g)

$P = T = .3678W$ lb.
$d = 4.0025$ in.
$\ell = 3.50$ in
$a = 1.91$ in $= 2.00 - 2(.045)$
$g = 0.230$ in
$= [7.87 - 2(2.00) + 2(.045) - 3.50]/2$

SHEAR STRESS

$f_v = P/2A_v$
$P = .3678W$
$A_v = \pi d^2/4$
$A_v = \pi(4.0025)^2/4$
$A_v = 12.582$ in²

$f_v = (.3678W)/(2 * 12.582)$
$= W(.014616)$
$\underline{\underline{f_v = 4063 \text{ PSI}}}$

BEARING STRESS

$f_c = P/A_c$
$P = .3678W$

INNER

$A_c = d\ell = (4.0025)(3.50)$
$= (14.009)$ in²
$f_c = (.3678W)/(14.009)$
$= W(.02626)$
$\underline{\underline{f_c = 7299 \text{ PSI}}}$

OUTER

$A_c = 2ad = 2(1.91)(4.0025)$
$= (15.290)$ in²
$f_c = (.3678W)/(15.290)$
$= W(.02406)$
$\underline{\underline{f_c = 6687 \text{ PSI}}}$

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4" DIA UPPER CLEVIS PIN

(10)

BENDING STRESS (1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$

$$I = \pi d^4/64$$

$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$

$$= 16P\left(\tfrac{1}{3}a + g + \tfrac{1}{4}l\right)/(\pi d^3)$$

$$= 16(.3678W)/\pi/4.0025^3 \ast \left(\frac{1.91}{3} + .230 + \frac{3.50}{4}\right)$$

$$= W(.050881)$$

f_b = 14,145 PSI

(1) ADAPTED FROM FASTENING AND JOINING, 4th ?, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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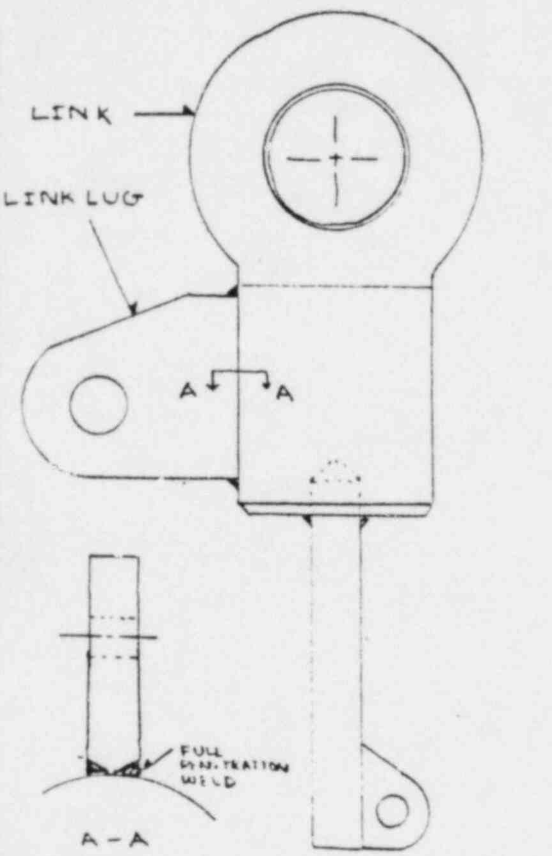
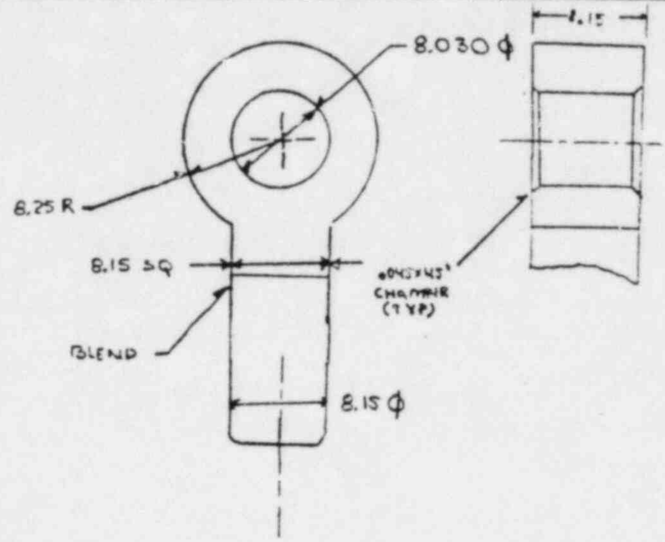
LIFTING LINK





MAT'L:
ASTM A-105 CL2

LINK (11)

BEARING ON PIN

THE BEARING STRESS IS THE SAME AS THE INNER BEARING STRESS ON THE BOTTOM PIN.

$f_c = W(.015509)$

$f_c = 4312$ PSI

TENSION AT PIN HOLE

$f_t = P/A_t$

$P = W$

$A_t = [8.25(2) - 8.030] \cdot 8.15 - 2(.045)^2$

$= 69.03 \text{ in}^2$

$f_t = W/69.03$

$= W(.014487)$

$f_t = 4027$ PSI

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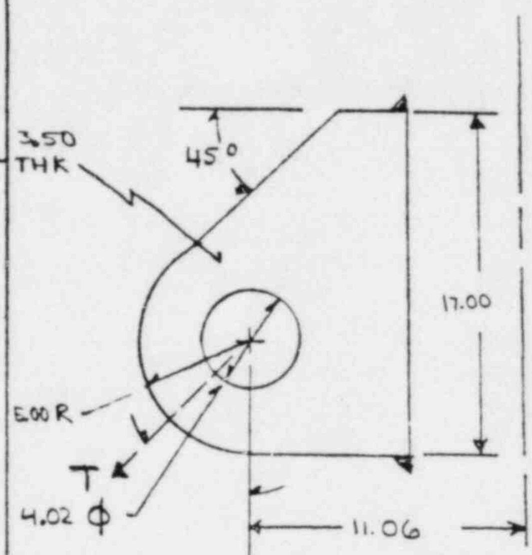
SLING ASSEMBLY LINK (11)



SHEAR TEAR-OUT @ PIN

$f_v = P/2A_v$

$P = W$

$A_v = (8.25 - \frac{8.030}{2}) * 8.15 - .045^2$

$= 34.513$

$f_v = W/2/34.513$

$= W(.014487)$

$f_v = 4027$ PSI

TENSION @ CYLINDRICAL SECTION

$f_t = P/A_t$

$P = W$

$A_t = \pi/4 (8.15)^2$

$= 52.168 \text{ in}^2$

$f_t = W/52.168$

$= W(.019169)$

$f_t = 5329$ PSI

TENSION @ PIN-HOLE

$f_t = P/A_t$

$P = T = .3678 W$

$A_t = [5.00(2) - 4.02] * 3.50$

$= 20.93 \text{ in}^2$

$f_t = .3678 W/20.93$

$= W(.017573)$

$f_t = 4885$ PSI

SHEAR TEAR-OUT @ PIN-HOLE

$f_v = P/2A_v = T/2A_v = .3678 W/(2A_v)$

$A_v = (5.00 - \frac{4.02}{2})(3.50) = 10.465 \text{ in}^2$

$f_v = .3678 W/2/10.465$

$= W(.017573)$

$f_v = 4885$ PSI

LINK LUG (12)

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LINK LUG (12)

BEARING @ PIN-HOLE

THE BEARING STRESS IS THE SAME AS THE INNER BEARING STRESS OF THE UPPER CLEVIS PIN

$f_c = W(.02626)$

$f_c = 7299$ PSI

SHEAR @ THE FULL-PENETRATION WELD

$f_v = P/A_v$

$P = W/3$

$A_v = bh = 17.00(3.50)$

$= 59.5 \text{ in}^2$

$f_v = W/3/59.5$

$= W(.005602)$

$f_v = 1557$ PSI

COMBINED STRESS FROM TENSION AND BENDING

$f_t = P/A_t$

$P = K = .15544W$

$A_t = bh = 17.00(3.50) = 59.5 \text{ in}^2$

$f_t = .15544W/59.5$

$= W(.002612)$

$f_t = 726$ PSI

$f_b = Mc/I$

$I = bh^3/12 = 3.50(17.00)^3/12$

$= 1433.0 \text{ in}^4$

$c = 17.00/2 = 8.50 \text{ in}$

$M = (W/3) * (11.06 - 8.15/2) - K\left(\frac{17.00}{2} - 5.00\right)$

$= W(2.329) - .15544W(3.50)$

$= W(1.7840)$

$f_b = W(1.7840)(8.50)/1433$

$= W(.010582)$

$f_b = 2942$ PSI

$f_{t_{COMBINED}} = f_t + f_b$

$= 726 + 2942$

$f_{t_{COMBINED}} = 3668$ PSI

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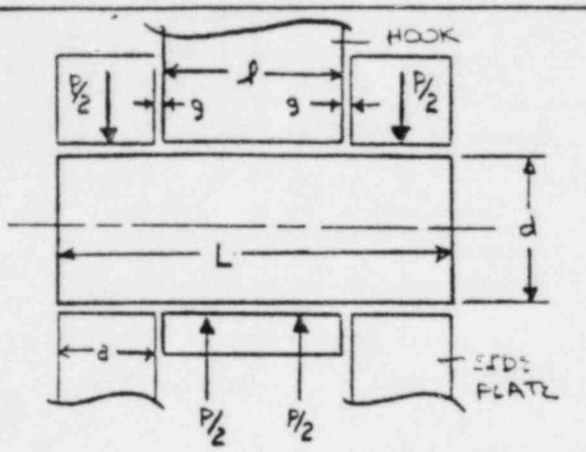
7½" DIA PIN (13)

MAT'L: ASTM A434 CLASS BD



P = force acting on assembly, lb.

d = diameter of pin, in.

ℓ = length of bearing surface of center body, in.

a = length of bearing surface one side of outer body, in

g = gap between bearing surfaces, in.

L = total active length of pin, in $= \ell + 2(a+g)$

P = W		lb.
d = 7.495		in.
ℓ = 8 nom		in
a = 3.00		in
g = 0.1875		in

$= [14.375 - 2(3.00) - 8]/2$

SHEAR STRESS

$f_v = P/2A_v$

$P = W$

$A_v = \pi d^2/4$

$A_v = \pi(7.495)^2/4$

$A_v = 44.120 \text{ in}^2$

$f_v = (W)/(2 \cdot 44.120)$

$= W(.011333)$

$f_v = 3151$ PSI

BEARING STRESS

$f_c = P/A_c$

$P = W$

INNER

$A_c = d\ell = (7.495)(8)$

$= (59.96) \text{ in}^2$

$f_c = (W)/(59.96)$

$= W(.016678)$

$f_c = 4636$ PSI

OUTER

$A_c = 2ad = 2(3.00)(7.495)$

$= (44.97) \text{ in}^2$

$f_c = (W)/(44.97)$

$= W(.02224)$

$f_c = 6183$ PSI

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7½" DIA PIN

(13)

BENDING STRESS (1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$

$$I = \pi d^4/64$$

$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$

$$= 16P\left(\tfrac{1}{3}a + g + \tfrac{1}{4}l\right)/(\pi d^3)$$

$$= 16W\left(\frac{3.00}{3} + 0.1875 + \frac{8}{4}\right)/\pi/7.495^3$$

$$= W(.03856)$$

$$f_b = 10{,}720 \text{ PSI}$$

(1) ADAPTED FROM FASTENING AND JOINING, 4th Ed, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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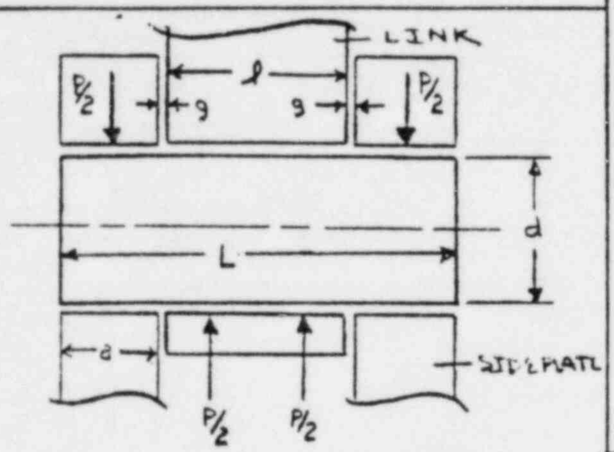
"8" DIA PIN (14)

MAT'L: ASTM A 434 CLASS BD



P = force acting on assembly, lb.
d = diameter of pin, in.
l = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
= $l + 2(a + g)$

P = W lb.
d = 8.000 in.
l = 8.06 in. = 8.15 - 2(.045)
a = 3.00 in
g = 0.1575 in
= [14.375 - 2(3.00) - 8.06]/2

SHEAR STRESS

$f_v = P/2A_v$
$P = W$
$A_v = \pi d^2/4$
$A_v = \pi (8.000)^2/4$
$A_v = 50.27 \text{ in}^2$

$f_v = (W)/(2 * 50.27)$
$= W(.009947)$
$f_v = 2765$ PSI

BEARING STRESS

$f_c = P/A_c$
$P = W$

INNER

$A_c = d\,l = (8.000)(8.06)$
$= (64.48) \text{ in}^2$
$f_c = (W)/(64.48)$
$= W(.015509)$
$f_c = 4312$ PSI

OUTER

$A_c = 2ad = 2(3.00)(8.000)$
$= (48.0) \text{ in}^2$
$f_c = (W)/(48.0)$
$= W(.02083)$
$f_c = 5791$ PSI

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8" DIA PIN

(14)

BENDING STRESS (1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$
$$I = \pi d^4/64$$
$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$
$$= 16P\left(\tfrac{1}{3}a + g + \tfrac{1}{4}l\right)/(\pi d^3)$$

$$= 16W\left(\frac{3.00}{3} + .1575 + \frac{8.06}{4}\right)/\pi/8.000^3$$
$$= W(.03156)$$

$$f_b = 8774 \text{ PSI}$$

(1) ADAPTED FROM FASTENING AND JOINING, 4th Ed, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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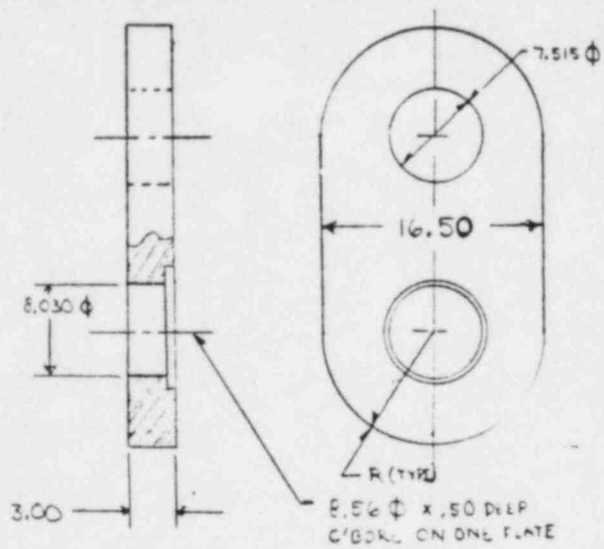
SIDE PLATE (15)



MAT'L:
ASTM A514
OR
USST-1

BEARING @ 7.515 ϕ

THE MAXIMUM BEARING STRESS IS THE SAME AS THE OUTER BEARING STRESS ON THE HOOK PIN

$f_c = W(.02224)$

$f_c = 6183$ PSI

TENSION @ 8.030 ϕ HOLE

$f_t = P/A_t$

$P = W/2$

$A_t = 16.50 * 3.00 - 8.030(3.00) - (8.56 - 8.030) * .50$

$= 25.145 \text{ in}^2$

$f_t = W/2/25.145$

$= W(.019885)$

$f_t = 5528$ PSI

SHEAR TEAR-OUT @ 8.030 ϕ HOLE

$f_v = F/2A_v$

$F = W/2$

$A_v = \left(\frac{16.50}{2} - \frac{8.030}{2}\right) * 3 - \left(\frac{8.56 - 8.030}{2}\right) * .50$

$= 12.573 \text{ in}^2$

$f_v = (W/2)/2/12.573$

$= W(.019885)$

$f_v = 5528$ PSI

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WESTINGHOUSE FORM 55213D

APPENDIX B

DETAILED STRESS ANALYSIS - REACTOR VESSEL INTERNALS LIFT RIG, LOAD CELL AND LINKAGE

This appendix provides the detailed stress analysis for the Turkey Point Units 3 and 4 reactor vessel internals lift rig, load cell and linkage, in accordance with the requirements of ANSI N14.6. Acceptance criteria used in evaluating the calculated stresses are based on the material properties given in section 5.

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FPL / FLA	[signature] 1/83	JW Richard 1/83	

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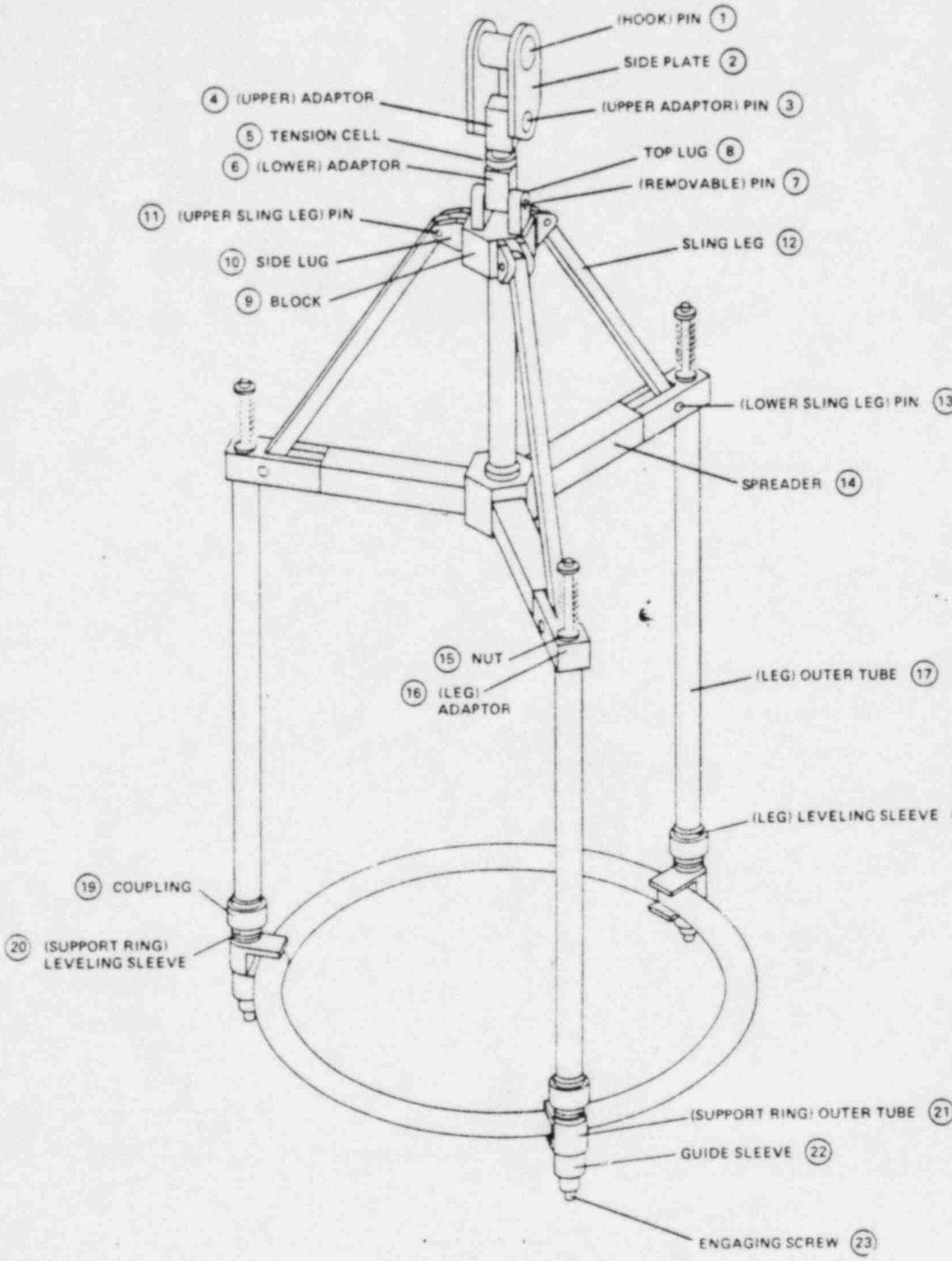
FJIP-93447			CHE



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SKETCH SHEET
WESTINGHOUSE FORM 54202

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FJIP-93447	Turkey Point Units 3 and 4	1 of 39

TITLE	CALCULATIONS NO.
R. V. Internals Lift Rig, Load Cell & Linkage	PDC-

AUTHOR & DATE	CHECKED BY & DATE
J. Urban [signature] 12/82	J. Matusz [signature] 12/82

PURPOSE AND RESULTS:

1. The purpose of this analysis is to determine the acceptability of this rig to the requirements of ANSI N14.6.

2. The results show that all stresses are within the allowable stresses with the exception of the internals lifting rig engaging screw.

COMMONWEALTH OF PENNSYLVANIA
REGISTERED PROFESSIONAL
JOHN M. MATUSZ
ENGINEER
No. 14370-E

J M Matusz 12/82

REVISION NO.	DATE	DESCRIPTION	BY
		Original Issue	J. Urban

RESULTING REPORTS, LETTERS OR MEMORANDA:

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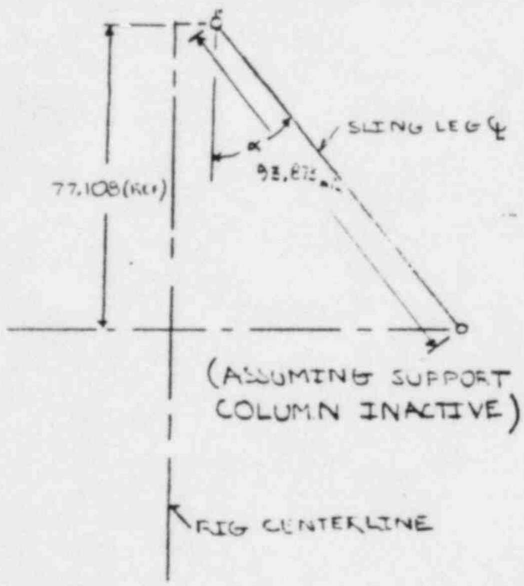
DESIGN WEIGHT

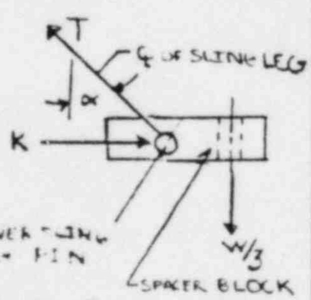
LIFT RIG (EST)	13,000
LOWER INTERNALS	236,000
CONTINGENCIES	11,000
DESIGN WEIGHT, W, =	260,000

SLING LEG ANGLE



$\cos\alpha = \frac{77.108}{93.873} = 34.774°$

SLING LEG TENSION FORCE, T, AND SPREADER COMPRESSIVE FORCE, K



$T\cos\alpha = W/3$

$T = W/3 / \cos 34.774°$

$T = W(.4058)$

$\tan\alpha = \frac{K}{(W/3)}$

$K = W/3 * \tan 34.774°$

$K = .(2314)W$

NOMENCLATURE

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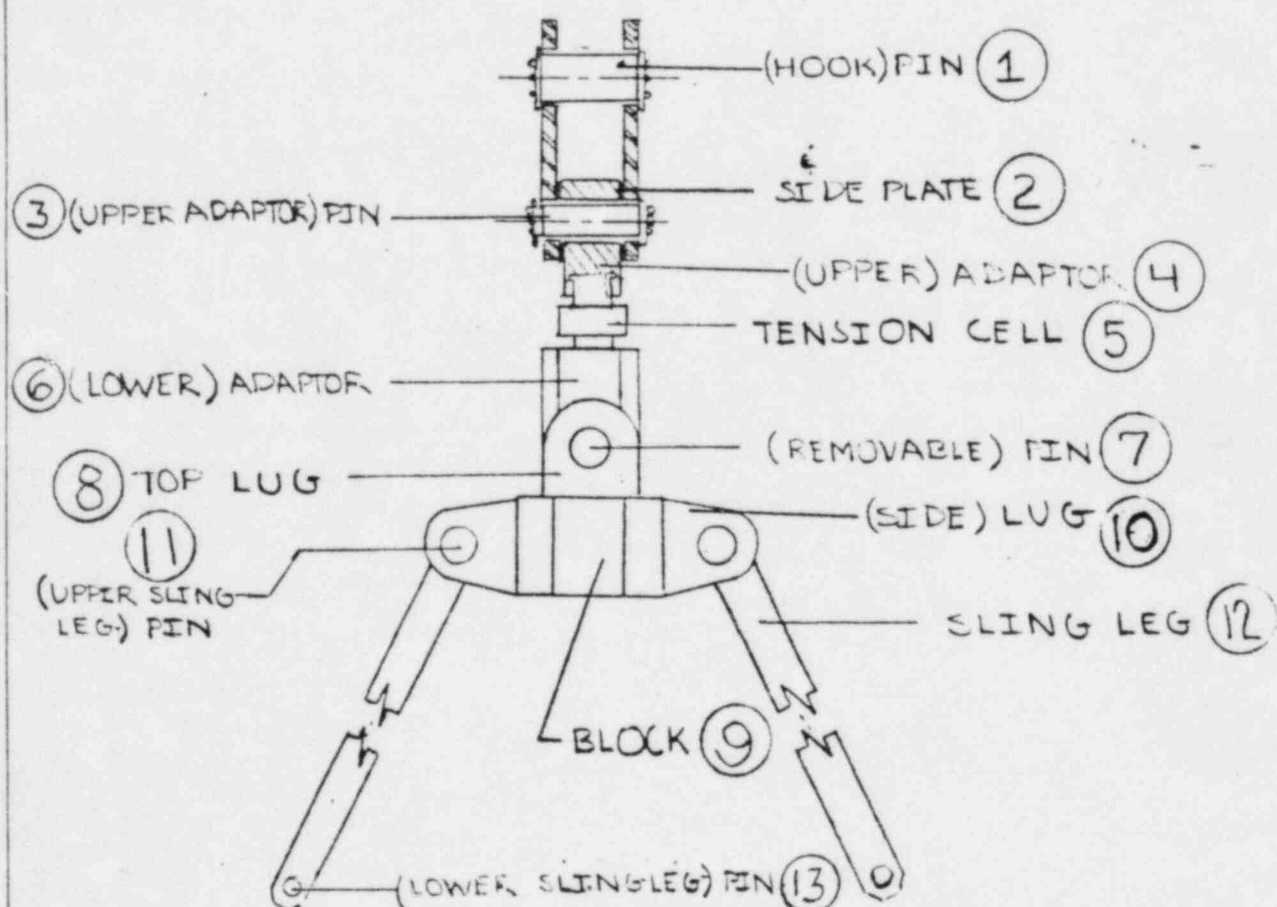
SLING ASSEMBLY



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FPL	[signature] 12/82	J M Martinez 12/82	

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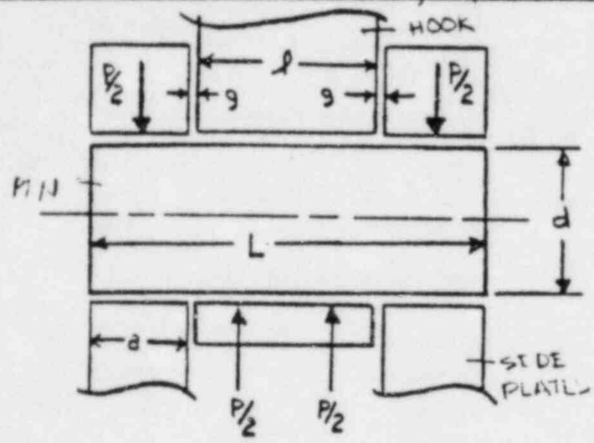
FJIP-93447			CHE

(HOOK) PIN ①

MAT'L: ASTM A434 CLASS BD
AISI 4340 HOT ROLLED, Q & T.



P = force acting on assembly, lb.
d = diameter of pin, in.
ℓ = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
= $\ell + 2(a + g)$

$P = W$ lb. = 260,000 LB.
$d = 7.455_{min}$ in.
$\ell_{nom} \cong 8$ in.
$a = 1.73$ min in
$g = .340$ in
[12.14 - 2(1.73) - 8nom]/2

SHEAR STRESS

$f_v = P/2A_v$
$P = W$
$A_v = \pi d^2/4$
$A_v = \pi (7.455)^2/4$
$A_v = 43.650 \text{ in}^2$

$f_v = (W)/(2 * 43.650)$
$= W(.011455)$
$f_v = 2978$ PSI

BEARING STRESS

$f_c = P/A_v$
$P = W$

INNER

$A_v = d\ell = (7.455)(8)$
$= (59.64) \text{ in}^2$
$f_c = (W)/(59.64)$
$= W(.016767)$
$f_c = 4359$ PSI

OUTER

$A_v = 2ad = 2(1.73)(7.455)$
$= (25.79) \text{ in}^2$
$f_c = (W)/(25.79)$
$= W(.03877)$
$f_c = 10,080$ PSI

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(HOOK)
PIN

(1)

BENDING STRESS(1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$

$$I = \pi d^4/64$$

$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$

$$= 16P\left(\frac{1}{3}a + g + \frac{1}{4}l\right)/(\pi d^3)$$

$$= W(16)\left(\frac{1.73}{3} + .340 + \frac{8}{4}\right)/(\pi * 7.455^3)$$

$$= W(.03585)$$

$$f_b = 9321 \text{ PSI}$$

(1) ADAPTED FROM FASTENING AND JOINING, 4th Ed, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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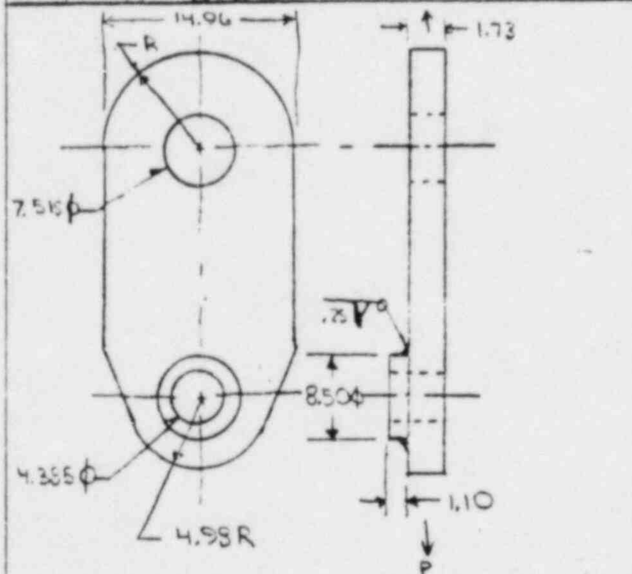
SIDE PLATE (2)

MAT'L: ASTM A-515 GR 70 OR ASTM A-516 GR 70 Q & T. CARBON CONTENT TO BE LESS THAN 0.30%



TENSION AT 7.515 Φ HOLE

$f_t = P/A_t$

$P = W/2$

$A_t = (14.96 - 7.515)(1.73) = 12.880\ in^2$

$f_t = W/2/12.880 = W(.03882)$

$f_t = 10{,}093$ PSI

BEARING AT 7.515 Φ HOLE

THE BEARING IS THE SAME AS THE OUTER BEARING ON THE HOOK PIN (ITEM 01)

$f_c = W(.03877)$

$f_c = 10{,}080$ PSI

SHEAR TEAR-OUT AT 7.515 Φ HOLE

$f_v = P/2A_v$

$P = W/2$

$A_v \cong (14.96 - 7.515)/2 * 1.73 = 6.440$

$f_v = W/2/2/6.440$

$f_v = W(.03882)$

$f_v = 10{,}093$ PSI

TENSION AT 4.385 Φ HOLE

$f_t = P/A_t = (W/2)/A_t$

$A_t = (8.50 - 4.385)(1.10) + (4.98(2) - 4.385)(1.73) = 14.171\ in^2$

$f_t = W/2/14.171 = W(.03528)$

$f_t = 9174$ PSI

SHEAR TEAR-OUT AT 4.385 Φ HOLE

$f_v = P/2A_v = (W/2)/2A_v$

$A_v \simeq (4.98 - 4.385/2) * 1.73 + (8.50 - 4.385)(1.10)/2 = 7.0855$

$f_v = W/2/2/7.0855 = W(.03528$

$f_v = 9174$ PSI

BEARING AT 4.385 Φ HOLE

THE BEARING IS THE SAME AS THE OUTER BEARING ON THE UPPER ADAPTOR PIN (ITEM 03)

$f_c = W(.04066) = 10{,}572$ PSI

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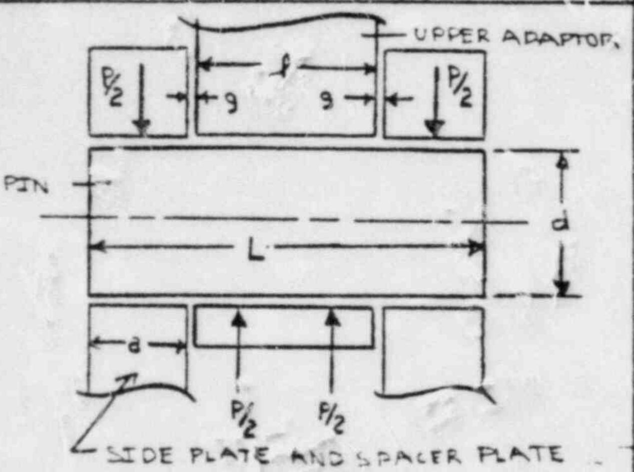
(UPPER ADAPTOR) PIN (3)

MAT'L: ASTM A-434 CLASS BD, OR AISI 4340 HOT ROLLED, QUENCHED & TEMPERED



P = force acting on assembly, lb.
d = diameter of pin, in.
l = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
= $l + 2(a+g)$

$P = W$ lb.
$d = 4.345_{min}$ in.
$l = 6.23_{min}$ in.
$a = 2.83_{min}$ in.
$g_{max} = .125$ in.
$= [12.14_{max} - 2(2.83) - 6.23]/2$

SHEAR STRESS

$f_v = P/2A_v$
$P = W$
$A_v = \pi d^2/4$
$A_v = \pi(4.345)^2/4$
$A_v = 14.828 \text{ in}^2$

$f_v = (W)/(2*14.828)$
$= W(.03372)$
$f_v = 8767$ PSI

BEARING STRESS

$f_c = P/A_v$
$P = W$

INNER

$A_v = dl = (4.345)(6.23)$
$= (27.069) \text{ in}^2$
$f_c = (W)/(27.069)$
$= W(.03694)$
$f_c = 9605$ PSI

OUTER

$A_v = 2ad = 2(2.83)(4.345)$
$= (24.59) \text{ in}^2$
$f_c = (W)/(24.59)$
$= W(.04066)$
$f_c = 10,572$ PSI

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(UPPER ADAPTOR)
PIN

③

BENDING STRESS (1)

$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$

$f_b = Mc/I$

$I = \pi d^4/64$

$c = d/2$

$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$

$= 16P\left(\frac{1}{3}a + g + \frac{1}{4}l\right)/(\pi d^3)$

$= W(16)\left(\frac{2.83}{3} + .125 + \frac{6.23}{4}\right)/(\pi * 4.345^3)$

$= W(.16303)$

$f_b = 42,388$ PSI

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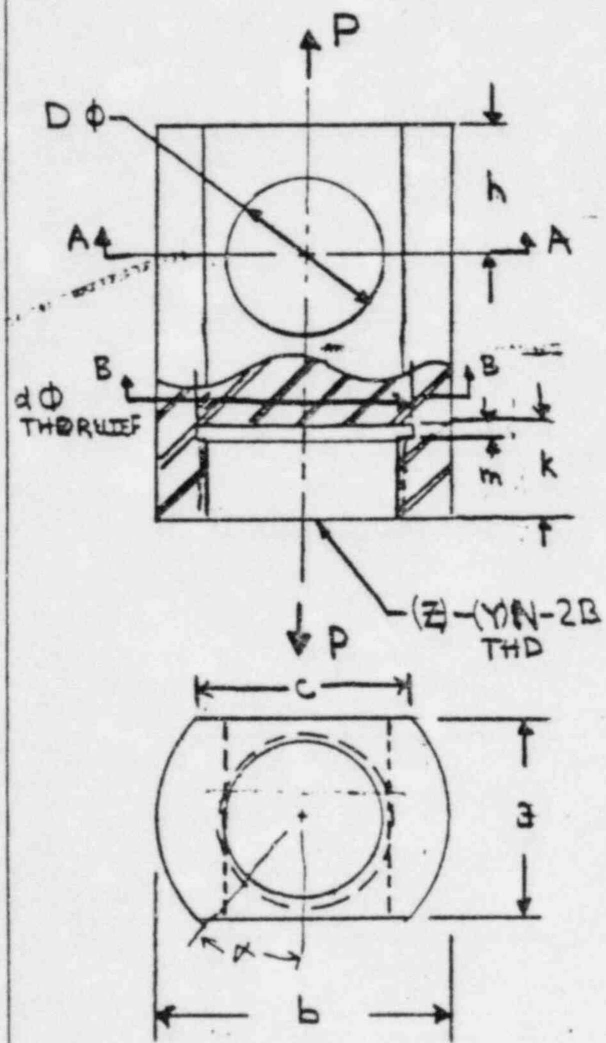
(UPPER AND LOWER) ADAPTOR ⑥ ④

MAT'L: ASTM A-540 GRADE B-24, $YIELD_{min} = 120{,}000$ PSI

$TENSILE_{min} = 135{,}000$ PSI



BODY AREA

$$A_B = \frac{a}{2}\sqrt{b^2 - a^2} + \pi b^2\left(\frac{360-4\alpha}{1440}\right)$$

WHERE

$\cos\alpha = a/b \rightarrow \alpha = 44.17°$

$A_B = 51.933\ in^2$

h =	4.48_{min}	in
a =	6.23_{min}	in
b =	$8.94\ \Phi_{min}$	in
(Z)-(Y)N-2B =	$4\frac{1}{4}$-8N-2B	in
k =	3.932_{min}	in
m =	$.39_{max}$	in
D =	4.387_{max}	in
d =	4.31_{max}	in
P = W =		lb

IS C < D ?

$$C = \sqrt{b^2 - a^2}$$
$$= \sqrt{(8.94)^2 - (6.23)^2}$$
$$= 6.412 \text{ in}$$
$$D = 4.387 \text{ in}$$

$\therefore$ C > D

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(UPPER AND LOWER) ADAPTOR ⑥ ④

TENSILE STRESS @ A-A

$f_t = P/A_t$

$P = W$

$A_t = A_B - A_{hole}$

$A_{hole} = Da$

$A_t = (51.933) - (4.387)(6.23)$

$= 24.602 \text{ in}^2$

$f_t = W/A_t$

$= W(.04065)$

$f_t = 10,569$ PSI

BEARING STRESS @ A-A

$f_c = P/A_c$

$P = W$

$A_c = aD$

$= (6.23)(4.387)$

$= 27.33 \text{ in}^2$

$f_c = W/A_c$

$= W(.03659)$

$f_c = 9513$ PSI

TENSILE STRESS @ B-B

$f_t = P/A_t = W/A_t$

$A_t = A_B - \pi d^2/4$

$= (51.933) - \frac{\pi}{4}(4.31)^2$

$= 37.34 \text{ in}^2$

$f_t = W(.02678)$

$f_t = 6963$ PSI

THREAD SHEAR

$f_v = P/A_v = W/A_v$

$A_v = \pi D_{pitch} * l/2$

$l = k - m$

$= (3.932) - (.39)$

$= 3.542$ in

$D_{pitch} = Z - .64952/N$

$= (4.1688)$ in.

$A_v = (23.194) \text{ in}^2$

$f_v = W/A_v$

$= W(.04311)$

$f_v = 11,209$ PSI

SHEAR TEAR-OUT*

$f_v = P/2A_v$

$A_v = (h - D/2)a$

$= (4.48 - 4.387/2)*(6.23)$

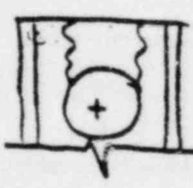
$= 14.245 \text{ in}^2$

$P = W$

$f_v = W/(2A_v)$

$= W(.035100)$

$f_v = 9126$ PSI

* 

LETTING A_v BE THE AREA OF THE CROSS-SECTION DIRECTLY ABOVE THE AXIS

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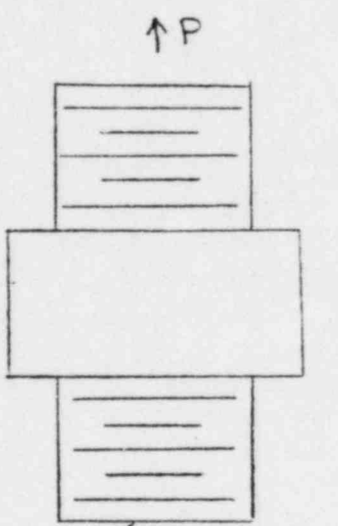
TENSION CELL (5)

MAT'L -
17-4 PH SS H 1100



TENSION AT THREADS

$$f_t = P/A_t = W/A_t$$

FROM MARKS HANDBOOK SEC 8-13

STRESS AREA: $A_t = 13.368 \; in^2$

$$f_t = W(.07480)$$

$$\underline{f_t = 19,448 \text{ PSI}}$$

THREAD SHEAR

THE THREAD SHEAR IS THE SAME AS THAT OF THE ADAPTORS (ITEMS 4 AND 5)

$$f_v = W(.04311)$$

$$\underline{f_v = 11,209 \text{ PSI}}$$

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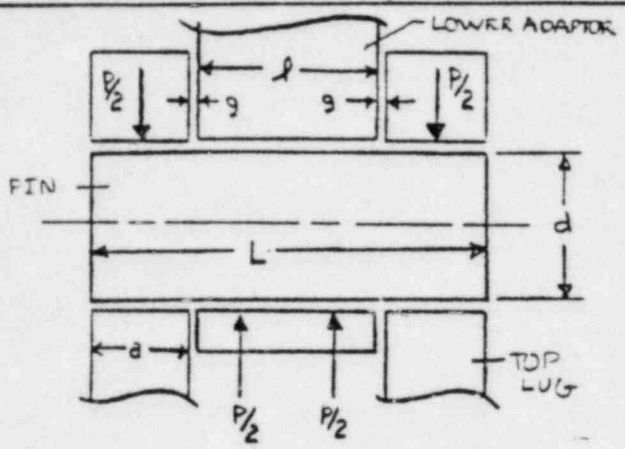
(REMOVABLE) PIN (7)

MAT'L: ASTM A434 CLASS BD
AISI 4340 HOT ROLLED, Q & T



P = force acting on assembly, lb.
d = diameter of pin, in.
ℓ = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
$= \ell + 2(a + g)$

P = W lb.
d = 4.345 min in.
ℓ = 6.23 min in.
a = 2.98 min in
g = 0.075 max in
$= (6.380 - 6.23)/2$

SHEAR STRESS

$f_v = P/2A_v$
$P = W$
$A_v = \pi d^2/4$
$A_v = \pi (4.345)^2/4$
$A_v = 14.828$ in²

$f_v = (W)/(2 * 14.828)$
$= W(.03372)$
$f_v = 8767$ PSI

BEARING STRESS -

$f_c = P/A_v$
$P = W$

INNER

$A_v = d\ell = (4.345)(6.23)$
$= (27.069)$ in²
$f_c = (W)/(27.069)$
$= W(.03694)$
$f_c = 9604$ PSI

OUTER

$A_v = 2ad = 2(2.98)(4.345)$
$= (25.896)$ in²
$f_c = (W)/(25.896)$
$= W(.03862)$
$f_c = 10{,}041$ PSI

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(REMOVABLE)
PIN (7)

BENDING STRESS (1)

$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}\ell\right)$

$f_b = Mc/I$

$I = \pi d^4/64$

$c = d/2$

$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}\ell\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$

$= 16P\left(\frac{1}{3}a + g + \frac{1}{4}\ell\right)/(\pi d^3)$

$= 16 * W * (2.98/3 + .075 + 6.23/4)/\pi/4.345^3$

$= W(.16303)$

$f_b = 42{,}388$ PSI

(1) ADAPTED FROM FASTENING AND JOINING, 4th Ed, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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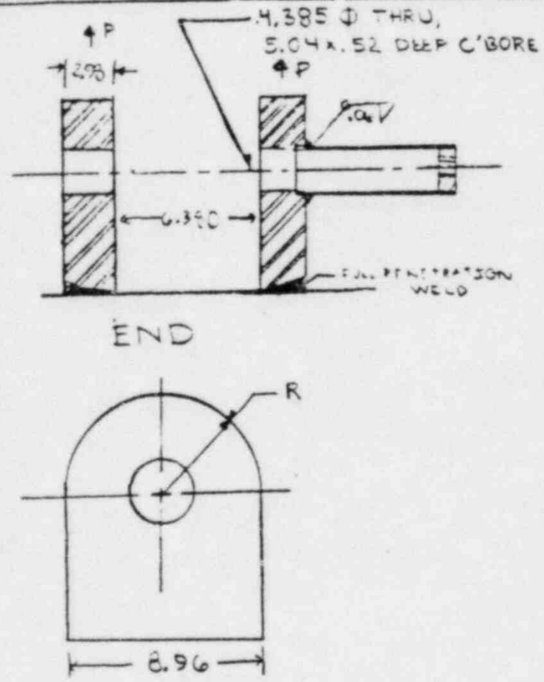
TOP LUGS (8)

MAT'L: ASTM A 515 GR. 70, OR ASTM A-516, GR. 70 Q & T. CARBON CONTENT TO BE LESS THAN 0.30%



BEARING ON PIN

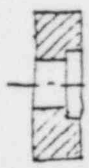
THE BEARING STRESS IS THE SAME AS THE OUTER BEARING STRESS ON THE REMOVABLE PIN (ITEM 7)

$f_c = W(.03862)$

$f_c = 10.041$ PSI

TENSION AT PIN-HOLE

IT IS CONSERVATIVE TO NEGLECT PIN HOUSING WELD & PIN SHROUD



$f_t = P/A_t$

$P = W/2$

$A_t = (8.96 - 4.385)(2.98 - .52) + (8.96 - 5.04)(.52)$

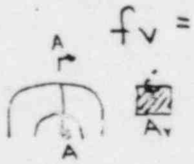
$= 13.293 \text{ in}^2$

$f_t = W/2/13.293$

$= W(.03761)$

$f_t = 9779$ PSI

SHEAR TEAR-OUT AT PIN HOLE *



$f_v = P/2A_v$

$P = W/2$

$A_v = [(8.96 - 4.385)(2.98 - .52) + (8.96 - 5.04)(.52)]/2$

$2A_v = 13.293 \text{ in}^2$

$f_v = W/2/2/A_v$

$= W(.03761)$

$f_v = 9779$ PSI

* ASSUMING EQUAL LOAD ON LUGS

TENSION AT WELD

$f_t = P/A_t$

$P = W/2$

$A_t = 8.96 * 2.98 = 26.70 \text{ in}^2$

$f_t = W/2/26.70$

$= W(.018726)$

$f_t = 4869$ PSI

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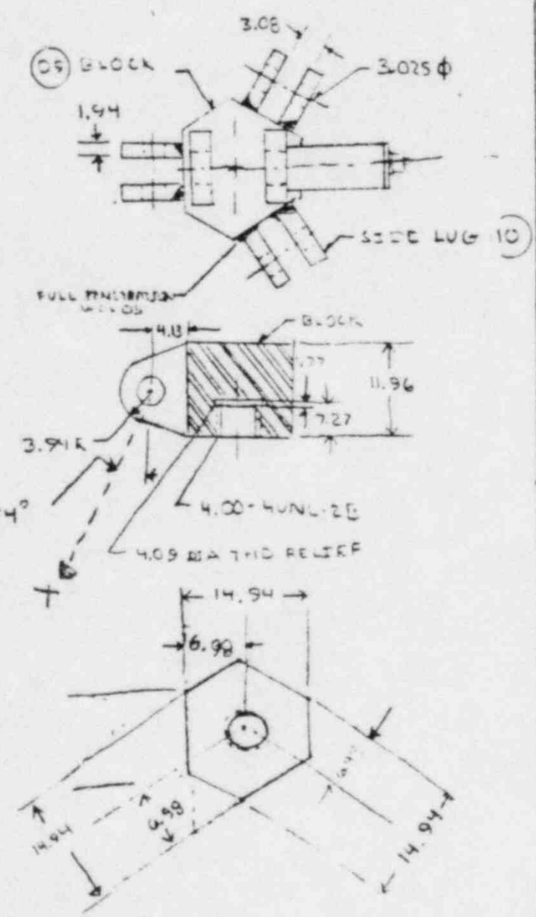
BLOCK AND (9)
SIDE LUGS (10)

BLOCK: SA 105 CL 1 OR 2, CARBON CONTENT TO BE LESS THAN 0.30 %, ALT. MAT'LS - ASTM A-266 CL 1 OR 2 OR SA 266 CL 1 OR 2, CARBON CONTENT TO BE LESS THAN 0.30%

SIDE LUG: ASTM A515 GR 70 OR ASTM A-516 GR 70. Q & T. CARBON CONTENT TO BE LESS THAN 0.35%



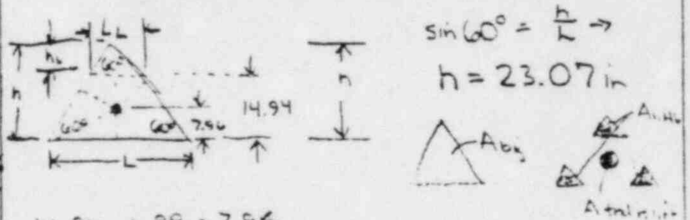
BLOCK (09)

TENSION IN BLOCK

$f_t = P/A_t = W/A_t$

$A_{t\,min}$:

$\tan 30° = \frac{7.69}{(L/2)} \rightarrow L = 26.639\text{ in}$



$\sin 60° = \frac{h}{L} \rightarrow$

$h = 23.07\text{ in}$

$14.94 - 6.98 = 7.96$

$A_{big} = \frac{1}{2} hL = (\frac{1}{2}) 26.639 (23.07)$

$= 307.29\text{ in}^2$

$h_{little} = h - 14.94 = 8.13\text{ in}$

$L_{little} = \frac{8.13}{23.07}(26.639) = 9.388\text{ in}$

$A_{little} = (\frac{1}{2})(8.13)(9.388) = 38.16\text{ in}^2$

$A_{t\,min} = A_{big} - 3A_{little} - \frac{\pi}{4}(4.09^2)$

$= 179.65\text{ in}^2$

$f_t = W/179.65$

$= W(.005566)$

$f_t = 1447$ PSI

SIDE LUGS (10)

TENSION AT HOLE

$f_t = P/A_t$

$P = T/2, \quad \cos\alpha = \frac{(W/3)}{T}$

$\alpha = 34.774°, \rightarrow T = .4058\,W$

$A_t = [3.94(2) - 3.025] \cdot 1.94 = 9.419\text{ in}^2$

$f_t = .4058W/2/9.419$

$= W(.02154)$

$f_t = 5600$ PSI

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SIDE LUG (10)

SHEAR TEAR-OUT AT PIN HOLE

$f_v = P/2A_v$

$P = T/2 = .4058W/2$

$A_v = (3.94 - 3.025/2) * 1.94$

$= 4.709 in^2$

$f_v = .4058W/2/2/4.709$

$= W(.02154)$

$f_v = 5600$ PSI

BEARING ON PIN

THE BEARING STRESS IS THE SAME AS THE OUTER BEARING STRESS ON THE UPPER SLING-LEG-PIN (ITEM 11)

$f_c = W(.03492)$

$f_c = 9079$ PSI

SHEAR AT F.P. WELD

$f_v = P/A_v$

$P = (W/3)/2$

$A_v = 11.96 * 1.94$

$= 23.202 in^2$

$f_v = W/3/2/23.202$

$f_v = W(.007183)$

$f_v = 1868$ PSI

COMBINED TENSION AND BENDING AT F.P. WELD

TENSION:

$f_t = P/A_t$

$P = T(\sin 34.774°)/2$

$A_t = 11.96 * 1.94 = 23.202 in^2$

$f_t = T * (\sin 34.774°)/2/23.202$

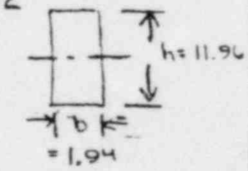
$= T(.012291) = .4058W(.012291)$

$= W(.004988)$

$f_t = 1297$ PSI

MAXIMUM BENDING:

$f_{b_{max}} = Mc/I; c = \frac{h}{2}$



$I_x = bh^3/12$

$c = h/2$

$M = P\ell$

$P = (W/3)/2$

$\ell = 4.13$ in

$f_{b_{max}} = (W/3/2) * 4.13 * \frac{11.96}{2} / (1.94 * 11.96^3/12)$

$= W(.014883)$

$f_{b_{max}} = 3870$ PSI

COMBINED:

$f_{t_{max}} = f_{b_{max}} + f_t$

$= W(.014883 + .004988)$

$= W(.019871)$

$f_{t_{max}} = 5166$ PSI

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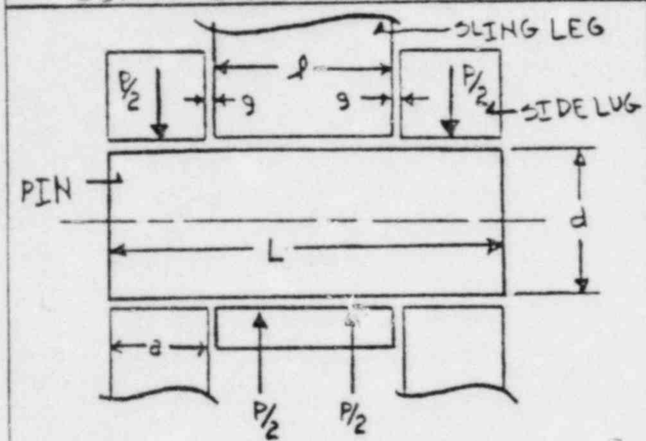
(UPPER SLING-LEG-)PIN (11)

MAT'L: ASTM A 276 TYPE 304 CENT. GRD. COND. A



P = force acting on assembly, lb.
d = diameter of pin, in.
l = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
$= l + 2(a+g)$

$P = T = W(.4058)$ lb.
$d = 2.995$ in.
$l = 2.98$ in
$a = 1.94$ in
$g = 0.05$ in
$= (3.08 - 2.98)/2$

SHEAR STRESS

$f_v = P/2A_v$
$P = T = .4058\,W$
$A_v = \pi d^2/4$
$A_v = \pi(2.995)^2/4$
$A_v = 7.0450\ in^2$

$f_v = (.4058\,W)/(2*7.0450)$
$= W(.028800)$
$f_v = 7488$ PSI

BEARING STRESS

$f_c = P/A_v$
$P = T = .4058\,W$

INNER

$A_v = d\,l = (2.995)(2.98)$
$= (8.9251)\ in^2$
$f_c = (.4058\,W)/(8.9251)$
$= W(.04547)$
$f_c = 11822$ PSI

OUTER

$A_v = 2ad = 2(1.94)(2.995)$
$= (11.621)\ in^2$
$f_c = (W*.4058)/(11.621)$
$= W(.03492)$
$f_c = 9079$ PSI

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(UPPER SLING-LEG-) PIN (11)

BENDING STRESS (1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$

$$I = \pi d^4/64$$

$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$

$$= 16P\left(\tfrac{1}{3}a + g + \tfrac{1}{4}l\right)/(\pi d^3)$$

$$= 16(.4058W) * \left(\frac{1.94}{3} + .05 + 2.98/4\right)/(\pi * 2.995^3)$$

$$= W(.11091)$$

$$f_b = 28,837 \text{ PSI}$$

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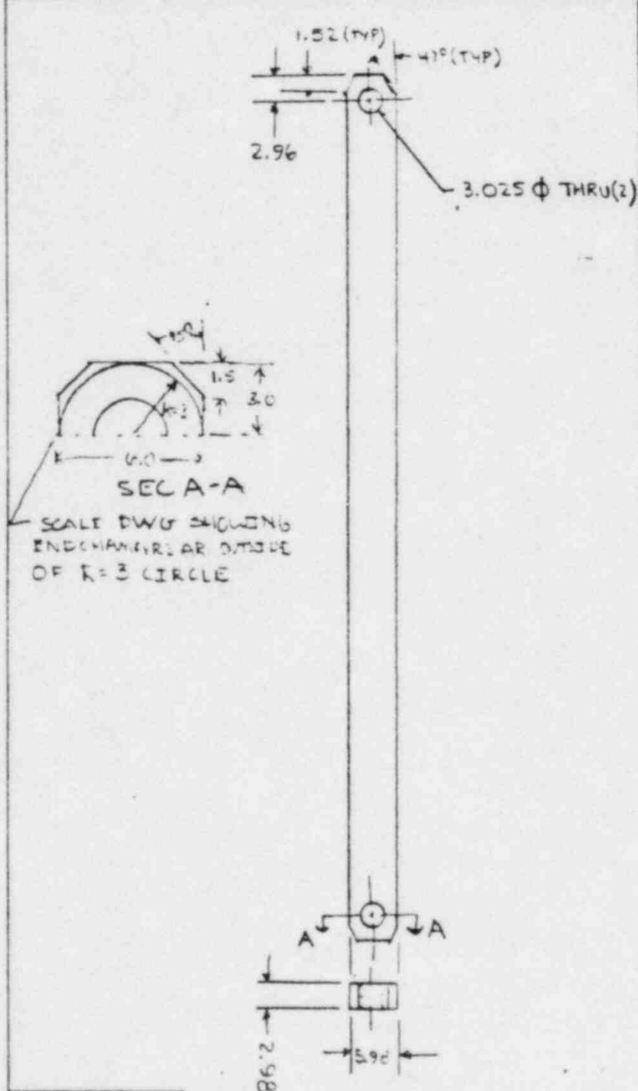
SLING LEG[1] (12)

MAT'L:
ASTM A-515 GR 70 OR
ASTM A-516 GR 70, Q & T.
$C_{12} < 0.30\%$



(1) ASSUMING SUPPORT COLUMN IS NOT ACTIVE

BEARING ON PINS

THE BEARING STRESS IS THE SAME AS THE INNER BEARING ON THE SLING-LEG-PINS (ITEMS 11 AND 13)

$f_c = W(.04547)$

$f_c = 11,822$ PSI

TENSION IN LEG

$f_t = P/A_t = T/A_t = .4058W/A_t$

$A_t = 2.98 * 5.98 = 17.820\ in^2$

$f_t = .4058W/17.820 = W(.02277)$

$f_t = 5920$ PSI

TENSION AT PIN-HOLE

$f_t = P/A_t = T/A_t = .4058W/A_t$

$A_t = (5.98 - 3.025) * 2.98 = 8.8059\ in^2$

$f_t = .4058W/8.8059 = W(.04608)$

$f_t = 11,981$ PSI

SHEAR TEAR-OUT AT PIN-HOLE

$f_v = P/2A_v = T/2A_v = .4058W/2A_v$

$A_v \cong (5.98 - 3.025) * 2.98/2 = 4.403\ in^2$

$f_v = .4058W/2/4.403 = W(.04608)$

$f_v = 11,981$ PSI

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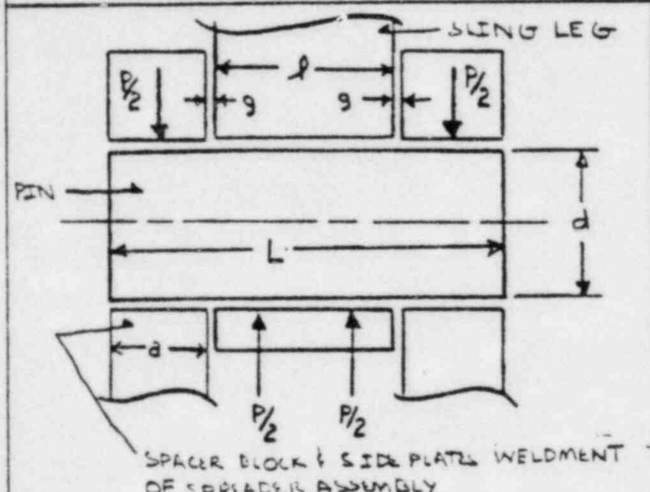
(LOWER SLING-LEG-)PIN (13)

MAT'L: ASTM A-276 TYPE 304
CENT. GRD. COND. A.



P = force acting on assembly, lb.
d = diameter of pin, in.
ℓ = length of bearing surface of center body, in.
a = length of bearing surface one side of outer body, in
g = gap between bearing surfaces, in.
L = total active length of pin, in
$= \ell + 2(a+g)$

$P = T = .4058W$ lb.
$d = 2.995$ in.
$\ell = 2.98$ in.
$a = 2.91$ in $= [5.98 + 4(.73) - 3.05]/2$
$g = .05$ in $= (3.08 - 2.98)/2$

SHEAR STRESS

$f_v = P/2A_v$
$P = T = .4058W$
$A_v = \pi d^2/4$
$A_v = \pi(2.995)^2/4$
$A_v = 7.0450$ in²

$f_v = (.4058W)/(2 * 7.0450)$
$= W(.02880)$
$f_v = 7488$ PSI

BEARING STRESS

$f_c = P/A_v$
$P = T = .4058W$

INNER

$A_v = d\ell = (2.995)(2.98)$
$= (8.9251)$ in²
$f_c = (.4058W)/(8.9251)$
$= W(.04547)$
$f_c = 11,822$ PSI

OUTER

$A_v = 2ad = 2(2.91)(2.995)$
$= (17.431)$ in²
$f_c = (.4058W)/(17.431)$
$= W(.02328)$
$f_c = 6053$ PSI

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(LOWER SLING-LEG-) PIN (13)

BENDING STRESS (1)

$$M_{max} = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)$$

$$f_b = Mc/I$$

$$I = \pi d^4/64$$

$$c = d/2$$

$$f_b = \frac{P}{2}\left(\frac{1}{3}a + g + \frac{1}{4}l\right)\left(\frac{d}{2}\right)\left(\frac{64}{\pi d^4}\right)$$

$$= 16P\left(\frac{1}{3}a + g + \frac{1}{4}l\right)/(\pi d^3)$$

$$= 16 * W * .4058 * (2.91/3 + .05 + 2.98/4)/(\pi * 2.995^3)$$

$$f_b = W(.13578)$$

$$f_b = 35{,}303 \text{ PSI}$$

(1) ADAPTED FROM FASTENING AND JOINING, 4th Ed, A REFERENCE ISSUE OF MACHINE DESIGN, PENTON PUBLISHERS PAGE 27

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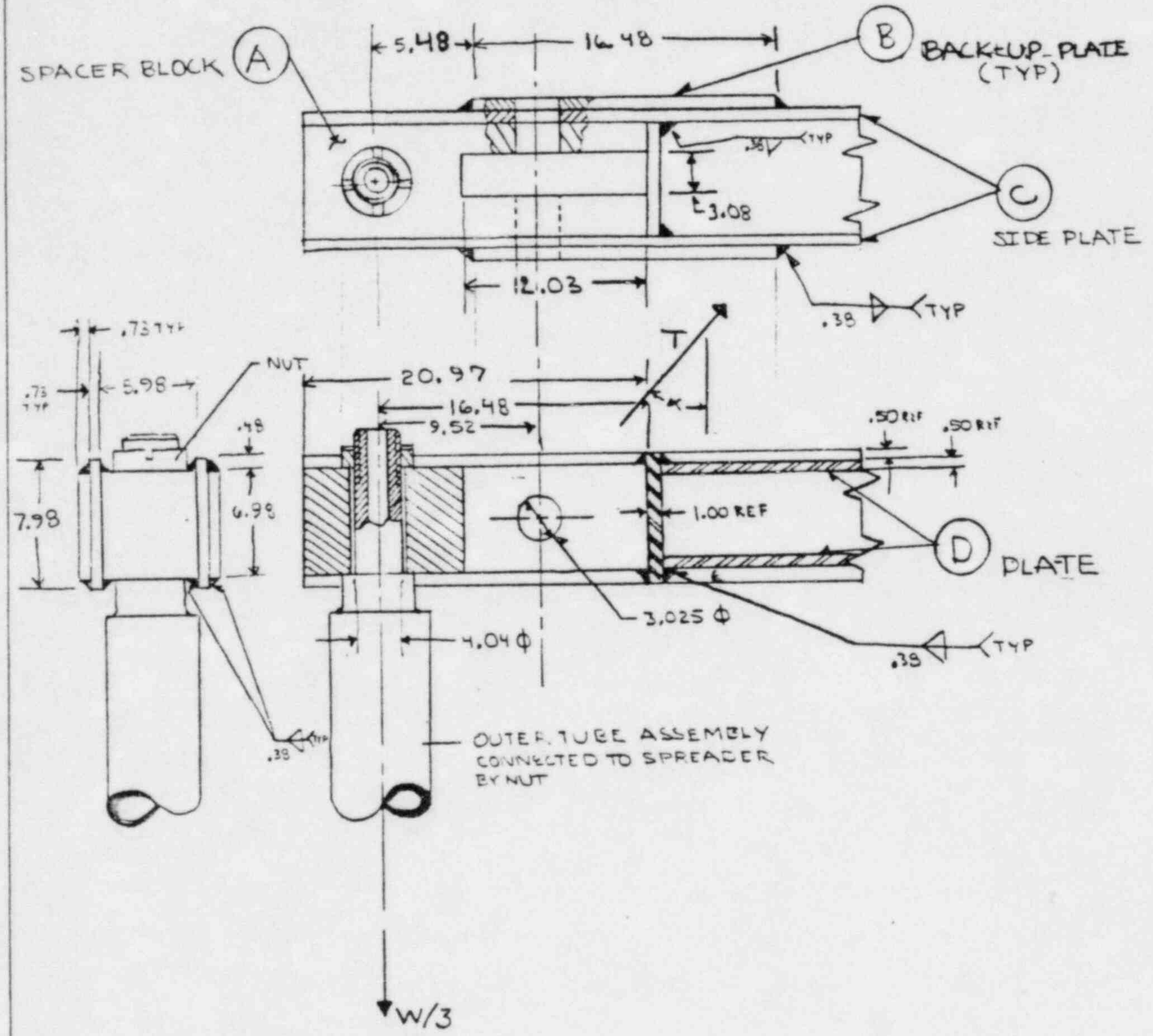
SPREADER ASSEMBLY (14)



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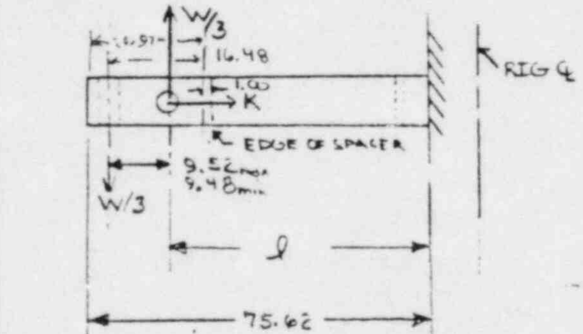
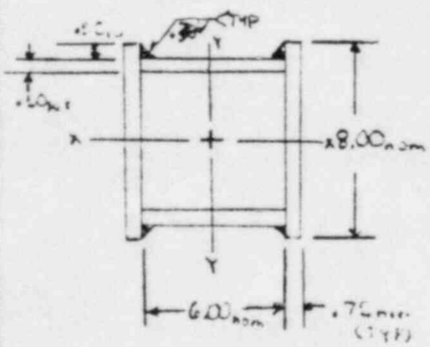
COMPARISON OF SPREADER ARM STRESSES TO AISC CRITERIA

THE SPREADER ARMS ARE SUBJECTED TO BENDING AND COMPRESSION



$\ell = 75.62 - 20.97 + 16.52 - 9.48$

$= 61.69$ in



REFERENCE:

MANUAL OF STEEL CONSTRUCTION

7th EDITION, AISC

MAT'L:

14-B,C,D

ASTM A515 GR 70
OR
ASTM A516 GR 70

CHAPTER 5, SECTION 1.6, PAGE 5-22

MEMBERS SUBJECTED TO BOTH AXIAL COMPRESSION AND BENDING STRESSES SHALL BE PROPORTIONED TO SATISFY THE FOLLOWING REQUIREMENTS:

$$\frac{f_a}{F_a} + \frac{C_{mx} f_{bx}}{\left(1 - \frac{f_a}{F'_{ex}}\right) F_{bx}} + \frac{C_{my} f_{by}}{\left(1 - \frac{f_a}{F'_{ey}}\right) F_{by}} \leq 1.0$$

AND

$$\frac{f_a}{0.60 F_y} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1.0$$

OR IN LIEU OF WHEN f_a/F_a [illegible]

$$\frac{f_a}{F_a} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}} \leq 1.0$$

F_a = axial stress that would be permitted if axial force alone existed

F_b = compressive bending stress that would be permitted if bending moment alone existed.

$F'_e = 12\pi^2 E / 23 (k\ell/r)^2$

where k, ℓ, and r are for the appropriate plane of bending

f_a = computed axial stress

f_b = computed compressive bending stress at the point under consideration.

C_m = a coefficient (see page 5-[illegible] and 5-23)

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COMPARISON OF SPREADER ARM STRESSES TO AISC CRITERIA

DETERMINING f_a

$f_a = K/A_t = .2314W/A_t$

$A_t = 2*.50*6.00+2*.75*8.00 + 4*.38*.38/2$

$A_t = 18.29\ in^2$

$f_a = .2314W/18.29 = W(.012653)$

$f_a = 3,290$ PSI

DETERMINING F_a

SECTION 1.5.1.3

DETERMINING C_c

$C_c = \sqrt{2\pi^2 E/F_y}$

$= (2\pi^2\ 30,000,000/38,000)^{1/2}$

$= 124.93$

DETERMINING $K\ell/r$

$\ell = 61.69$ in

DETERMINING K'S (page 5-138)

THE INNER END IS FIXED, AND HELD FROM TRANSLATION BY THE CENTRAL COLUMN. RELATIVELY CLOSE TO THE COLUMN END IS THE LOWER SLING LEG PIN. THE SPREADER MAY ROTATE ABOUT THIS PIN, BUT THE PIN IS HELD FROM TRANSLATING BY THE SLING LEG. ASSUMING THE VERTICAL TO BE RELATIVELY LONG AND FLEXIBLE, AND SO THEREFORE INCAPABLE OF EXERTING AN INFLUENTIAL MOMENT, K THEN IS

$K_x = 0.80$ (case b)

IN THE HORIZONTAL PLANE, THE INNER END IS AGAIN FIXED, BUT THE OUTER END MAY TRANSLATE AND ROTATE. THE THEORETICAL K VALUE IS 2.0. THE AISC RECOMMENDED VALUE IS

$K_y = 2.1$

DETERMINING RADII OF GYRATION

FOR A RECTANGLE $I_G = bh^3/12$

FOR A TRIANGLE $I_G = bh^3/36$

ALSO, $I = I_G + Ad^2$

$I_{x-x} = 2(.75*8.00^3/12)$
$+ 2(6.00*.50^3/12 + .50(6.00)(3.25)^2)$
$+ 4(.38*.38^3/36 + \frac{.38^2}{2}(3.50+\frac{.38}{3})^2)$

$I_{x-x} = 131.30\ in^4$

$r_{x-x} = \sqrt{I_{x-x}/A}$

$= (131.30/18.29)^{1/2}$

$r_{x-x} = 2.679$ in

$I_{y-y} = 2(.50*6.00^3/12)$
$+ 2(8.00(.75)^3/12 + .75(8.00)(3.00-\frac{.75}{2})^2)$
$+ 4(.38*.38^3/36 + \frac{.38^2}{2}(3.00-\frac{.38}{3})^2)$

$I_{y-y} = 157.64\ in^4$

$r_{y-y} = \sqrt{I_{y-y}/A}$

$= (157.64/18.29)^{1/2}$

$r_{y-y} = 2.936$ in

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COMPARISON OF SPREADER ARM STRESSES TO AISC CRITERIA

THEREFORE

$(K\ell/r)_x = 0.80(61.69)/2.679$
$= 18.422$

$(K\ell/r)_y = 2.1(61.69)/2.936$
$= 44.124$

FOR BOTH AXIS $K\ell/r < C_c$
SO EQUATION 5-1.5-1
GOVERNS (page 5-16)

$F_a = (1 - \frac{1}{2}N^2)F_y/(\frac{5}{3} + \frac{3N}{8} - \frac{N^3}{8})$

WHERE $N = (K\ell/r)/C_c$

$N_x = 18.422/124.83$
$= .14758$

$N_y = 44.124/124.83$
$= .35347$

$\therefore F_{a_x} = 21{,}832$ PSI
$F_{a_y} = 19{,}862$ PSI

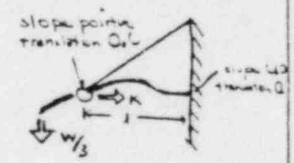
$F_a = F_{a_{min}} = F_{a_y}$
$F_a = 19{,}862$ PSI

DETERMINING IF f_a/F_a IS BELOW .15

$f_a/F_a = 3{,}290/19{,}862$
$f_a/F_a = .16564 \not\leq .15$

$\therefore$ THE FIRST SET OF EQUATIONS (page 5-22) GOVERN AND C_m MUST BE DETERMINED.

SINCE $f_{by} = 0$ ONLY C_{mx} MUST BE DETERMINED. IN THE VERTICAL PLANE, AS DETERMINED IN THE SELECTION OF K_x, JOINT TRANSLATION IS PREVENTED. THE LOADING CONDITION THEREFORE CONFORMS TO CATEGORY B (PAGE 5-131) AND $M_1 = M_2$ (SEE PAGE 5-132 FOR SIGN CONVENTION) AND WOULD PRODUCE REVERSE CURVATURE



SO THAT
$C_m = .6 - .4$ OR
.4, WHICHEVER IS LESS
$\therefore C_m = .4$

DETERMINING f_{b_x}

$f_{bx} = Mc/I_{x\text{-}x}$
$M = \frac{W}{3}(9.52) = 825{,}067$ in-lb
$c = 8.0/2 = 4$ in
$f_{bx} = 825{,}067 * 4/131.30$
$f_{bx} = 25{,}135$ PSI

DETERMINING F_b

$F_b = .60F_y$ (page 5-17)
$= .60 * 38{,}000$
$F_b = 22{,}800$ PSI

DETERMINING F'_{ex}

$F'_{ex} = 12\pi^2 E/23/(K\ell/r)^2$
$= 12\pi^2 30{,}000{,}000/23/18.442^2$
$F'_{ex} = 455{,}199$

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COMPARISON OF SPREADER ARM STRESSES TO AISC CRITERIA

CONDITION (1)

$$\frac{f_a}{F_a} + \frac{C_{mx} f_{bx}}{\left(1-\frac{f_a}{F'_{ex}}\right)F_{bx}} + \frac{C_{my} f_{by}}{\left(1-\frac{f_a}{F'_{ey}}\right)F_{by}}$$

BECOMES

$$\frac{3,290}{79,862} + \frac{.4(25,135)}{\left(1-\frac{3290}{455,199}\right)22,800} + 0$$

$$= .6098 \leq 1$$

CONDITION (2)

$$\frac{f_a}{.60F_y} + \frac{f_{bx}}{F_{bx}} + \frac{f_{by}}{F_{by}}$$

BECOMES

$$\frac{3,290}{.60(38,000)} + \frac{25,135}{22,800} + 0$$

$$= 1.247 \nleq 1$$

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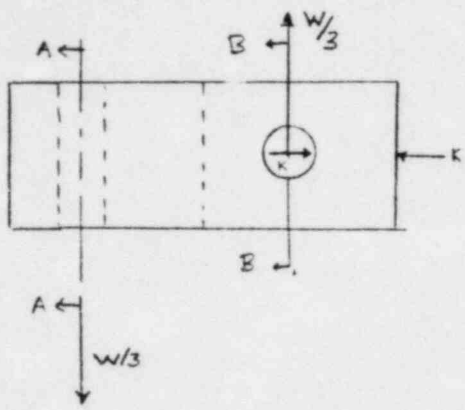
SPREADER SPACER BLOCK



SPACER BLOCK:

SA-105 CL 1 OR 2 QUENCHED & TEMPERED.

$C_{12} < .30\%$

OR

ASTM A-266 CL 1 OR 2

OR

SA-508 CL 1 OR 2, Q & T. $C_{12} < .30\%$

BENDING STRESS (1)

$f_b = Mc/I$

$I_{B-B} = 177.28$

$M = \frac{W}{3}(9.52) = 825{,}067 \text{ in-lb}$

$c = 3.50$

$f_{bmax} = (W/3) * 9.52 * 3.50 / 177.28$

$f_{bmax} = 16{,}289$ PSI

bending conservative, because V not completely applied at ℄ of hz

SHEAR

MINIMUM CROSS-SECTION OCCURS AT B-B (2)

$A_{B-B} = 25.14 \text{ in}^2$

$f_v = P/A_v$

$= (W/3)/25.14$

$f_v = 3{,}447$ PSI

BEARING

THE BEARING IS THE SAME FOR THE SPREADER AS FOR THE OUTER BEARING ON THE LOWER SLING-LEG PIN (ITEM B)

$f_c = W(.02328)$

$f_c = 6053$ PSI

(1) EXAMINING THE SHEAR-MOMENT DIAGRAM ON PAGE 29, IT CAN BE SEEN BY INSPECTION THAT FOR ANY CROSS-SECTION WHERE A MOMENT ACTS, THE SMALLEST MOMENT OF INERTIA IS AT CROSS-SECTION B-B. I_{B-B} IS CALCULATED ON PAGE 32

(2) SKETCH SHOWN ON PAGE 32

$A_{BB} = [6.00 - 3.06 + 2(.75)] * 7.00$

$+ 8 * [2 * .75]$

$- 3.02 * (6 - 3.06 + 2(.75) + 2(.75))$

$= 25.14 \text{ in}^2$

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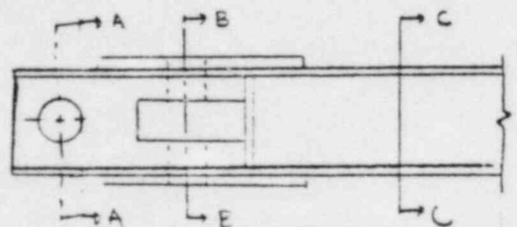
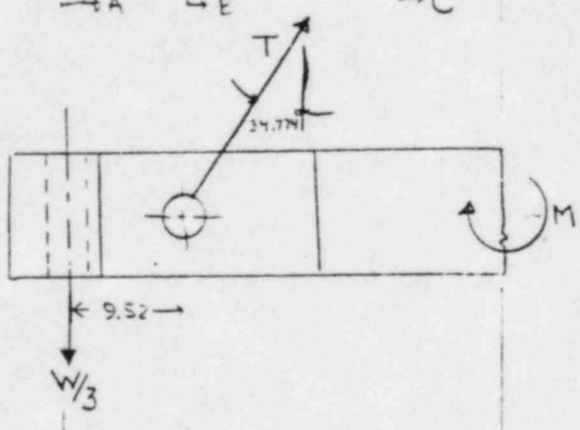
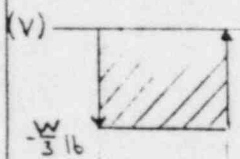
SPREADER WELDS

LOADS ON SPREADER ARM

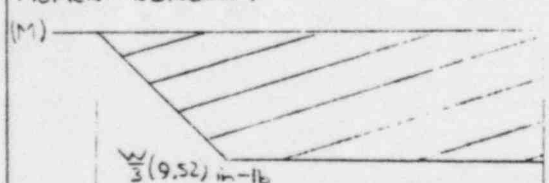




SHEAR DIAGRAM

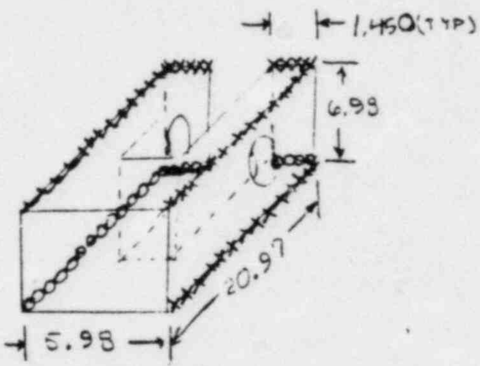


MOMENT DIAGRAM



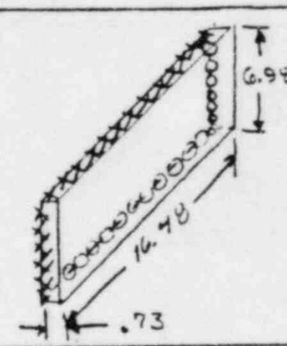
WELD GEOMETRIES

SPACER BLOCK TO SIDE PLATES

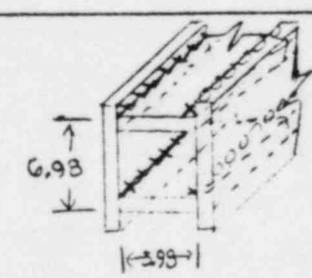


5.98 - 3.08 = 2.90

BACK-UP PLATE TO SIDE PLATES



PLATES TO SIDE PLATES



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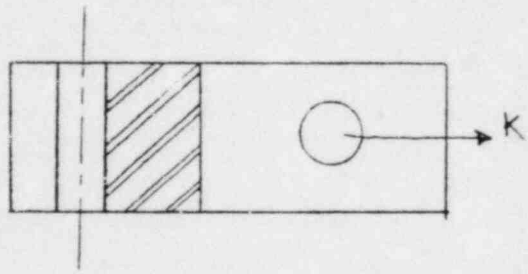
WELD STRESS DUE TO SHEAR

CONSIDER SPACER BLOCK LOADED BY SHEAR FORCE K



$f_v = P/A_{v(welds)} = K/A_{v(welds)}$

$A_v = 4(.3896 + 5.634)$

$= 24.09 \text{ in}^2$

$f_v = K/24.09 = .2314W/24.09$

$= (.00960)W$

$\underline{\underline{f_v = 2497 \text{ PSI}}}$

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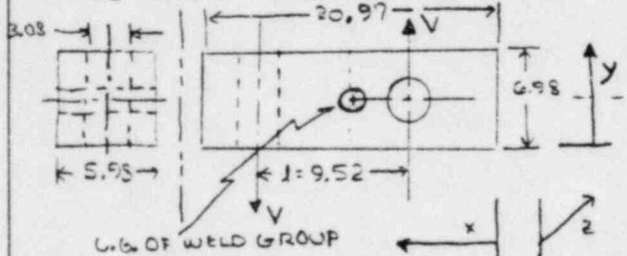
WELD STRESS DUE TO TWISTING[1]

CONSIDER SPACER BLOCK LOADED BY COUPLE $M = V\ell$



$V = W/3$

$M = W/3 \times 9.52 = 825{,}067$ in-lb

CENTER OF GRAVITY OF WELD GROUP:

FROM SYMMETRY OF THE WELD GROUP

$\bar{y} = 6.98/2 = 3.49$ in, $\bar{z} = 5.98/2 = 2.99$ in

$\bar{x} = \sum A_i \bar{x}_i / \sum A_i$ (SPOTTS 9)

$A_1 = 1.450(.38 \times .707) = .3896$ in^2

$x_1 = 0$ in

$A_2 = 20.97(.38 \times .707) = 5.634$ in^2

$x_2 = 20.97/2 = 10.485$ in

4 TYPE 1 WELDS, 4 TYPE 2 WELDS

$$\bar{x} = \frac{4(.3896)(0) + 4(5.634)(10.485)}{4(.3896 + 5.634)}$$

$\bar{x} = 9.807$ in

$J = J_o + A r_1^2$

r_1 extends from the center of gravity of the weld to the center of gravity of the group

$J_o = A\ell^2/12$

$J_{o(1)} = .3896(1.450)^2/12 = .06826$ in^4

$r_1 = (9.807^2 + 3.49^2)^{1/2} = 10.409$ in

WHERE FROM PHYSICAL CONSIDERATIONS IT IS ASSUMED THAT THE STRESS IS CONSTANT IN THE Z DIRECTION

$J_{(1)\Sigma} = 4[.06826 + .3896 \times 10.409^2] = 169.12$ in^4

$J_{o(2)} = 5.634(20.97)^2/12 = 206.4$ in^4

TAKING ALL TYPE 2 WELDS TOGETHER

$r_{2\Sigma} = ((10.485 - 9.807)^2 + 3.49^2)^{1/2} = 3.555$

$J_{(2)\Sigma} = 4(206.4) + 4(5.634)(3.555)^2 \cong 1110.4$ in^4

$J_\Sigma = 169.12 + 1110.4 = 1279.5$

$f_v = Mr/J = 825{,}067(r)/1279.5 = 644.8r$ psi

FROM GEOMETRY IT CAN BE SEEN THAT STRESS DOES NOT VARY IN THE Z DIRECTION SO

$r_{max} = (3.49^2 + (20.97 - 9.807)^2)^{1/2} = 11.70$ in

$f_{v_{max}} = 644.8(11.70)$ 7,545 psi

THIS STRESS OCCURS ONLY IN THE FAR END CORNERS.

(1) REFERENCE CHAPTER 7 OF SPOTTS' DESIGN OF MACHINE ELEMENTS 5th Edition

$.786 \times 202 / 260 = 7603 \approx 7619$

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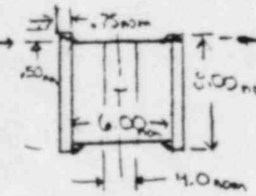
WELD STRESS DUE TO SHEAR FLOW

THE WELDS ALSO SEE SECONDARY STRESSES DUE TO SHEAR FLOW (1)

$f_v = Vay/It$

V = external vertical shear on beam
I = moment of inertia of entire section
t = thickness of section at plane where stress is desired
a = area of section beyond plane where stress is desired
y = distance of center of gravity of area to neutral axis of entire section

AT SEC A-A V = W/3
B-B V = W/3
C-C V = 0

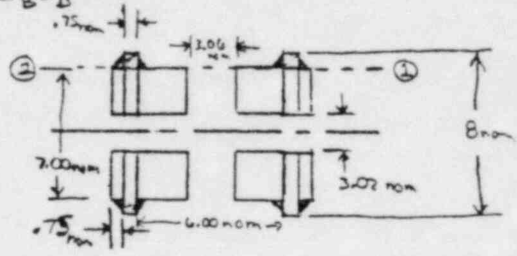


$I_{A-A} = 6 \cdot 7^3/12 + 2(.75)(8^3/12) - 4(7^3)/12$

$I_{A-A} = 121.17 \text{ in}^4$

(1) REF. DESIGN OF WELDMENTS
LINCOLN ARC WELDING FOUNDATION
PAGE 2.6-3

I_{B-B}



$I_{B-B} = [6 + 2(.75)]\, 7^3/12 + 2(.75)\, 8^3/12 - 3.06(7.00^3/12) - (6 + 4(.75) - 3.06)(3.02)^3/12 = 177.28 \text{ in}^4$

AT AXIS-1-1

$t_H = 2(.75) = 1.50 \text{ in}$

$a_H = 2(.75)(.50) = .75 \text{ in}^2$

$y = 3.5 + .50/2 = 3.75 \text{ in}$

$f_{v_{A-A}} = (W/3) * .75 * 3.75/121.17/1.50 = W(.005158)$
1,341 PSI

$f_{v_{B-B}} = W/3 * .75 * 3.75/177.28/1.50 = W(.003525)$

$f_{v_{B-B}} = 917 \text{ PSI}$

THE HORIZONTAL SHEAR FORCE IS

$F = f_v t$

$F_{A-A} = 2012 \text{ lb/in}$

$F_{B-B} = 1376 \text{ lb/in}$

$f_{v_{WELD_{A-A}}} = F_{A-A}/(2(.38)(.707)) = $ <u>3,745 PSI</u>

$f_{v_{WELD\,B-B}} = F_{B-B}/(4(.38)(.707)) = $ <u>1280 PSI</u>

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COMBINING STRESSES IN WELDS

THE STRESSES IN FILLET WELDS ARE CONSIDERED SHEAR STRESSES AND SO WILL BE ADDED ALGEBRAICALLY

AT THE END OF THE SPREADER

$f_{v_\Sigma} = f_{v_{max\,(TWISTING)}} + f_{v\,(SHEAR\,FORCE)}$

$= 7{,}545 + 2497$

$f_{v_\Sigma} = 10{,}042$ PSI

AT SEC A-A

$f_{v_\Sigma} = f_{v\,(TWISTING)} + f_{v\,(SHEAR\,FORCE)} + f_{v\,(SHEAR\,FLOW)}$

$f_{v(TWISTING)} = 644.8r$

$r = \left(3.49^2 + [16.48^{(2)} - 9.807]^2\right)^{1/2}$

$r = 7.531$

$f_{v(TWISTING)} = 4856$ PSI

$f_{v_\Sigma} = 4856 + 2497 + 3{,}745$

$f_{v_\Sigma} = 11{,}098$ PSI

(2) SEE SKETCH, PAGE 23.

$f_{v_{WELD\,MAX}} = 11{,}098$ PSI

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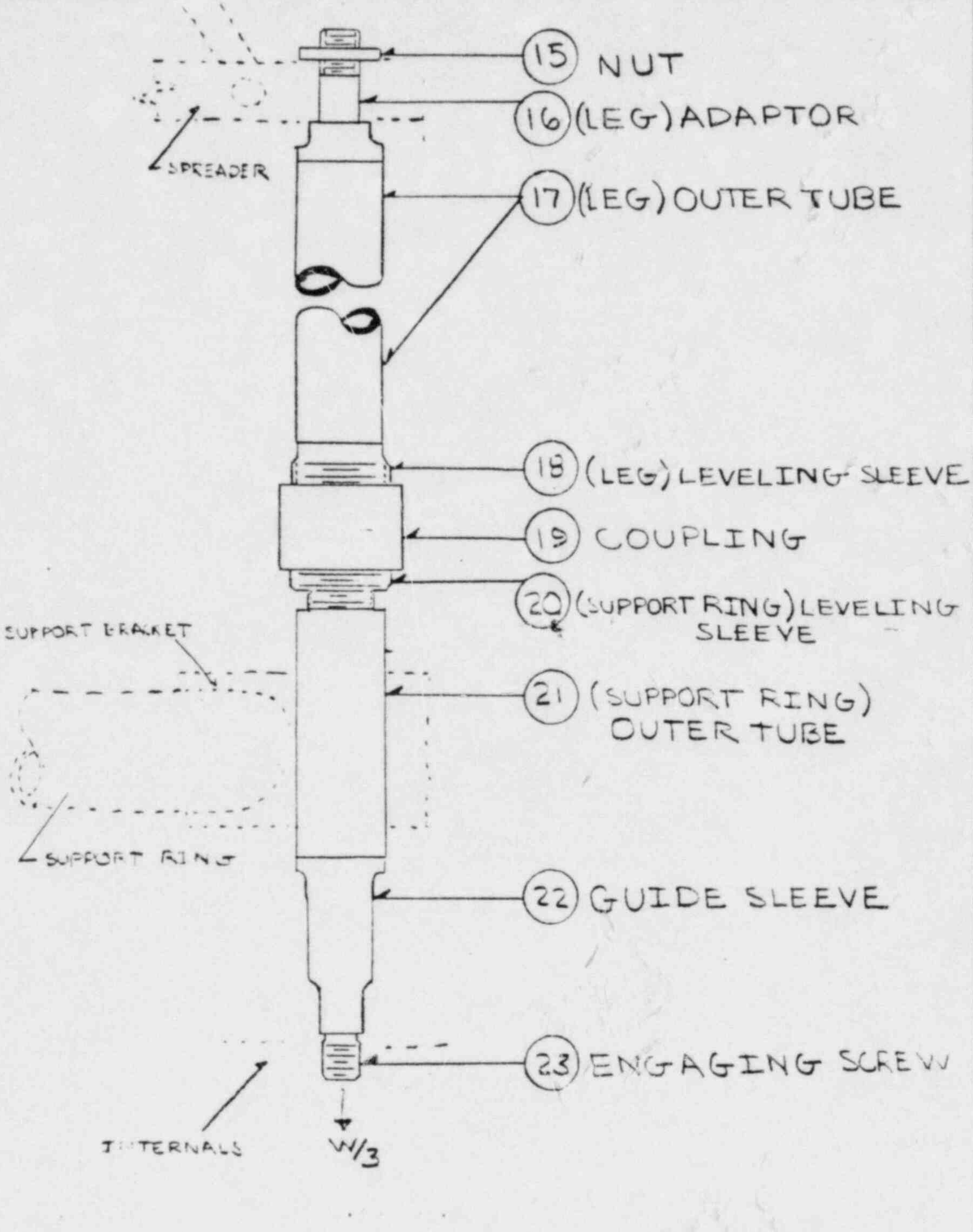

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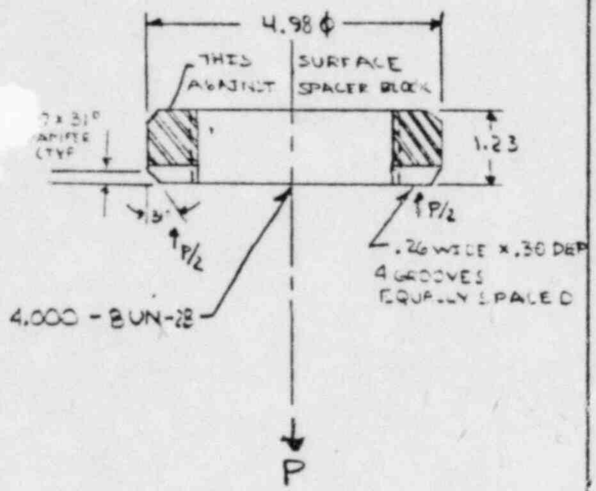
NUT (15)

MAT'L:
ASTM A-276, TYPE 304,
H.R. & PKLD. COND. A



THREAD SHEAR

$f_v = P/A_v = (W/3)/A_v$

$A_v = \pi D_p \ell/2 - 4A_n$

$D_p = D_s - .64952/n$

$= 4.000 - .64952/8$

$= 3.9188$ in

$A_n = A_{notch} = .26 \times .30$

$= .078$

$A_v = \pi(3.9188)(1.23)/2 - 4(.078)$

$= 7.259\ in^2$

$f_v = W/3/7.259$

$= W(.04592)$

$f_v = 11,939$ PSI

BEARING ON SPACER BLOCK (ITEM)

$f_c = P/A_c = (W/3)/A_c$

$A_c = \frac{\pi}{4}(4.98 - 2 * .06/\sqrt{3})^2 - 4.06^2 * \frac{\pi}{4}$

$= 5.994\ in^2$

$f_c = W/3/5.994$

$= W(.05561)$

$f_c = 14,459$ PSI

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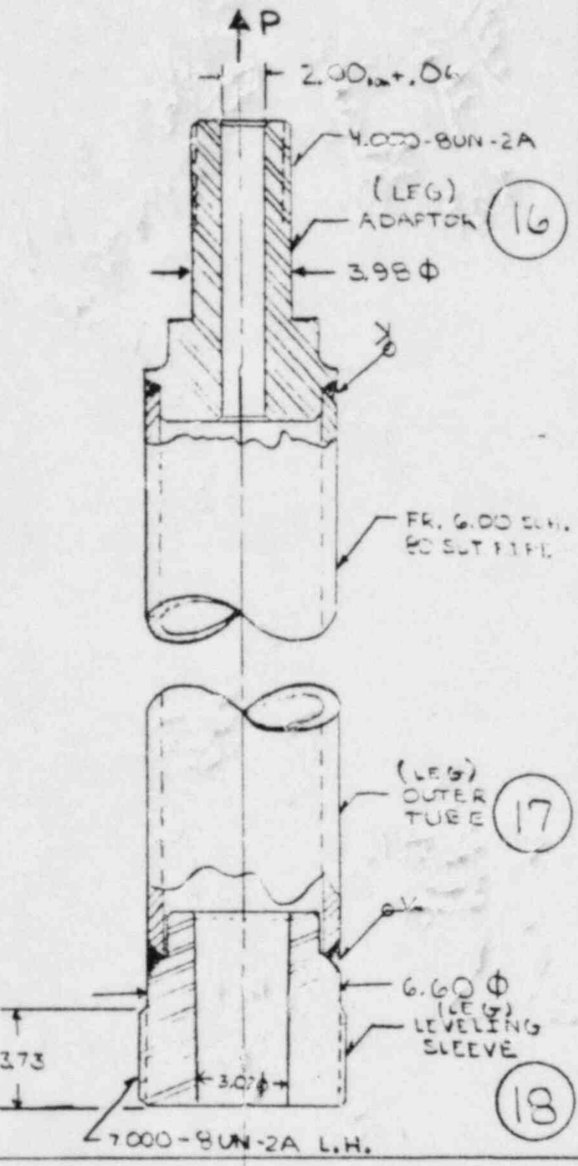
OUTER TUBE ASSEMBLY (16) (17) (18)



MAT'L:

(LEG) LEVELING SLEEVE - ASTM A-276 TYPE 304 HR & PKLD. COND A.

(LEG) OUTER TUBE - ASTM A312 TYPE 304 SMLS CF AND HT. TR.

(LEG) ADAPTOR - ASTM A-276, TYPE 304 H.R. & PKLD. COND A.

(LEG) ADAPTOR (16)

THREAD SHEAR

SAME AS FOR THE NUT (ITEM 15)

$f_v = W(.04592)$

$f_v = 11{,}939$ PSI

TENSION AT THREADS

$f_t = P/A_t = (W/3)/A_t$

$A_t = \frac{\pi}{4}(D_s - .9743/n)^2 - \frac{\pi}{4}(d^2)$

$= \frac{\pi}{4}\left[(4.000 - .9743/8)^2 - 2.06^2\right] = 8.480$

$f_t = W/3/8.480 = W(.03931)$

$f_t = 10{,}221$ PSI

(LEG) OUTER TUBE (17)

TENSION

$f_t = P/A_t = (W/3)/A_t$

$A_t = 8.40\ in^2$ (FROM MARKS HANDBOOK)

$f_t = W/3/8.40 = W(.03968)$

$f_t = 10{,}317$ PSI

(LEG) LEVELING SLEEVE (18)

TENSION AT 6.60 DIA

$f_t = P/A_t = (W/3)/A_t$

$A_t = \frac{\pi}{4}(6.60^2 - 3.02^2) = 27.05$

$f_t = W/3/27.05 = W(.012323)$

$f_t = 3204$ PSI

LEVELING SLEEVE THREAD SHEAR

THREAD SHEAR IS THE SAME AS FOR THE COUPLING (ITEM)

$f_v = W(.015335) = 3987$ PSI

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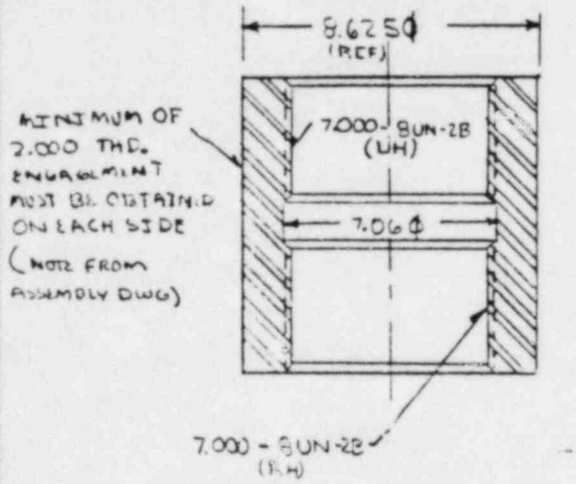
COUPLING (19)



MAT'L:

ASTM A312 TYPE 304
SMLS CF & HT TR.

THREAD SHEAR

$f_v = P/A_v = (W/3)/A_v$

$A_v = \pi D_p \ell/2$

$D_p = D_s - .64952/n$

$= 7.000 - .64952/8 = 6.919 \text{ in}$

$A_v = \pi(6.919)(2.00)/2 = 21.74$

$f_v = W/3/21.74 = W(.015335)$

$f_v = 3987$ PSI

TENSION AT THREAD RELIEF

$f_t = P/A_t = (W/3)/A_t$

$A_t = \frac{\pi}{4}(8.625^2 - 7.06^2) = 19.279 \text{ in}^2$

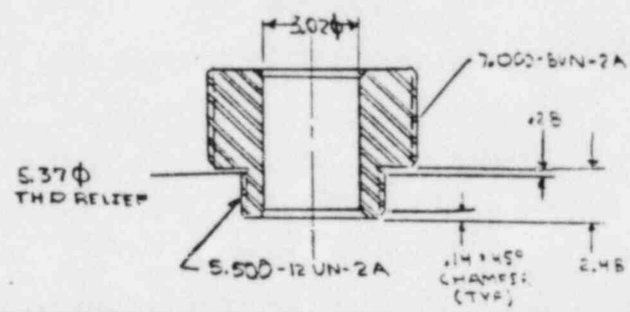
$f_t = W/3/19.279 = W(.017290)$

$f_t = 4495$ PSI

(SUPPORT RING) LEVELING SLEEVE (20)

MAT'L:

ASTM A276 TYPE 304
HR & PKLD. COND A.



THREAD SHEAR ON 7.000-8UN THD.

SAME AS THREAD SHEAR FOR COUPLING (ITEM 19) $f_v = W(.015335)$

$f_v = 3987$ PSI

TENSION AT THD RELIEF

$f_t = F/A_t = (W/3)/A_t$

$A_t = \pi/4\,(5.37^2 - 3.02^2) = 15.485 \text{ in}^2$

$f_t = W/3/15.485 = W(.02153)$

$f_t = 5597$ PSI

THREAD SHEAR ON 5.500-12UN-2A THD.

$f_v = P/A_v = (W/3)/A_v$

$A_v = \pi D_p \ell/2$

$D_p = D_s - .64952/n = [5.500 - .64952/12]$

$\ell = 2.48 - .28 - .14 = 2.06 \text{ in}$

$A_v = \pi(5.5 - .64952/12)(2.06)/2 = 17.622 \text{ in}^2$

$f_v = W/3/17.622 = W(.018916)$

$f_v = 4918$ PSI

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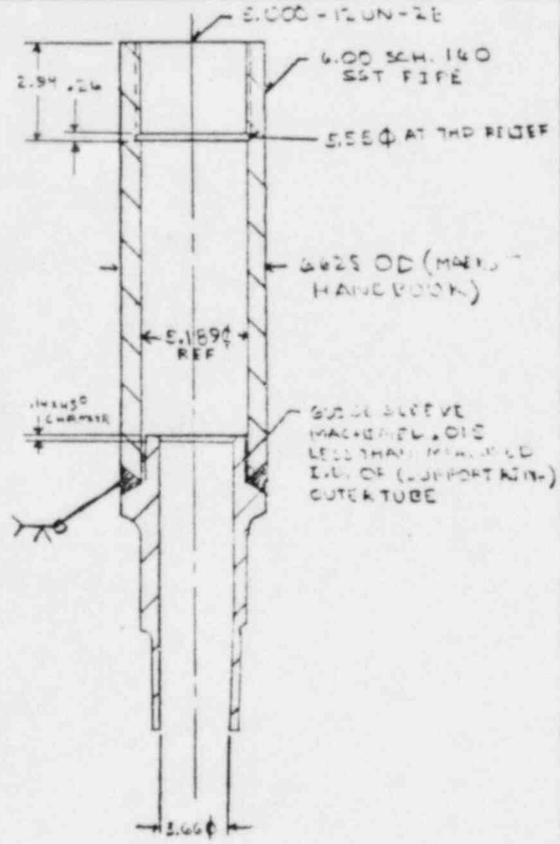
(SUPPORT RING) OUTER TUBE (21)
AND
GUIDE SLEEVE (22)

MAT'L: (SUPPORT RING) OUTER TUBE- ASTM A312 TYPE 304 SMLS CF & HT. TR.
GUIDE SLEEVE- ASTM A276 TYPE 304 HR & PKLD. COND A



(SUPPORT RING) OUTER TUBE (21)

THREAD SHEAR

THE THREAD SHEAR IS THE SAME AS THE THREAD SHEAR AT THE 5.500 THD ON THE (SUPPORT RING) LEVELING SLEEVE (ITEM 20)

$f_s = W(.018916) = 4918$ PSI

TENSION AT THD. RELIEF

$f_t = P/A_t = (W/3)/A_t$

$A_t = \pi/4(6.625^2 - 5.55^2)$

$= 10.279\ in^2$

$f_t = W/3/10.279$

$= W(.03243)$

$f_t = 8432$ PSI

TENSION AT WELD

$f_t = P/A_t = (W/3)/A_t$

A_t = AREA OF PIPE = $13.33\ in^2$

(FROM MARKS HANDBOOK)

f_t W/3/13.33 = W(.025006)

f_t = 6,502 PSI

GUIDE SLEEVE (22)

BEARING ON ENGAGING SCREW

$f_c = P/A_c = (W/3)/A_c$

$A_c = \frac{\pi}{4}\left[\{4.98 - 2(.06)\}^2 - \{3.66 + 2(.19)\}^2\right]$

$= 6.3586\ in^2$

$f_c = W/3/6.3583$

$= W(.05242)$

$f_c = 13,629$ PSI

NOMINAL COMPRESSION BELOW ENGAGING SCREW

$f_t = F/A_c = (W/3)/A_c$

$A_c = [(5.189 - .015)^2 - 3.66^2]\pi/4 = 10.504\ in^2$

$f_t = W/3/10.504 = W(.03173)$

$f_t = 8250$ PSI

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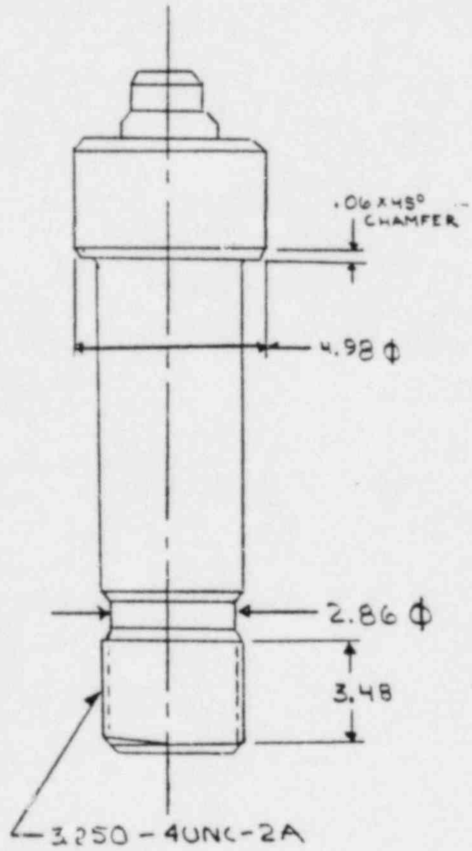
ENGAGING SCREW (23)

MAT'L:
ASTM A-276 TYPE 304
HR & PKLD. COND A.



BEARING ON GUIDE SLEEVE

THE BEARING STRESS IS THE SAME AS THE BEARING STRESS ON THE GUIDE SLEEVE (ITEM 22)

$f_c = W(.05242)$

$f_c = 13,629$ PSI

TENSION AT MINIMUM SECTION

$f_t = P/A_t = (W/3)/A_t$

$A_t = \pi/4\,(2.86)^2 = 6.424\ in^2$

$f_t = W/3/6.424$

$= W(.05189)$

$f_t = 13,491$ PSI

THREAD SHEAR

$f_v = P/A_v = (W/3)/A_v$

$A_v = \pi D_p l/2$

$D_p = D_s - .64952/n$

$= 3.250 - .64952/4$

$= 3.0876\ in$

$l = 3.48\ in$

$A_v = \pi \cdot (3.0876) \cdot 3.48/2$

$= 16.878\ in^2$

$f_v = W/3/16.878$

$= W(.019750)$

$f_v = 5135$ PSI

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APPENDIX C

DETAILED STRESS ANALYSIS - REACTOR COOLANT PUMP MOTOR LIFT RIG

This appendix provides the detailed stress analysis for the Turkey Point Units 3 and 4 reactor coolant pump motor lift sling, in accordance with the requirements of ANSI N14.6. Acceptance criteria used in evaluating the calculated stresses are based on the material properties given in section 5.

SKETCH SHEET
WESTINGHOUSE FORM 54202

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FJIP-93447	Turkey Point Units 3 and 4	1 OF 20

TITLE	CALCULATIONS NO.
R. C. Pump Motor Lift Sling	PDC-

AUTHOR & DATE	CHECKED BY & DATE
M. F. Hankinson 12/82	J. M. Matusz 12/82

PURPOSE AND RESULTS:

1. The purpose of this analysis is to determine the acceptability of this sling to the requirements of ANSI N14.6.

2. The results show that all stresses are acceptable.

COMMONWEALTH OF PENNSYLVANIA
REGISTERED PROFESSIONAL
JOHN M. MATUSZ
ENGINEER
No. 14370-E

J M Matusz 12/82

REVISION NO.	DATE	DESCRIPTION	BY
		Original Issue	M. Hankinson

RESULTING REPORTS, LETTERS OR MEMORANDA:

WESTINGHOUSE NUCLEAR TECHNOLOGY DIVISION

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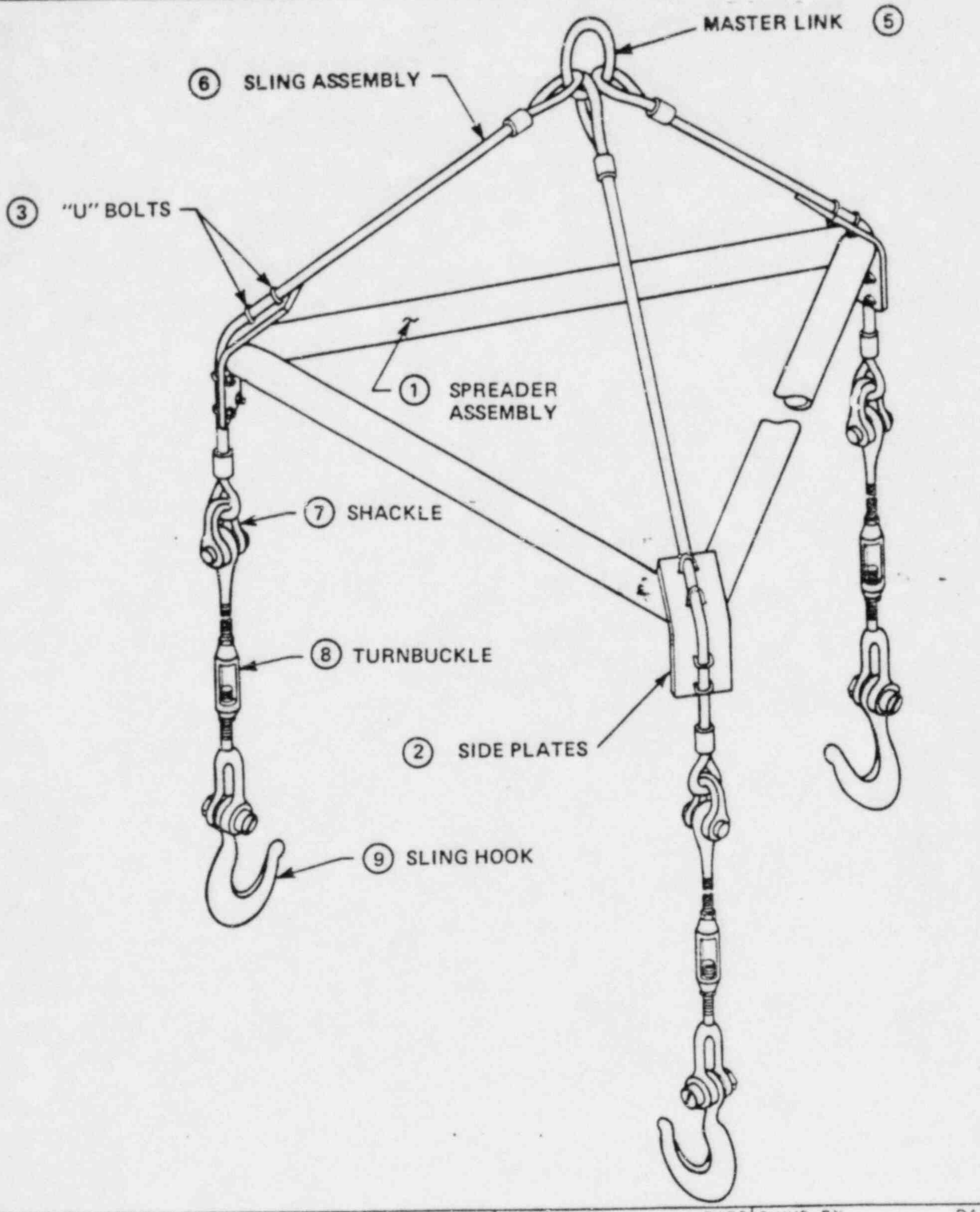

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WESTINGHOUSE FORM 55213D

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Design weight for R.C.P. Motor Sling

weight of Motor - 66,000 to 72,000 lbs.

weight of sling Assembly -	1,000 lbs.
	73,000 lbs.
Plus 10% for contingencies	7,300 lbs.
	80,300 lbs

Use 81,000 lbs. which is rated lift of sling for design weight.

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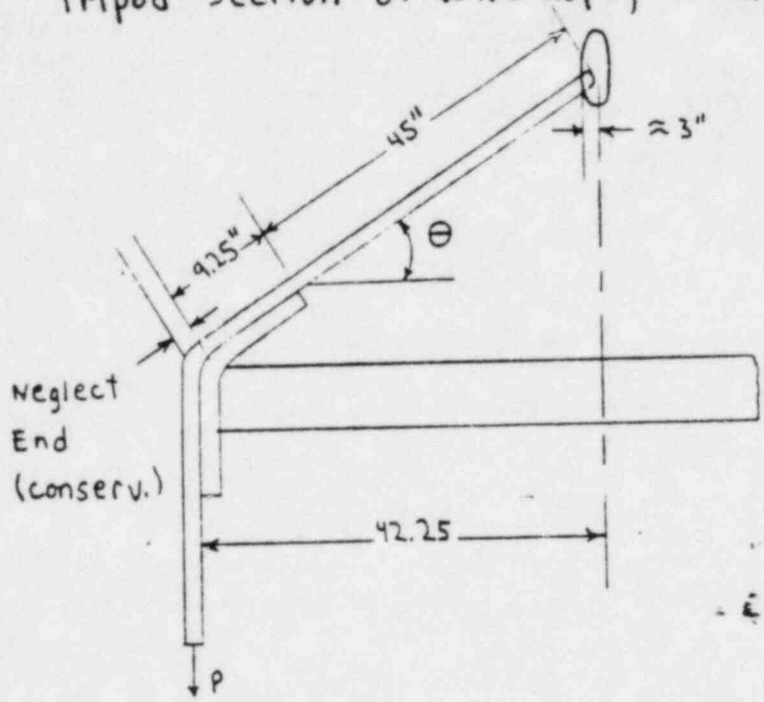
I Establish Tension Value In Tripod Section of wire Rope, Tmax



w = Rated Lift weight of RIG = 81,000 lbs.

with no information to the contrary, assume load is approximately centered, i.e.

P = Tension in each vertical rope = $\frac{W}{3}$

= 81,000 lbs. / 3 = 27,000 lbs.

$$\theta = \cos^{-1}\left(\frac{42.25-3}{45+9.25}\right) = \cos^{-1}\left(\frac{39.25}{54.25}\right) = 43.66^\circ$$

$T_m = P/\sin\theta = 27{,}000/.6903 = 39{,}109$ lbs.

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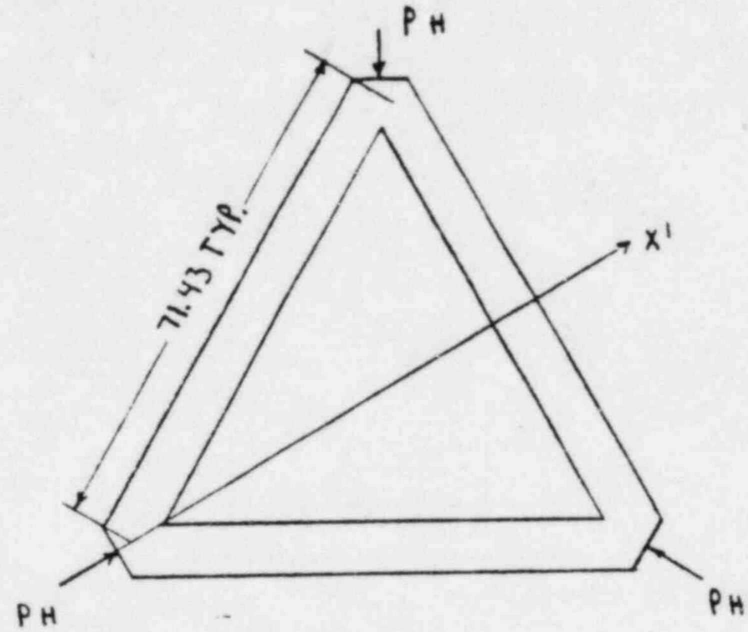
SPREADER ASSY.



P_H = Horizontal Force on Speader
(see page 4)

$= T_m \cos\theta = 39109 \cos(43.66) = 28293$ lb

Let F_c = compressive force in speader

$\Sigma F_{x'} = 0 : 2(.866)F_c - P_H = 0$

$F_c = \frac{28293}{2(.866)} = 16,335$ lb

For 4" SCH 40 PIPE OD = 4.500

Wall = .237", ID = 4.500 - .237 x 2 = 4.026"

$A = AREA = \pi/4\,(4.5^2 - 4.026^2) = 3.174\ in^2$

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The compressive stress is fc

$$f_c = \frac{16335\ lb}{3.174\ in^2} = 5146\ psi$$

For ASTM A106 GRB

σ_y = yield str. = 35 KSI
σ_u = ultim str. = 60 KSI

Check compressive allowable

From AISC CODE SEC 1.5.1.3

$$C_c = \sqrt{\frac{2\pi^2 E}{F_y}} = \sqrt{\frac{2(3.1416)^2(29\times10^6 psi)}{35,000\ psi}}$$

$= 127.89$

$\ell = 71.43$ in

$$I_{bend} = \frac{\pi}{64}(d_o^4 - d_i^4) = \frac{\pi}{64}(4.500^4 - 4.026^4) = 7.232\ in^4$$

r = RAD. OF GYR.

$$= \sqrt{\frac{I_B}{A}} = \sqrt{\frac{7.232}{3.174}} = 1.510$$

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For Ideal fixed ends, K = .5

This symmetric case probably approaches K = .5

use K = .65 for conservativism

$$\frac{Kl}{r} = \frac{.65(71.43)}{1.510} = 30.75 < Cc$$

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Since $\frac{kl}{r} < C_c$:

F_a = Max Comp. stress

$$= \frac{\left[1 - \frac{(kl/r)^2}{2C_c^2}\right] F_y}{\frac{5}{3} + \frac{3(kl/r)}{8C_c} - \frac{(kl/r)^3}{8C_c^3}}$$

$$= \frac{\left[1 - \frac{(30.75)^2}{2(127.89)^2}\right] F_y}{\frac{5}{3} + \frac{3}{8}\frac{(30.75)}{127.89} - \frac{(30.75)^3}{8(127.89)^3}}$$

$$= \frac{[1 - .0289] F_y}{\frac{5}{3} + .091 - .0017}$$

$$= \frac{[1 - .0289] F_y}{1.755} = .553 F_y = .533(35 \text{ KSi})$$

$$= 19.35 \text{ KSi}$$

$f_c < F_a$

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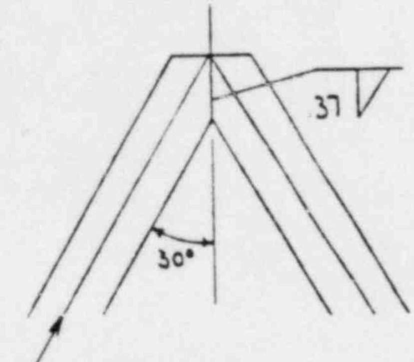
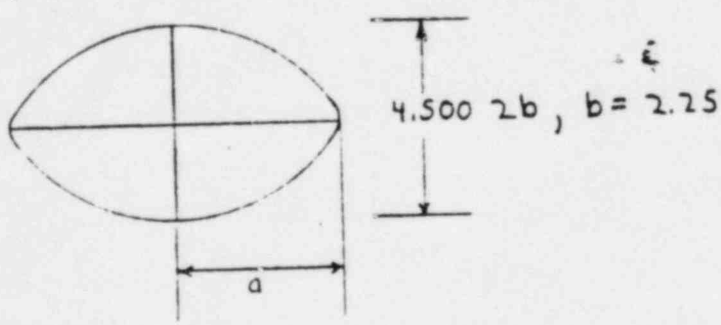
S.O.: FJIP-93447	CALC. NO.:	FILE NO.:	GROUP: CHE

Analyze compressive stress on Tube to Tube Weld



16,942 (page 5)

The length of weld will be length of a half ellipse (projection of ID of pipe)



$$a = \frac{1}{2}\left[\frac{4.50}{\sin 30^\circ}\right] = 4.50 \qquad m = \frac{a-b}{a+b} = \frac{2.25}{6.75} = .333$$

From Marks Handbook, 7 TH ED. Pages 2-18

L = Length of perimeter of half-ellipse

$$= \tfrac{1}{2}[\pi(a+b)k] = \tfrac{1}{2}[\pi(6.750)(1.029)] = 10.91 \text{ in}$$

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The throat area of the weld is:

$$A_t = (10.91)(.707)(.37) = 2.854 \text{ in}^2$$

The compressive force on the weld

$$F_c = 16335(\sin 30^\circ) = 8167 \text{ lbs.}$$

The compressive stress on the weld

$$f_c = \frac{8167}{2.854 \text{ in}^2} = 2861 \text{ psi}$$

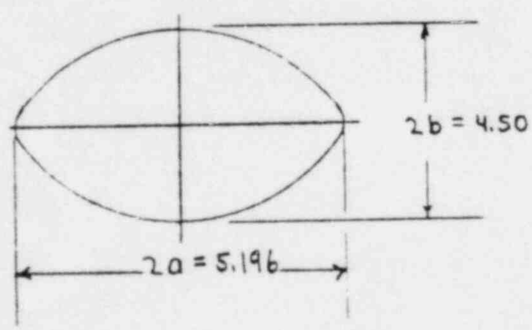
F_c = Max allowable stress (AISC, 8th ED, Table 1.5.3)

= Same as base metal

$= .60F_y = .60(35000) = 21000$

$\therefore f_c < F_c$

Check Plate to Tube Weld



$$2a = \frac{4.500}{\cos 30^\circ} = 5.196$$

$a = 2.598$

$2b = 4.5$

$b = 2.25$

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$$m = \frac{2.598 - 2.25}{2.598 + 2.25} = .0717$$

L = Perimiter of Ellipse

$= \pi (2.598 + 2.25) K$

$= \pi (4.848)(1.00K)$

$= 15.246$ in

$A_t = (15.246)(.707)(.37) = 3.988 \text{ in}^2$

F_c = (Page 5) Compressive force on weld

$= 29344$ lb

f_c = Compressive stress on weld

$$= \frac{28293 \text{ lb}}{3.988 \text{ in}^2} = 7094 \text{ psi}$$

$f_c < .60(35000 \text{ psi}) = 21000$ psi ; OK

Note that we assumed on page 9 that the tube-to-tube compression would be taken by the tube-to-tube weld. Even if this force went through the tube to plate weld the max stress would be:

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T= Shearing stress

$$= \frac{8167}{1.994\ in^2} \text{ (Page 10) (Half of tube to Pl. weld)}$$

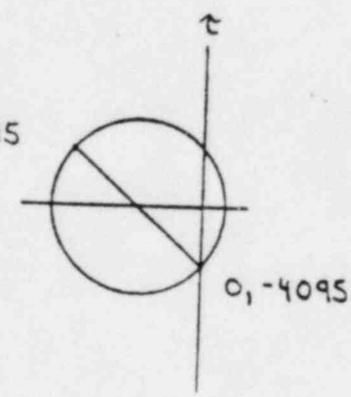
$= 4095\ psi < .4\sigma_y = .4(35000) = 14000$

$$\sigma_{1,2} = \frac{0-7094}{2} \pm \sqrt{\left(\frac{7094}{2}\right)^2 + 4095^2}$$

$= -3547 \pm 5417$

$= -8964,\ 1870\ psi$

$\sigma_{1,2} < .60\sigma_y = 21,000\ psi$ ok



The force trying to slide the U-bolts is the difference between the rope tensions. (No credit for friction) This force will also put the plate in tension.

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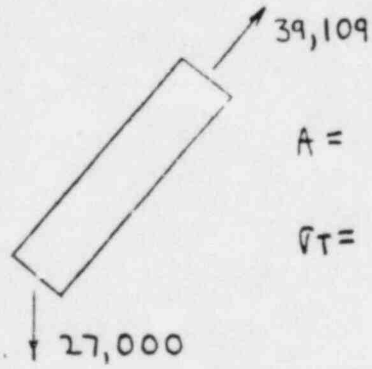
SIDE PLATE

Tensile strength of Plate - (Assume taken by top bolts)

$F_{vB} = 39,109 - 27,000 = 12,109$ lb.



$A = 6\frac{1}{2}" \times 1" = 6.50\ in^2$

σ_T = Tensile stress = $\frac{F_{vB}}{A}$

$= \frac{12109\ lb}{6.50\ in^2} = 1863$ psi

From AISC Rules, allowable stress for members with bolts holes is $.45\,\sigma_y$

$\sigma_A = .45 \times 30,000 = 13,500$ psi

$\sigma_T < \sigma_A \therefore$ OK

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The following items are bought from sling suppliers per the referenced drawing. Stress calculations are not performed since material requirements are, for most items, unknown.

The attached Bill of Materials describes these items with the available information and lists design loads together with the catalog and/or drawing rated loads.

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TITLE: R. C. Pump Motor Lift Sling

PROJECT: FPL/FLA | AUTHOR: M F Hankins DATE 12/82 | CHK'D. BY: J M Matney DATE 12/82 | CHK'D. BY: DATE

S.O.: FJIP-93447 | CALC NO: | FILE NO.: | GROUP: CHE

TABLE 1: BILL OF MATERIALS FOR NON-DESIGN ITEMS OF THE R.C. MOTOR LIFT SLING

NO.(1)	Item	DESCRIPTION: DRAWING(2)	DESCRIPTION: CATALOG(8)	RATED(1) LOAD VALUE (POUNDS)	DESIGN LOAD (POUNDS)
3	"U" Bolts	$\frac{7}{8}$" "U" Bolts + Cable Saddle	(6) "U" Bolts - G-450 $1\frac{1}{2}$" Dia. Wire $\frac{7}{8}$" Dia. "U" Bolt wire rope clips	——	——
5	MasterLink	$2\frac{3}{4}$" x 9" x 18"	(3)(4) $2\frac{3}{4}$" x 9" x 16"	160,000	81,000
6	Sling	$1\frac{1}{2}$" x 11'4" (Total inc. turnbuckle, shackle, etc.) 6x7 Improved Plow I.W.R.C 3 Bridle Sling (7'6" sling length only incl. eye)	—— (5)	81,000(9)	81,000 Each leg 39,000
7	Shackle	Anchor Safety Shackle	2" size (6) G-213 or S-213	70,000	27,000
8	Turnbuckle	2" x 6" Jaw & Eye turnbuckle	G-227 Jaw & eye (6) turnbuckle 2" Dia. thd., 24" take-up Class 8	37,000	27,000

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FJIP-93447			CHE

TABLE 1 : BILL OF MATERIALS
FOR NON-DESIGN ITEMS OF THE R.C. MOTOR LIFT SLING

NO.(1)	Item	DESCRIPTION: DRAWING(2)	DESCRIPTION: CATALOG(3)	RATED(7) LOAD VALUE (POUNDS)	DESIGN LOAD (POUNDS)
9	HOOK	TAYCO A 73	TAYCO sling hook (4) A-73, 1" Dia. chain with 2 1/2" Dia. eye	38,750	27,000

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NOTES : TABLE 1

(1) See page 2 for identification of item no's

(2) Description is from Westinghouse drawing AED SK 618J644 TXK SUB 5

(3) Actual purchase order changed link to $2\frac{3}{4}$" x 9" x 16.

(4) S.G. Taylor Chain Co. Inc., Bulletin AS-67 alloy steel chain assemblies, attachments.

(5) Pennsylvania Sling Co.

(6) Crosby Group, 950 General Catalog, June 1981

(7) Rated load value: The maximum recommended load that should be exerted on the item. The following terms are also used for the term Rated Load: "SWL", "Safe working Load", "working Load Limit", and the "Resultant Safe Working Load". All rated load values are for in-line pull with respect to the centerline of the item. Information is from catalogs identified in (4), (5), (6).

(8) Information from catalogs may not be identical to that which is INSTALLED. HOWEVER, THIS INFORMATION IS ALL THAT IS AVAILABLE AND IS REPRESENTATIVE.

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NOTES: Continued

(9) Information from the note 2 drawing states: The safe working load of this sling assembly is 81,000 lb and a safety factor of 5:1".

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The rated loads of all items are greater than the design loads. Catalog information on the sling is questionable, since the design is a special sling. However, reviewing the PENCO sling catalog (circa 1975) shows that for 3 legs, lift angle of 45°, 6X37, 1X½ Dia. bridle slings, the safe working LOAD is 60,400 POUNDS. (Latest catalog information (1982) states the S.W.L. is 74,000 pounds.) However, this CATALOG sling rating may not be for the special manufactured sling described on the W drawing. The W drawing states that the S.W.L. for the sling is 81,000 lb with a safety factor of 5:1 with a proof test of 47 tons (94,000 pounds) at assembly.

For comparison, the ANSI B 30.9 -1971 Slings table 8- for 3 leg bridle sling, 6X37 IWRC, 1½" Dia.

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45 degree angle with mechanical splice rating is 37 ton (74,000 pound). The proof load (section 9.2.3.1) is twice the single leg vertical capacity which is (Table 4) 17 tons (34,000 pounds). This table 4 value indicates that the slings are acceptable.